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# canadian acoustics
# acoustique canadienne

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## PROCEEDINGS ISSUE

ACOUSTICS WEEK IN CANADA
SEMAINE CANADIENNE D'ACOUSTIQUE
1993

## 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.

**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** 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.

**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.

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# PRESIDENT'S MESSAGE

This will be my last President's Message, as the CAA membership will be choosing a new President at the October elections in Toronto. Some of you have wondered why my term has lasted only 2 years instead of the "traditional" 3 years. There is no mystery; I can only say that I am following the fine example set by my predecessor! Of course, as Past President, I will remain an Officer of the CAA and I will continue to be involved in several projects.

The outstanding issues in the CAA continue to be declining membership, under-involvement of students, and lack of submissions to our journal, *Canadian Acoustics*. If all goes well, those who receive this issue by mail should find a copy of the new CAA brochure. If everyone passed on their copy to someone who is not a member or who is a lapsed member, this might help address the problem of declining membership. By the way, many thanks to Marek Roland-Mieszkowski and Frédéric Laville for producing the brochure. The information is in a word processor, so it should be easy to keep the brochure up-to-date.

I still find it astonishing that many of the CAA prizes (which are mostly for students) are not awarded due to lack of applications. To those of you who supervise students or are in a position to encourage student participation in the Association, please do your best to publicize the CAA awards, including the Directors' Award for the best student paper in *Canadian Acoustics*. The future of the CAA will be in their hands.

Speaking of *Canadian Acoustics,* it may be time for the CAA to review the function of the journal in the Association. Your views are welcome, but I am sure the editor would prefer manuscripts! As a researcher who has published in both *Canadian Acoustics*, and other journals, I recognize that it is tempting to publish in the journals with wider circulation. At the same time, I think we all can identify aspects of our work that may be more appropriate for publication in our national acoustics journal, especially if the topic may appeal to a broad spectrum of acousticians.

Thanks to all those who have contributed their time and effort over the last couple of years, and to those who continue to serve. Through involvement with the CAA, I have established several friendships that I am certain will endure. See you in Toronto!

# MOT DU PRÉSIDENT

Voici mon dernier message á titre de Président puisque l'ACA se choisira un nouveau Président lors des élections d'octobre à Toronto. Certains d'entre vous se demanderont pouquoi mon terme n'a duré que deux ans plutôt que le "traditionnel" trois ans. Il n'y a pas de secret: je peux tout simplement dire que je suis l'exemple de mes prédécesseurs. À titre de Président sortant je demeurerai toutefois membre de l'Exécutif et m'impliquerai encore dans plusieurs projets de l'ACA.

Les faits marquants de l'ACA demeurent la baisse d'adhésion de nouveaux membres, l'implication peu marquée des étudiants et le peu de soumissions de publications à notre journal, *l'Acoustique Canadienne*. Si tout se déroule bien, ceux qui reçoivent ce numéro par la poste devrait trouver une copie de la nouvelle brochure présentant l'ACA. Si chacun d'entre vous remettait cette copie à un collègue qui n'est pas membre ou qui est un ancien membre, ce geste pourrait aider à "solutionner" le problème de la baisse d'adhésion. Je profite de l'occasion pour transmettre mes remerciements à Marek Roland-Mieszkowski et Frédéric Laville pour leur travail de conception de la brochure. Les informations sont conservées dans un traitement de texte permettant ainsi de tenir la brochure à jour.

Je suis toujours surpris de constater que plusieurs des prix de l'ACA (qui s'adressent particulièrement aux étudiants) ne sont pas décernés à cause d'un manque de candidats. Je m'adresse à ceux qui supervisent des étudiants ou qui sont en position d'encourager la participation étudiante dans l'Association: faites votre possible pour publiciser les prix de l'ACA, incluant le Prix des Directeurs pour le meilleur article étudiant publié dans *l'Acoustique Canadienne*. L'avenir de l'ACA est entre leurs mains.

Concernant *l'Acoustique Canadienne*, il est peut-être temps pour l'ACA de se questionner sur la fonction du journal de l'association. Votre point de vue est bienvenu mais je suis persuadé que le rédacteur préférerait recevoir des manuscrits! A titre de chercheur qui a publié dans *l'Acoustique Canadienne* et dans d'autres revues, je reconnais qu'il est plus attrayant de publier dans des journaux à grand tirage. En même temps, je pense que nous pouvons tous identifier des aspects de notre travail qui puissent être appropriés pour une publication dans notre journal national d'acoustique, principalement si le suject peut intéresser un large éventail d'acousticiens.

Je remercie tous ceux qui ont consacré du temps et des efforts au cours des dernières années et tous ceux qui continuent de s'impliquer. Grâce à mon implication dans l'ACA, j'ai établi des liens d'amitié qui dureront longtemps. Au plaisir de vous rencontrer à Toronto!

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# PROCEEDINGS OF ACOUSTICS WEEK IN CANADA 1993
# ACTES DE LA SEMAINE CANADIENNE D'ACOUSTIQUE 1993

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**ACOUSTICAL MEASUREMENT & COMPUTATIONAL & ELECTRO-ACOUSTICS**



**NOISE BYLAWS**



**PSYCHOPHYSICS & PHYSIOLOGICAL ACOUSTICS**



**HUMAN RESPONSE TO VIBRATION**

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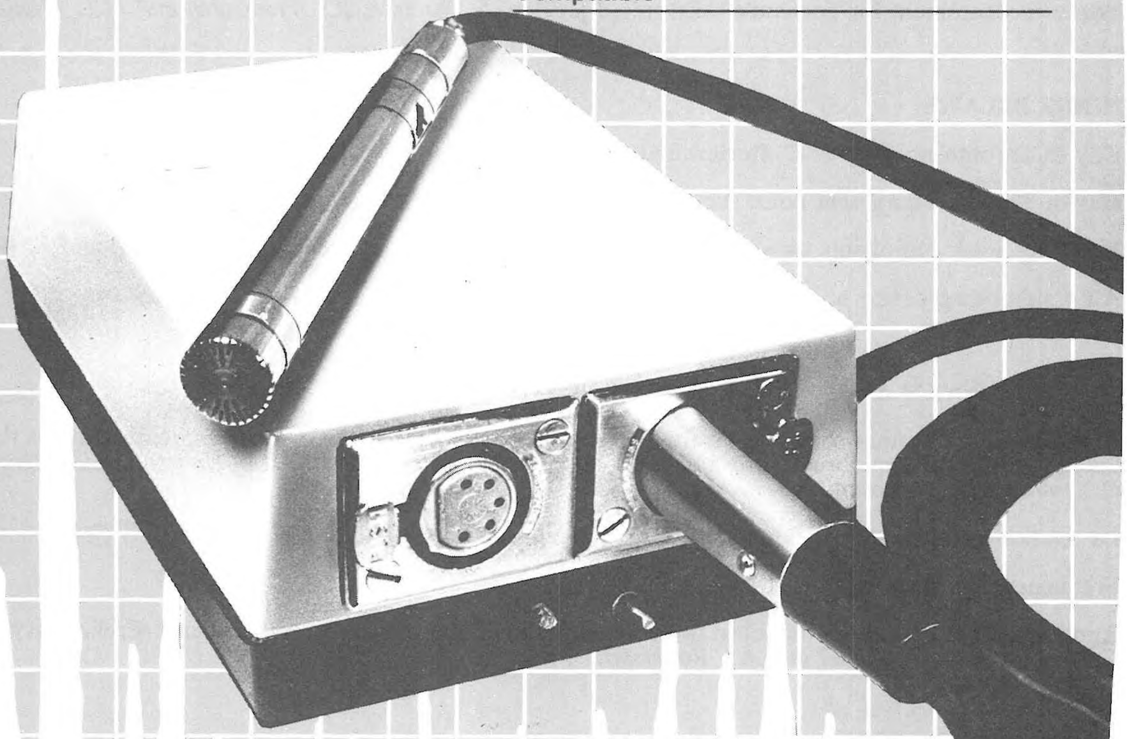

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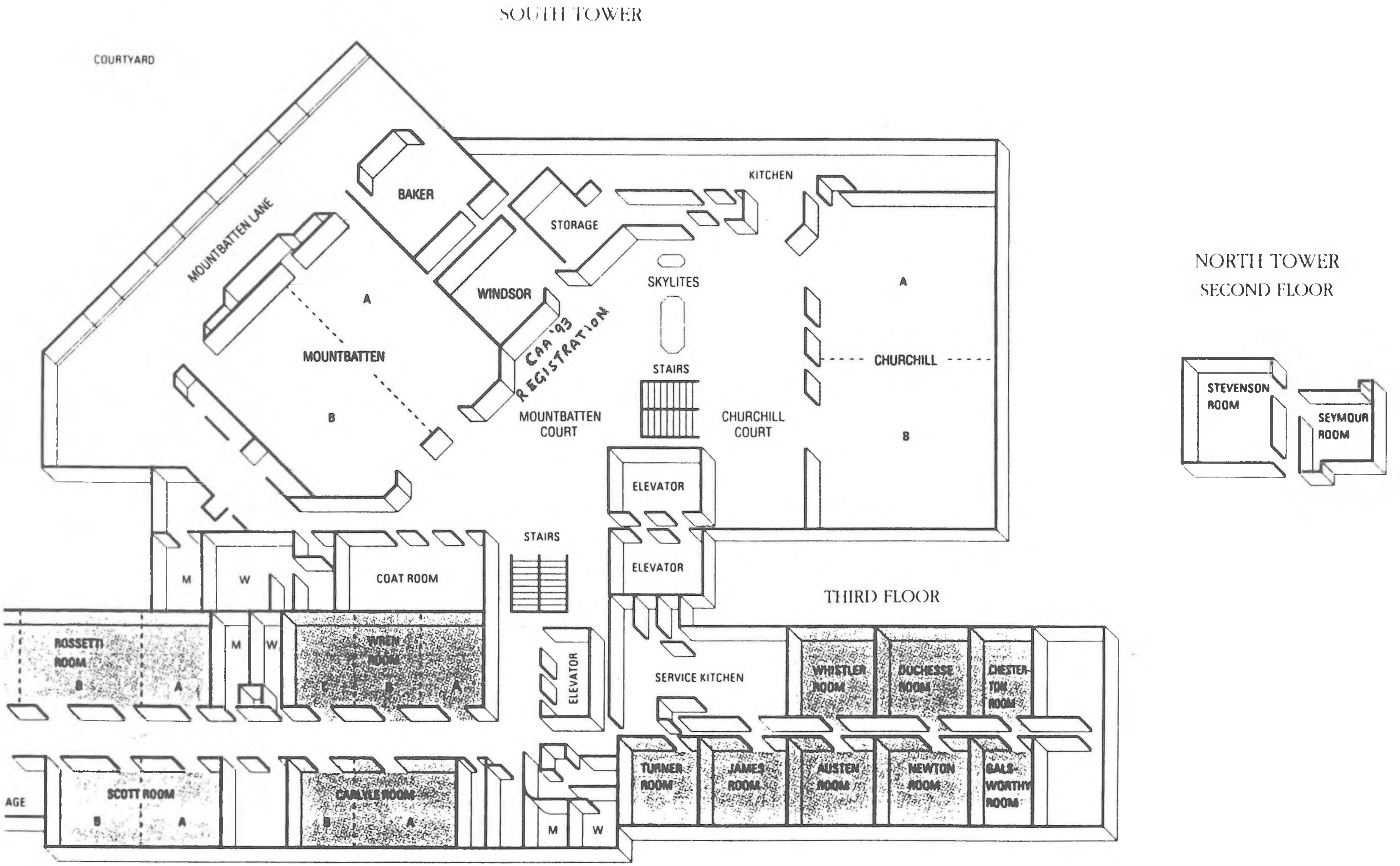
# CAA 93 - TORONTO
# FINAL SYMPOSIUM PROGRAM OCT 6-8, 1993

| | WEDNESDAY | | | THURSDAY | | | FRIDAY | |
|---|---|---|---|---|---|---|---|---|
| 8:30-9:15 | Mountbatten A - Prof. W. Richarz "The Acoustics of Thunder with An Acoustical Demo" | | | Mountbatten A - Mr. G. Rasmussen "Progress in Measurement Techniques in Acoustics" | | | Baker - Mr. B. Arnott "The Past and Future of Massey Hall" | |
| 9:15-9:40 | Mountbatten A Rm | Baker Rm | Windsor Rm | Mountbatten A Rm | Baker Rm | Windsor Rm | Windsor Rm | Baker Rm |
| 9:40-10:00 | Hearing in the Workplace | Land Use Planning | Speech Production & Speech Percep. | Acoustics in Rehabilitation | Noise in the Workplace Part II | Acoustical Measurement & Computational & Electro-Acoustics | Origins of CAA | Human Response to Vibration |
| 10:00-10:20 | | | | | | | | |
| 10:20-10:40 | | | | | | | | |
| 10:40-11:00 | ** C O F F E E ** | | | ** C O F F E E ** | | | ** C O F F E E ** | |
| 11:00-11:20 | Chair Raymond Hetu | Chair Leslie Kende | Chair Elzbieta Slawinski | Chair Don Jamieson | Chair Don Allen | Chair Richard Guy | Chair John Manuel | Chair Tony Brammer |
| 11:20-11:40 | | | | | | | | |
| 11:40-12:00 | | | | | | | | |
| 12:00-1:20 | LUNCH | | | LUNCH | | | Mountbatten A<br>AWARDS<br>Luncheon | |
| 1:20-1:40 | cont'd | Architectural & Performance Acoustics | Musical Acoustics | cont'd | Noise By Laws | Psychophysics & Physiological Acoustics | | |
| 1:40 - 2:00 | | | | | | | | |
| 2:00-2:20 | | | | | | | | |
| 2:20-2:40 | | Chair John O-Keefe | Chair Lola Cuddy | | Chair Chris Andrew | Chair Robert Harrison | | |
| 2:40-3:00 | ** C O F F E E ** | | | ** C O F F E E ** | | | | |
| 3:00-3:20 | Noise in the Workplace Part I | | Machinery Vibration | Mountbatten A<br>ANNUAL MEETING | | | | |
| 3:20-3:40 | | | | | | | | |
| 3:40-4:00 | | | | | | | | |
| 4:00-4:20 | Chair Jean Nicolas | | Chair Tom Moore | | | | | |
| 4:20-4:40 | | | | | | | | |

Tues., Oct 5, 5:30 - 7:30 p.m. Welcome Wine & Cheese Mountbatten A/B
Wed., Oct 6, 5:30 - 6:30 p.m. Reception/Cash Bar, Mountbatten Lane
6:30 - 9:00 p.m. Banquet, Mountbatten A

# THE DELTA CHELSEA INN - FLOOR PLAN



# ACOUSTICS WEEK IN CANADA
# SEMAINE CANADIENNE DE L'ACOUSTIQUE

## REGISTRATION FORM/FORMULAIRE D'INSCRIPTION

**4-8 October, 1993**
**Delta Chelsea Inn, 33 Gerrard West**
**Toronto, Ontario**
**Canada**

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**SYMPOSIUM/CONGRES**
**6-8 October 1993**

| | MEMBERS/MEMBRES | STUDENTS/ ETUDIANT(E)S |
|---|---|---|
| REGISTRATION/INSCRIPTION | | |
| -before August 15/avant le 15 août | $125 | $20 |
| -after August 15/après le 15 août | $150 | $25 |
| BANQUET TICKET/BILLET DE BANQUET | Incl. | $35 |
| - additional, $35 ea./en plus, $35 | $_____ | $_____ |
| TOTAL/SOMME | $_____ | $_____ |

**TOURS** - Two afternoon tours of CBC have been arranged. If you are interested, please indicate your prefered date. Deux visites pendent l'après-midi avant ete organises pour Radio Canada. Si vous êtes intéressés, indiquez S.V.P. la date préferée 5 Oct. ______ or/ou 8 Oct. ______ (Limit: 20).

**SEMINARS**
- Intensity Measurement Techniques, 5 Oct. __________
- Modern Techiques in Estimating and Troubleshooting HVAC Noise, 4-5 Oct. __________

**HOTEL RESERVATION** - Delta Chelsea Inn ($95, single or double/un ou deux personnes)
Date of Arrival/Date d'arrivée:______________________________
Date of Departure/Date de départ:____________________________
Name(s)/Nom(s): (1)____________________ (2)____________________

**Please make cheques payable in Canadian funds to CAA '93 (Toronto) and mail to:**
**S.V.P. faites vos chèques à l'ordre de CAA'93 en fonds canadiens et postez à:**

**Dr. Sharon Abel**
**Mount Sinai Hospital, Room 843**
**600 University Avenue**
**Toronto, Ontario**
**M5G 1X5**

# PROGRAMME

## WEDNESDAY MORNING, OCTOBER 6, 1993

**************************************************************************************

**8:30-9:15 a.m. PLENARY SESSION Mountbatten A**

**H. S. Ribner and the Acoustics of Thunder**

**Werner G. Richarz**

**Associate Professor**
**Dept. of Mechanical & Aerospace Engineering**
**Carleton University**

**************************************************************************************

### *ROOM - MOUNTBATTEN A*

### HEARING IN THE WORKPLACE
### PART I

**Chair: Raymond Hetu**

9:40-10:00 K. Momtahan, R. Hetu, and B. Tansley
Audibility and identification of auditory alarms in the operating room and intensive care unit.

10:00-10:20 H. Kunov, H. Dajani, and B. Seshagiri
Real-time method for measurement of noise exposure from sources in close proximity to the ear.

10:20-10:40 M. Roland-Mieszkowski, D. Roland-Mieszkowski, A. Czyzewski, and B. Kostek
Digital computer-based audiometry: Technical considerations and clinical applications.

10:40-11:00 ***Coffee***

11:00-11:20 J.G. Ryan, E.A.G. Shaw, A.J. Brammer, and G. Zhang
Enlosure for low-frequency assessment of active noise reducton circumaural head sets and hearing protectors.

11:20-11:40 S.M. Abel
Indices of communication handicap in mildly hearing-impaired listeners.

11:40-12:00 R. Hetu, H. Tran Quoc, and Y. Tougas
The hearing aid as warning signal receiver in noisy workplaces.

## *ROOM - BAKER*

## LAND USE PLANNING

### Chair: Leslie Kende

9:40-10:00 C. Andrew, and J. Prashad
The noise impact statement process in Toronto.

10:00-10:20 A.D. Lightstone
Practical aspects of implementing sound isolation and noise control.

10:20-10:40 T. Lowrey
Aircraft noise management at Canadian airports.

10:40-11:00 ***Coffee***

11:00-11:20 D. Chin-Quee, T. Harding, and A.D. Lightstone
Implementation of noise control measures.

11:20-11:40 A.D. Lightstone, and J. Emeljanow
Environmental noise aspects of landfill site selection.

11:40-12:00 C.A. Krajewski
Uncertainty in prediction of environmental noise immission due to ground effect.

## *ROOM - WINDSOR*

## SPEECH PRODUCTION & SPEECH PERCEPTION

### Chair: Elzbieta Slawinski

9:40-10:00 E.B. Slawinski
Perception and production of syllable-initial English /r/ and /l/ by English and Japanese speakers.

10:00-10:20 S. Rvachew, and E.B. Slawinski
The role of the auditory environment in the development of speech production abilities during infancy.

10:20-10:40 D. O'Shaughnessy
Handling false starts in recognition of spontaneous speech.

10:40-11:00 ***Coffee***

11:00-11:20 J.F. MacNeil, and E.B. Slawinski
Discrimination of rising and falling frequency glides as a function of onset and offset frequency.

11:20-11:40 S. Shaiman, M.D.Z. Kimelman, and S.G. Adams
Kinematic analysis of coarticulation and speaking rate.

11:40-12:00 M.D.Z. Kimelman, S. Shaiman, S.C. Adams, and C.A. Skory
Voice onset time and vowel duration across multiple speech rates in normal and Parkinsonian speakers.

# WEDNESDAY AFTERNOON, OCTOBER 6, 1993

## *ROOM - MOUNTBATTON A*

## HEARING IN THE WORKPLACE
## PART II

### Chair: Raymond Hetu

1:20- 1:40 C. Laroche, M.J. Ross, L. Lefebvre, R. Larocque, R. Hetu, and A. L'Esperance.
Sound propagation of reverse alarms used on heavy vehicles.

1:40- 2:00 A. Behar
Hearing protectors' ratings and the user.

2:00- 2:20 G.E. Clunis
Proposed state-of-the-art criteria for office building acoustics.

2:20- 2:40 E.A.G. Shaw
Occupational noise exposure: A broad informal review of regulations, guidelines and practices in Canada.

2:40- 3:00 ***Coffee***

## NOISE IN THE WORKPLACE
## PART I

### Chair: Jean Nicolas

3:00- 3:20 J. Nicolas, and N. Atalla
Acoustic design for noise reduction.

3:20- 3:40 O. Foin, N. Atalla, and J. Nicolas
A formulation for the vibro-acoustic behaviour of a rectangular plate with constrained-layer damping.

3:40- 4:00 S. Brunet, A. Berry, Nicolas, and Y. Champoux
Active noise control simulations in a cavity-backed flexible plate system.

4:00- 4:20 N. Lessard, F. Laville, and F. Charron
An acoustic design method for light weight structures supporting a vibrating source.

4:20- 4:40 B. Gosselin, and J.C. Fortin
Noise management at Hydro-Quebec construction sites.

*ROOM - BAKER*

## ARCHITECTURAL & PERFORMANCE ACOUSTICS

### Chair: John O'Keefe

1:20- 1:40 S. Norcross, J.S. Bradley, and R.E. Halliwell
Errors incurred using a dummy head to make omni-directional room acoustics measurements.

1:40- 2:00 J.S. Bradley, and G. Soulodre
Objective comparisons of Massey Hall and Boston Symphony Hall.

2:00- 2:20 G. Soulodre, and J.S. Bradley
Subjective comparison of Massey and Boston Symphony Hall.

2:20- 2:40 W.T.Chu
An apparatus for measuring the dynamic stiffness of materials used under floating floor.

2:40- 3:00 *Coffee*

3:00- 3:20 M. Hodgson
Design and testing of an acoustical student simulator.

3:20- 3:40 M. Hodgson
Preliminary results of the UBC-classroom acoustical survey.

3:40- 4:00 M. Barman, A. Gambino, R. Ramakrishnan and J.C. Swallow
Reverberation time?

4:00- 4:20 J.C. Swallow
Case study: Design of room for competing acoustic objectives.

4:20- 4:40 R. Stevens, and R. Ramakrishnan
High temperature effects on the insertion loss of absorptive duct silencers.

*ROOM - WINDSOR*

## MUSICAL ACOUSTICS

### Chair: Lola Cuddy

1:20- 1:40 W.F. Thompson, F.A. Russo, and A. McKinnell
Musical influences on the perception of time.

1:40- 2:00 W.R. Steinke, L.L. Cuddy, and R.R. Holden
Perception of musical tonality as assessed by the probe-tone method.

2:00- 2:20 M. Njegovan, R. Weisman, S. Ito, and D. Mewhort
How grouping improves the categorisation of frequency in song birds and humans and why song birds do it better.

2:20- 2:40 J. Simpson, and D. Huron
The perception of rhythmic similarity.

2:40- 3:00 *Coffee*

*ROOM - WINDSOR*

## MACHINERY VIBRATION

### Chair: Tom Moore

3:00- 3:20 H.R. Martin
Detection of bearing failure in machines.

3:20- 3:40 J.S.Y. Tjong, D.K. Chang, D. Mathias, S.A.Zena, J.R Runge, and Z. Reif
The development of a cost effective engine dynamic signal monitoring and diagnostic system.

3:40- 4:00 V.M. Huynh, and S. Desai
An optical method for chatter and form error detection in grinding.

4:00- 4:20 R. Gaspar and L. Hawker
Automotive accessory drive system modelling.

4:20- 4:40 E.M. DeSouza, and T.N. Moore
Reduction of hand transmitted vibration in rock drills.

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## RECEPTION & CASH BAR

### Mountbatton Lane

### 5:30 p.m. - 6:30 p.m.

## BANQUET

### Mountbatton A

### 6:30 p.m. - 9:00 p.m.

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## THURSDAY MORNING, OCTOBER 7, 1993

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**8:30-9:15 a.m. PLENARY SESSION Mountbatten A**

**Progress in Measurement Techniques in Acoustics**

**Gunnar Rasmussen**

**Bruel & Kjaer**
**Naerum, Denmark**

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*ROOM - MOUNTBATTEN A*

### ACOUSTICS IN REHABILITATION
### PART I

**Chair: Don Jamieson**

9:40-10:00 D.G. Jamieson
Overview of recent research on hearing aid evaluation at Ontario's Hearing Health Care Research Unit.

10:00-10:20 C.A. Laszlo
Assistive listening devices and acoustical design for hard of hearing people.

10:20-10:40 L.E. Cornelisse and D.G. Jamieson
Advanced clinical audiometry: Measuring residual auditory capacity using a standard audiometer under computer control.

10:40-11:00 *Coffee*

11:00-11:20 M.K. Pichora-Fuller, C. Johnson, B. Schneider, M. Daneman, and M. Stainton
Effect of speech and noise conditions on perception and comprehension.

11:20-11:40 A. De
Auditory distortion measures for coded speech quality evaluation.

11:40-12:00 K. Yu, and D.G. Jamieson
Acquiring new speech sounds as an adult: A summary and some new results.

## *ROOM - BAKER*

## NOISE IN THE WORKPLACE
## PART II

### Chair: Don Allen

9:40-10:00 J. Desormeaux
Noise measurements in truck cabins.

10:00-10:20 M.G. Faulkner, K.R. Fyfe, L. Cremers, and E.H. Bolstad
Acoustic modelling and low frequency control of furnace noise.

10:20-10:40 R. Gaspar, and S. Brackett
Proposed active control of engine induction noise.

10:40-11:00 *Coffee*

11:00-11:20 G.L. Gould, and B. Alavi
Noise control of coal pulverizers: An implementation of user friendly design.

11:20-11:40 M. Hodgson
Sound power/pressure-level environmental-correction factors for typical industrial workrooms.

11:40-12:00 B. Kotek, A. Czyzewski, G. Budzynski, M. Sankiewicz, G. Whitehead, and M. Roland-Mieszkowski
Sound exposure in the entertainment business.

## *ROOM - WINDSOR*

## ACOUSTICAL MEASUREMENT & COMPUTATIONAL & ELECTRO-ACOUSTICS

### Chair: Richard Guy

9:40-10:00 A. Abdou and R.W. Guy
Three dimensional transient sound intensity measurements for comprehensive room-acoustic evaluations.

10:00-10:20 T.R.T. Nightingale
Determining flanking transmission and field sound transmission loss in wood-framed constructions using intensity methods.

10:20-10:40 J.-G. Migneron, R. Lemieux, and W. Wu
Application of sound intensity measurement techniques in architectural acoustics.

10:40-11:00 *Coffee*

11:00-11:20 R. Ramakrishnan
Effect of perforated sheets on rectangular silencer performance.

11:20-11:40 V. Ramesh, R.J. Vermeulen, and M.L. Munjal
Power measurement for an acoustically pulsed jet flow.

11:40-12:00 M. Roland-Mieszkowski, and D. Roland-Mieszkowski
Use of digital function generator software for neXT computer teaching and research.

# THURSDAY AFTERNOON, OCTOBER 7, 1993

## *ROOM - MOUNTBATTEN A*

## ACOUSTICS IN REHABILITATION PART II

### Chair: Don Jamieson

1:20- 1:40 M.F. Cheesman, and D.G. Jamieson
Using the Canadian speech research environment (CSRE) to teach speech perception to undergraduates.

1:40- 2:00 S. Rvachew
Clinical application of computer-driven methods for the assessment and treatment of speech perception disorders.

2:00- 2:20 I. MacKay, and A.W. Kramer
Tests for nasometric evaluation: Follow-up on adult subjects.

2:20- 2:40 R.D. Sainani, and D.G. Jamieson
Application of speech recognition technology for dysarthric speakers.

2:40- 3:00 *Coffee*

## *ROOM - BAKER*

## NOISE BYLAWS

### Chair: Chris Andrew

1:20- 1:40 C. Andrew, A. Lightstone, and J. Feilders
City of Toronto noise by-law: Review and amendment.

1:40- 2:00 G. Cummings
City of Toronto equipment noise suppression programme.

2:00- 2:20 J. Feilders
Air conditioning device noise control: Ontario model by-law and Toronto by-law.

2:20- 2:40 C. Andrew, N. Rockhill, and K. Walker
Handling noise complaints through conflict resolution in Toronto.

2:40- 3:00 *Coffee*

*ROOM - WINDSOR*

## PSYCHOPHYSICS & PHYSIOLOGICAL ACOUSTICS

**Chair: Robert V. Harrison**

1:20- 1:40 R.V. Harrison, S.G. Stanton, D. Ibrahim, S. Takeno, and R.J. Mount
Developmental plasticity of the central auditory pathways: Study of frequency respresentation in the auditory cortex and midbrain after long-term high frequency hearing loss.

1:40- 2:00 R. Hetu, and H.T. Quoc
La relation entre la largeur des filtres auditifs et les seuils d'audibilite.

2:00- 2:20 W. Wong, and K.H. Norwich
Frequency and the sensation of sounds.

2:20 -2:40 B. Kimberley
A comparison of distortion product emissions with pure tone thresholds.

2:40- 3:00 C. Giguère, P.C. Woodland, and A.J. Robinson
Application of an auditory model to the computer simulation of hearing impairment: Preliminary results.

3:00- 3:20 *Coffee*

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## ANNUAL MEETING

## OF THE

## CANADIAN ACOUSTICAL ASSOCIATION

**Mountbatten A**

**3:00 p.m. - 5:00 p.m.**

**All Welcome**

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# FRIDAY MORNING, OCTOBER 8, 1993

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## 8:30-9:15 PLENARY SESSION Baker

**The Past and Future of Massey Hall**

**Brian Arnott**

**Brian Arnott Associates**
**Division of NOVITA Ltd.**

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*ROOM - BAKER*

## HUMAN RESPONSE TO VIBRATION

**Chair: Tony Brammer**

9:40-10:00 J.C. Swallow
Developments in whole-body vibration evaluation leading to the new draft international standard.

10:00-10:20 P.-E. Boileau, S. Rakheja, and J. Feng
Suspension seat adjustment parameters and their influence on measured vibration response: Preliminary results.

10:20-10:40 A.J. Brammer, G. Roddan, J. Village, and J. Morrison
Machine identification of waveform characteristics, with application to seat motion.

10:40-11:00 *Coffee*

11:00-11:20 J. Village, J. Morrison, G. Roddan, and B. Remedios
The effect of shock frequency and amplitude on lumbar and thoracic spinal response.

11:20-11:40 J. Village, J. Morrison, G. Roddan, and B. Remedios
Effect of shock frequency and amplitude on internal pressure.

11:40-12:00 R. Gurram, S. Rakheja, and A.J. Brammer
Response of the hand-arm system to vibration.

*ROOM - WINDSOR*

**ORIGINS OF CAA**

**Chair: John Manuel**

This session will comprise an informal round-table discussion by the first members of CAA.

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**AWARDS LUNCHEON**

**Mountbattton A**

**12:00 - 2:30 p.m.**

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# Real-time method for measurement of noise exposure from sources in close proximity to the ear

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Sound sources close to the ear, such as communication headsets present a special challenge when it comes to the measurement of the sound exposure. This is due to two facts: Firstly, it is very difficult to obtain precision measurements of the sound field in a person's ear; Secondly, noise exposure standards refer to sound measured in a "free" field, i.e., in the place where the ear would normally be found, but with the listener removed.

In order to obtain data for exposure that can be interpreted in terms of conventional standards and codes, we developed a method based on an accurate acousto-mechanical model of the human head (ATF, Acoustic Test Fixture) (Kunov, Giguère and Simpson, 1989; Kunov, 1989). With the help of the ATF, a sound level meter, and an attached filter it is possible to read the equivalent free-field sound levels from a communications headset or any other source in close proximity to the ear, including any environmental noise that finds its way into the ear canal. This can be done for any type of device, including insert headsets.

In particular, there was an interest in obtaining a sample of measurements of the noise exposure of workers who use communication headsets under very different working conditions, and with different headsets, environmental noise, etc.

### Real-time measurement method

The Acoustic Test Fixture used in this study complies with ANSI S3.36-1985, a standard pertaining to manikin acoustic measurements (Kunov and Giguère, 1989). The geometrical dimensions of the ATF are based on those of the KEMAR manikin (Burkhard and Sachs, 1975), approximating the physical head dimensions of the median human adult. Unlike the KEMAR and other commonly available manikins, the ATF includes soft tissues (artificial skin) in and around the ear with acousto-mechanical properties closely resembling those of the human adult. These soft tissues are important in headset/tissue interactions, particularly for insert-type and circum-aural headsets. The ear canal is terminated by a 1/2" precision microphone and Zwislocki coupler, thus accurately simulating the loading effect of the middle ear. The pinna used in the ATF is the KEMAR larger ear with a reduced base. The mass of the ATF head unit is the same as the effective mass of the human head in a sound field. The ATF head unit can be supported in the KEMAR torso by a compliant neck section or in a custom made stand for higher portability. Finally, the ATF head unit has a high degree of acoustic isolation which is important when testing communications devices that also provide some protection against environmental sounds.

Current noise exposure criteria are based on sound levels recorded in the diffuse field. We designed a filter, allowing the exposure data to be available in real-time, as opposed to the original method where 1/3 octave bands of noise levels were transformed to equivalent diffuse free field values using a work table. The filter performs the transformation, and attaches to Bruel & Kjaer Type 2230, 2231, and 2233 Sound Level Meters, forming a compact portable unit. Thus time-averaged sound levels, maximum levels, absolute peaks, and other measurements can be readily obtained in real-time (Kunov, Skobla, and Munshi, 1991).

A schematic diagram of the setup used for the measurement of noise from headsets is shown in Figure 1. The Duplicator Box is an active signal splitter that uses impedance matching circuitry to produce two output signals that are independent and identical in shape and level to the input signal. It was not always possible to use this box, however, but in all such cases, the console where the worker plugged in the headset had a parallel output we could use. As a result, the signals at the headsets were attenuated versions of the original signal. However, the level of the original signal was restored when there was volume control at the signal source. When there was no volume control at the signal source, the attenuation was measured with a test signal and a compensation factor was then added to the measured level (the maximum possible attenuation is 6 dB).

A headset, connected to an output of the signal splitter, is worn by the operator. Another headset of the same type, connected to the second output of the splitter, is mounted on the ATF. Ideally, the two headsets would be perfectly matched in their operating characteristics. However, the headsets need not be closely matched if, the operator can adjust the volume of the sound from his headset and then this headset is mounted on the ATF, while the operator is given another headset to use during the measurement.

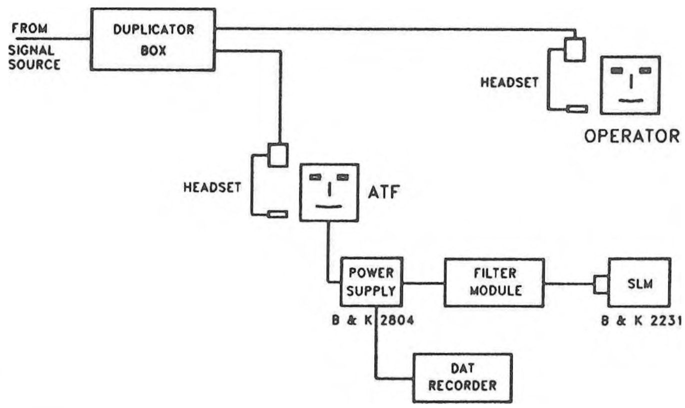


**Figure 1:** Diagram of the setup for measurement of headset noise.

### Validation

A number of aspects of the method used in this study have been validated by Kunov et al. (1989) through probe microphone measurements, repeatability measurements, and loudness balance measurements. The accuracy of the method, with the addition of the filter for real-time measurements, is investigated here.

Four speakers, driven by a pink noise generator and two amplifiers, were used to create a reasonably diffuse sound field in a small region of space of a sound proof booth. The measured overall levels in this region, with a sound level meter pointing in 8 different directions in the horizontal plane and 2 in the vertical plane, were within 1 dB.

The readings obtained by the sound level meter, with its microphone inside this region and facing one wall, were considered as the "true" measurement of the field. Then the ATF manikin head was placed such that the entrance of its ear canal was in the same direction and approximately the same location as the microphone in the "true" measurement. The filter was connected and the system calibrated as in a field measurement.

With a "true" level of 85.0 dB(A) (80 s Leq) for the broadband noise, the level measured by the ATF plus Filter was 85.8 dB(A). Levels obtained by the sound level meter alone and by the ATF plus Filter system, with third octave bands of pink noise are shown in Figure 2.

### Comparison with other methods

Earlier studies employed either a miniature microphone placed in or at the entrance of the ear canal, or a probe tube inserted in the canal and coupled to an external microphone (Kunov et al., 1989). Other investigators have used the KEMAR manikin to study the noise exposure from "Walkman" headsets (Rice et al., 1987 and Skrainar et al., 1987).

As part of this study, we evaluated equipment developed by Barron & Associates and used by Forshaw et al. (1982). The equipment consisted of a Knowles miniature microphone placed at the entrance of the ear canal and an electrical filter that restores the signal to the equivalent external diffuse field. To test this method, the ATF manikin head was placed in exactly the same broadband noise field described above. The miniature microphone was placed in the ATF pinna, at the entrance of the ear canal.

With a "true" level of 85.0 dB(A) (i.e. the level obtained with the sound level meter alone), the level measured by the earlier method was 84.2 dB(A). Thus, with the particular acoustic field used in validation tests, both the earlier method and the method used in this study proved to be very accurate when overall levels were measured. However, the performance of the Miniature Microphone plus Electrical Filter was poor at higher frequencies (above around 3000 Hz) and very low frequencies (below around 100 Hz). The ATF, with its high precision microphone, performed better across the frequency spectrum. Good frequency response would be especially important in some environments, and when there is concern about loud impulsive sounds.

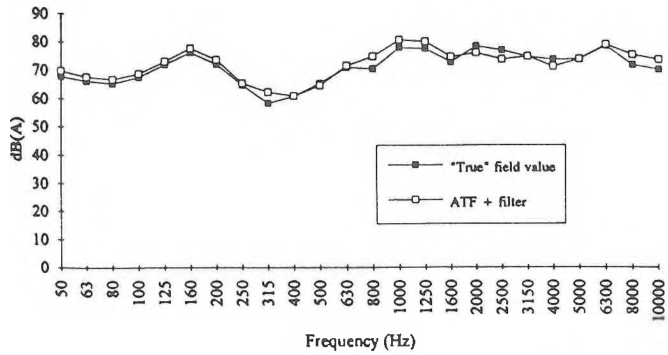


**Figure 2:** Comparison between free quasi-diffuse field and inverse filtered ATF signal in the same acoustic field. Third-octave bands.

Noise levels from insert-type headsets cannot be measured with a miniature microphone at the entrance of the ear canal. This restriction also applies to the widely used supra-aural headsets which have ear-pieces that press against the opening of the ear canal. With headsets that resemble earmuffs, the cable of the miniature microphone can affect the seal against circum-aural skin. This is important in high noise environments.

Placing the miniature microphone requires taping the ribbon portion of the microphone cable to the wearer's cheek and neck with surgical tape. Although Forshaw et al. (1982) affirm that this allows unrestrained head movement, their measurements were not conducted outdoors and not with workers who continually moved around or operated vehicles. Moreover, sometimes it is not possible to interfere with workers by attaching microphones to them (we faced such a situation at the control tower of a busy airport). In contrast, after the several minutes that are required to set up the ATF and its attached equipment, the headset user can continue with his/her work without interference. Measurements inside moving vehicles and other mobile situations can be readily taken with the ATF, as we have shown in this study.

One drawback of the method using the ATF manikin head is that it is more expensive and more complex (in terms of equipment) than the method using the miniature microphone. Also, although this is not usually a drawback, the ATF method estimates the noise exposure for a "median" human head and not of a particular individual.

### Measurements

With this method, we performed detailed measurements of the noise exposure of workers who use headsets at eight different sites. The workers included air traffic controllers, telephone operators, telephone cable maintenance workers, and ground crew at two airports. They used different types of communication headsets (intra-, supra-, and circum-aural) of different makes.

Based on the measurements and information about the work schedules, we estimated the equivalent 8-hour noise exposure for the worker. All measurements were A-weighted and transformed to the diffuse field.

Workers in quiet office settings (telephone operators, air traffic controllers) with environmental noise < 60 dB(A) experienced noise exposure with a range of 64 - 81 dB(A) and a median of 68.9 dB(A). Both supra-aural and intra-aural headsets were used in this environment, with the latter producing the two highest readings. Workers in moderately noisy environments (environmental noise in the range 60 - 80 dB(A)) used supra-aural headsets and had exposure in the range of 70 - 84 dB(A) and a median of 74.2 dB(A). Workers in noisy workplaces (environmental noise in excess of 80 dB(A)) used circum-aural headsets which act as hearing protectors as well. Their noise exposure had a range of 76 - 95 dB(A) and a median of 81.6 dB(A). High environmental noise contributes to the exposure both directly and indirectly by causing the worker to raise the volume of the audio in the headset.

The range of noise exposures overall was 64 - 95 dB(A). The upper end of the range was found in connection with a hearing protector modified as a headset by non-experts. Disregarding this anomalous case, the highest noise exposure was 88 dB(A).

The maximum RMS levels were 85 - 98 dB(A) for "office" settings, 72 - 120 dB(A) for "street" settings, and 88 - 107 dB(A) for "airport" settings. An issue of current interest is the levels of impulsive noise in industrial settings. The measured maximum peak levels ranged between 87 and 129 dB(A). Although these readings are above 120 dB, they are lower than 140 dB, a critical level in some jurisdictions.

### Conclusions

Communication headsets, personal stereo devices ("Walkman"), flight helmets, and other gear attached to, or very near to the ear and generating sound, or shielding the ear in some way from sound, render conventional noise measurements meaningless.

The measurement method presented in this report compares favourably with other methods, both in accuracy and in efficiency. This is particularly true with broad-band noise signals. Another advantage is that the method can be used with any type of headset. A disadvantage is inconvenience of extra equipment needed (filter and head simulator).

The entire setup is battery operated, and is therefore completely mobile. Because of the number of pieces of equipment, it is best if there is an assistant available, but it is not absolutely necessary.

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### Acknowledgment


This work was carried out under contract with Labour Canada.

# Indices of Communication Handicap in Mildly Hearing-Impaired Listeners

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## 1.0 Introduction

The aim of the present experiment was to measure auditory performance decrements among normal-hearing and mildly hearing-impaired Canadian Forces personnel with communications experience. In previous research with inexperienced subjects, we demonstrated that aging, without concomitant hearing loss, resulted in decreased acuity for changes in both stimulus frequency and duration. Compared with aged-matched controls, older individuals with bilateral high-frequency hearing loss showed decrements in frequency discrimination at 4000 Hz, and in consonant discrimination and word recognition in noise. The degree of high-tone hearing loss was correlated with the decrement in speech perception [1].

The measurements chosen for the present study were detection and masked detection thresholds for 2000 Hz and 4000 Hz pure tones, frequency selectivity in the region of 2000 Hz, consonant discrimination of CVCs in quiet and speech spectrum noise (S/N = -4 and +8 dB) and the recognition of the final word in sentences presented in multi-talker babble noise (S/N = 0, +5 dB). Subjects also completed two questionnaires to document their occupational noise exposure history and perceived difficulty with speech perception. The results of the questionnaire surveys are presented elsewhere [2].

## 2.0 Methods and Materials

### 2.1 Subjects

Two groups of subjects, aged 24-52 years, were tested. The first group comprised 15 subjects with screened normal hearing, i.e., audiometric thresholds less than 10 dB HL on average from 500 Hz to 4000 Hz. The second group comprised nine subjects with average thresholds less than 10 dB HL from 500 to 2000 Hz, and about 30 dB HL at 2000 Hz and 4000 Hz.

### 2.2 Apparatus

Subjects were tested individually in a sound proof booth. For details see [2]. For all tests, the stimuli were presented binaurally over a Telephonics TDH-39 matched headset. Levels were calibrated by means of a Bruel & Kjaer artificial ear (Type 4153). Subjects responded using a computer terminal keyboard.

### 2.3 Procedure

For each of detection, detection in noise and frequency selectivity, a four-interval forced-choice signal detection paradigm was used [3]. On each trial, the subject was presented a ½s warning light, followed by a sequence of four listening intervals of 300 ms, cued by flashing lights on the subject's terminal. In the detection tasks, the 300 ms-pure tone to be detected occurred during one of the intervals randomly determined from trial to trial, while the remaining intervals were silent. For detection in noise, pre-recorded 90 dB SPL-helicopter noise was present continuously throughout the trial. The subject chose the "correct" interval. The intensity of the stimulus was varied across blocks, so as to generate a psychometric function from which the detection threshold, the value of intensity yielding P(C) of 0.625, was interpolated.

In frequency selectivity, a narrowband masker was gated on in each of the four intervals and the 2000 Hz pure tone probe was presented simultaneously with the masker in one of intervals, randomly determined from trial to trial. The duration of the probe and masker were 300 ms. The level of the probe was fixed at 10 dB SL. Across blocks of 24 trials, the level of the masker was varied, so as to generate a psychometric function from which the critical masker level, that value of the masker intensity yielding P(C) of 0.625, was interpolated. The critical masker value was determined for maskers centred at 1250, 1600, 2500 and 3150 Hz.

Consonant discrimination in quiet and in speech spectrum noise and speech perception in noise were assessed by means of the California Consonant Test [4], the Four Alternative Auditory Feature Test [5], and the Speech Perception in Noise Test [6], respectively.

## 3.0 Results

### 3.1 Detection and Frequency Selectivity

The mean detection thresholds for the groups were not different at 2000 Hz, but were significantly greater for the hearing-impaired at 4000 Hz. Except for the hearing-impaired at 4000 Hz, the masked threshold was significantly greater than the threshold in quiet. For frequency selectivity, although the probe was on average 6 dB higher in the hearing-impaired, the difference was not significant. The critical masker levels were virtually the same for the two groups.

### 3.2 Speech Intelligibility

Consonant discrimination, using CVCs in quiet, was over 90% in both groups, regardless of the method of scoring (overall percent correct or the percent correct for items contrasting the initial or the final consonant). For CVCs in noise, a decrease in the S/N resulted in a significant decrement in performance of 11% to 20% depending on the group and method of scoring the data. Mild hearing loss resulted in significantly lower scores of about 8%. With regard to the word recognition, performance improved in both groups as the S/N increased. For sentences with high and low contextual cues, the differences due to S/N were 9% and 25%, respectively, for the normal-hearing group, and 20% and 36% for the hearing-impaired group. With S/N = 0, the results were significantly worse in the impaired group by 12% and 19% for the sentences with high and low contextual cues, respectively.

To assess the relationship between hearing and speech perception, correlation coefficients were computed within group between the detection thresholds for 2000 Hz and 4000 Hz pure tones and the audiometric thresholds for 250, 500, 1000, 2000, 4000 and 8000 Hz and each of the speech perception measures. In the normal group, the audiometric threshold at 2000 Hz was significantly correlated with consonant discrimination in quiet. For the hearing-impaired group, the detection threshold at 2000 Hz in quiet and the audiometric threshold at 2000 Hz were significantly correlated with consonant discrimination in quiet and speech perception for sentences with low contextual cues and S/N of 0 dB. The audiometric threshold at 8000 Hz was significantly correlated with consonant discrimination in quiet and in noise with low S/N and speech perception with low contextual cues in noise and low S/N.

### 4.0 Discussion

In line with previous outcomes, in the hearing-impaired, low frequency masking noise had no effect on detection at the frequency of hearing loss, i.e., 4000 Hz, likely because its perceptual effect was diminished. Mild hearing loss did not affect consonant discrimination in quiet, although it did result in a greater decrement in noise than normal. This same result was evident for word recognition in noise, especially with poor contextual cues. The audiometric and detection thresholds at 2000 Hz were highly correlated with measures of speech perception. Although the audiometric threshold ranged only between -5 and 30 dB HL, word recognition with poor contextual cues and low S/N ranged from 36% to 8% - - an important finding because it shows the detrimental effect of a borderline hearing loss on intelligibility under adverse listening conditions. The predictive value of the 8000 Hz threshold supports further study of loss this region as an index of handicap.

The relationship between hearing and speech perception is not well-understood. There are reports of wide intersubject variability in the latter given similar hearing loss [7]. The trends in the present study are in line with findings that the main determinant of speech recognition in the hearing-impaired is the average threshold at 1000, 2000 and 4000 Hz [8]. Noticeable handicap has been reported for those with a PTA of 30 dB HL [9].

**Acknowledgements**

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# The hearing aid as warning signal receiver in noisy workplaces

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Since assessment of employability should take into account to what extent a hearing aid may restore hearing capabilities [1], this study was undertaken in order to test the possibility of the hearing aid induction coil to act as an effective receiver for sound warning signals in noisy surroundings.

## Method

Experiments were carried out in a hemi-anechoic chamber using the acoustic head simulator designed by Kunov and Giguère [2]. This acoustic test fixture (ATF) approximates the physical dimensions and the acoustical eardrum impedance of the median human adult. The ATF includes a mechanical reproduction of the human circumaural and intraaural tissues. The acoustic isolation of the head simulator is greater than the bone conduction limitations to hearing protection.

The hearing aids were tested on the ATF, the left ear of which was equipped with the large KEMAR pinna. The sound pressure level at the output of the aid was picked up in a Zwislocki coupler fitted with of a condenser microphone (BK-4134) connected with a real time analyzer (BK-2123) by means of a preamplifier (BK-AO009).

Magnetic signal reception was tested using the Comtex and the Phonic Ear (System 4, model PE 475) FM transmitters, electrical signals serving as input. The induction coil of the Phonak Pico behind-the-ear (BTE) aid was used as a receiver. The response curve of the transmitters were assessed using a magnetic loop, a silhouette and a direct coupling. The Widex Q16 multi-programmable BTE aid was also used as a receiver to test the effect of the amplification curve setting on magnetic signal gain.

The influcence of background noises on masked thresholds was assessed for (a) the median hearing sensitivity of normal hearing males aged 55 years (ISO-7029), (b) a sloping high frequency hearing loss, ranging from 30 dB HTL at 2 kHz to 70 dB at 6 kHz, (c) a sloping loss ranging from 25 dB at 1 kHz down 75 dB at 6 kHz. Calculated attenuation values from a lucite earmold [3] were entered in *Detectsound*, a signal detection model [4] and masked thresholds were computed for a 85 dBA pink noise.

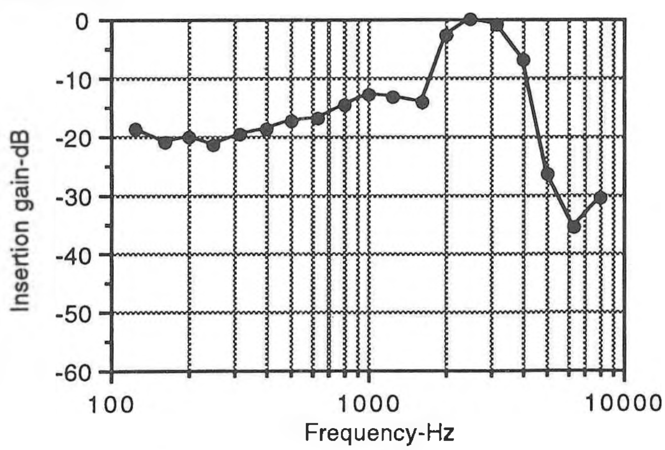


Figure 3. Insertion gain of Phonac PICO BTE aid measured on the acoustic test fixture.

## Results

As a reference point, Figure 1 presents the insertion gain of the Phonak Pico BTE aid set for a high frequency hearing loss. Figure 2A depicts curves of the two FM systems tested. It can be seen that signal tranmission is not perfect. The Comtex system, in particular, serioulsy limits the passband of signal transmission. Figure 2B shows the corresponding frequency response when magnetically coupled with the Phonak Pico BTE aid. In the acoustic mode, the aid leads to a difference of 20 dB between the frequency of maximum gain (2.5 kHz: see Fig. 1) and the lower frequencies (e.g. at 200 Hz); in the magnetic mode, the difference amounts to 50 dB. This imperfection is partly avoided when the signals are received by direct audio input as shown in Figure 3. The response curve is closer to that from the normal acoustic mode, although there is still a systematic gain difference of 10 dB in the lower frequencies.

Such an effect can however be compensated by means of proper amplification settings. This is illustrated in Figure 6 with a multi-programmable aid. It can be seen that with a maximum bandwidth amplification, the passband is sufficient to cover a wide range of frequencies. This allows to transmit adequately sound warning signals in the range of 250 and 4000 Hz.

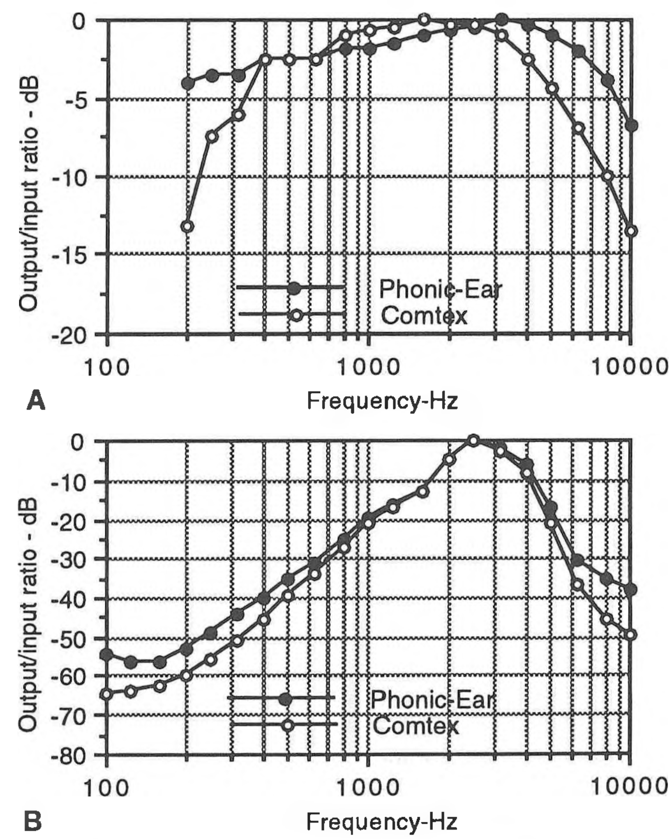


Figure 2. Frequency response of two FM transmission systems, measured (A) at the output of the FM receiver and (B) at the output of the Phonak Pico BTE aid operating in the magnetic mode.

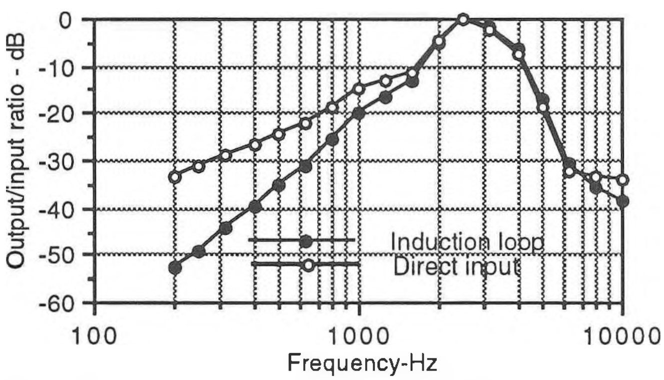


Figure 3. Frequency response of the Phonik Ear FM transmission system coupled with the Phonak Pico BTE aid through an induction loop and direct input.

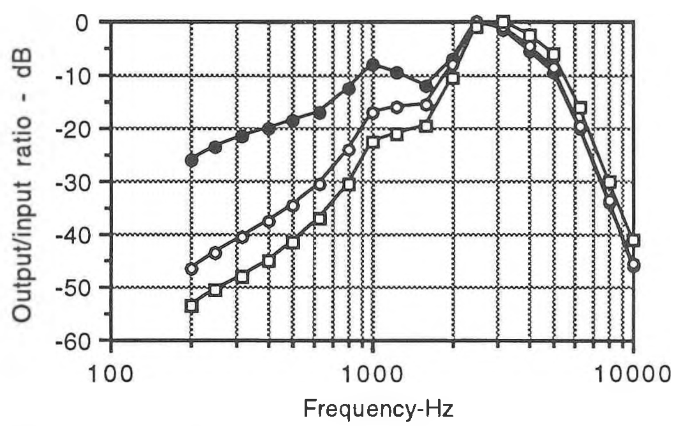


Figure 4. Frequency response curve of the Phonic Ear FM transmitter for magnetic signals measured at the output of a Widex Quattro BTE aid using direct input coupling; the three curves depict the response for different amplification settings.

A simulation of signal detection in noise was perfomed using our computerized model. It was assumed that a shell unvented earmold acted as an attenuator [3]. Masked thresholds were estimated for three audiometric configurations and a noise with a flat spectrum presented in a free-field at 85 dBA (Table I). It must be recalled that, in order to be detected and recognized, signals must be presented at 10 to 15 dB above the masked thresholds. The computer model used includes a +12 dB margin above the masked threshold [4]. Signal levels were computed so as to estimate the levels required at the output of the hearing aid. As shown in Table I, the resulting levels of recognition thresholds are all below 75 dB SPL. With high frequency hearing losses, the limiting condition is actually the absolute threshold at 3.15 kHz.

It is noticeable from Table I that the values of the signal recognition thresholds do not parallel the audiometric thresholds. This is mainly due to the poorer background noise attenuation in the lower frequencies provided by the earmold. As a consequence, the hearing aid setting for such a situation will strongly differ from that required for the normal acoustic mode. This calls for the use of a multi-programmable aid.

Table I. Audiometric thresholds and signal recognition thresholds (masked thresholds + 12 dB) in a background pink noise at 85 dBA, when attenutted by a lucite shell earmold of a BTE aid. The audiometric thresholds refer to normal male listerners aged 55 years (N) and to listeners with two degrees of sloping losses, S1 and S2. Signal recognition thresholds are computed at the output of the hearing aid. Bold characters refer to masked thresholds equal to absolute threshold.

| Frequency Hz | Hearing threshold levels - dB ISO | | | Recognition thresholds - dB SPL | | |
|---|---|---|---|---|---|---|
| | N | S1 | S2 | N | S1 | S2 |
| 250 | 5 | 5 | 5 | 72 | 72 | 73 |
| 500 | 5 | 5 | 5 | 61 | 61 | 62 |
| 1000 | 6 | 6 | 25 | 59 | 60 | 61 |
| 2000 | 8 | 30 | 35 | 53 | 54 | 57 |
| 3150 | 16 | 50 | 55 | 47 | **68** | **73** |
| 4000 | 22 | 65 | 70 | | | |

### Discussion

According to the present results, the attenuation provided by unvented earmolds and the amplification of magnetic or direct input signals by hearing aids can be combined to maximize the signal-to-noise ratio of sound warning signals transmitted to hearing impaired workers in noisy surroundings.This can help removing obstacles to the integration of people with hearing impairments in the workplace [5]. It can also represents a first step in occupational rehabilitation of hearing impaired industrial workers, that is a means to adapt to reduced auditory capacities in the workplace. Finally, the hearing aid could be used, as illustrated above, with normal listeners who need to receive sound signals in highly unfavorable acoustic conditions. This type of solution is potentially more effective than conventional earmuffs equipped with earphones [6]. The background noise attenuation would probably be higher and the response curve of the receiver would be finely tuned to both the sensitivity of the listener and the masking power of the attenuated noise.

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# Sound Propagation of reverse alarms used on heavy vehicles

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## INTRODUCTION

Each year, serious accidents occur in noisy workplaces because a warning sound is not heard.[1] In the last 15 years, a minimum of 22 deadly accidents occured in Quebec on construction sites because workers did not hear the reverse alarm or the noise emitted by a heavy vehicule. In Canada and United States, such accidents have occured and involved trucks and buses.[2-4]

In Quebec, the regulation related to the use of reverse alarms on heavy vehicules referred to the SAE J994 Standard.[5] Regarding acoustical characteristics, this standard states that the predominant sound frequency of the alarm should fall within the frequency range of 700-2800 Hz. The cycles of sound pulsations from the alarm shall be of the order of 1-2/sec. The duration of the "on" and "off" intervals shall be approximately equal in length. The alarm should be tested in free field, 4 feet above a horizontal reflecting plane or laboratory equivalent, with the microphone 4 feet from the alarm's horn along its 0 degree axis. The alarm is then classified according to its sound pressure level: Type A: 112 dBA, Type B: 107 dBA, Type C: 97 dBA, Type D: 87 dBA and Type E: 77 dBA. The accidents reported above raise the question as to what extent such a standard insures that every worker will perceive the alarm at any position behind the heavy vehicule.

## METHOD

In order to answer this question, measurements were made in a typical situation under which reverse alarms are used. Five reverse alarms of two kinds (pulsed alarms and modulated alarms) were tested at the rear of four heavy vehicles (ten wheel truck, grader, cement mixer, loader). Two positions of the alarms were tested: at the rear and under the vehicle. The ground was similar to what is found on construction sites: earth and gravel. There were no obstacles near the measurement sites.

Figure 1 illustrates the positions where measurements were made. They were made at 1 meter behind the vehicles, along the axis 1 to 9 and on axis 5 at 1, 2 and 4 meters . The sound emitted by the alarms was collected with a type 1 microphone (Cirrus MK-224) plugged into a numerical audio-tape recorder (TEAC, DA-P20). The recordings were transferred to a 1/3 octave band analyser (dBFETE) in the lab.

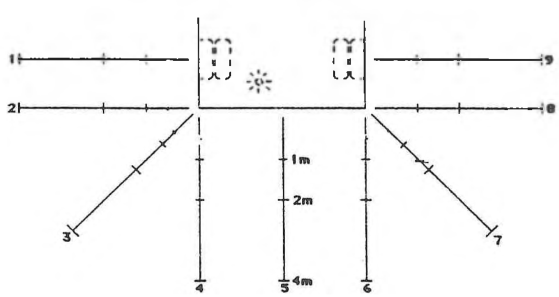


Fig.1 Axis position behind the heavy vehicles.

## RESULTS

The frequency content of the alarms varied from 1140 to 3025 Hz. It is characterized by a pure tone in the case of the pulsed alarms and by two pure tones for the modulated alarms. These alarms were typical of what is found on the market.

The most interesting results were obtained at 1 meter behind the vehicles. Figure 2 presents the results of a serie of measurements for a modulated alarm (Warrick #12). From this figure, it is clear that strong attenuations, amounting to 20 dB, occur along this axis. The same kind of attenuations were noted on the other vehicles. When the alarms were put under the vehicles, the attenuations were more important.

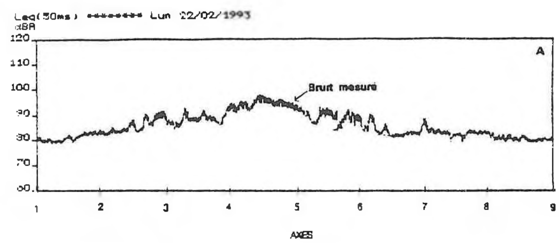


Fig.2 Sound pressure levels of the Warrick #12 modulated alarm measured at 1 meter at the rear of the grader , along axis 1 to 9.

Measurements were also done along the number 5 axis. The same kind of attenuations were noted. Near the vehicle, direct and reflected sound waves are highly diffracted on the sides of the vehicle before reaching the worker ears. At around 1 meter, levels are higher because there is less diffraction. Farther from the vehicle, there are attenuations due most probably to the cancellation of the direct and reflected waves.

A computerized model (ALARM) based on the state of the art knowledge in outdoor sound propagation (6,7) has been developped to simulate the cancellation phenomena. Figure 3 shows that the model accurately simulates, in terms of shape, the results obtained on the sites.

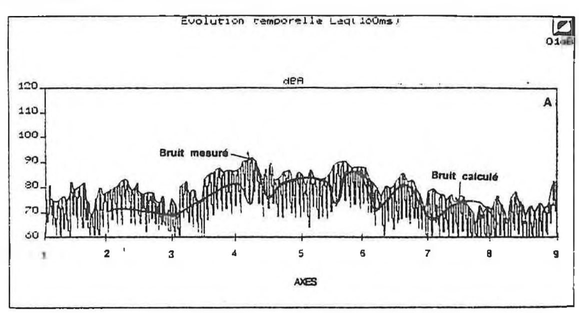


Fig.3 Comparisons between the measured and the predicted sound pressure levels based on ALARM software.

## DISCUSSION

These results show an important flaw in the SAE J994 standard[5]: even if the studied alarms had met the requirements of the standard 4 feet behind the truck on the 0 degree axis, it does not mean that the sound pressure levels is uniform everywhere behind this same truck. Strong attenuations occur due to the use of pure tones in the majority of reverse alarms available on the market.

Based on these results, the detection of reverse alarm is largely dependent on the position of the receiver behind the truck. At some positions, the alarm is clearly audible whereas at others, it is inaudible due to the cancellation of waves and to the background noise which can be high enough to mask the alarm.[8]

Faced with these results, it is important to propose modifications in the design and the positioning of reverse alarms on vehicules. Any change must also take into account other factors which can explain the fact that reverse alarms are not heard. For example, high noisy backgrounds and noise-induced hearing loss can also interfere with the perception of reverse alarms.[8] Habituation to the pulsed tones is also often mentionned by workers.[9] There are so many alarms sounding at the same time in some worksites that workers become insensitive to the different signals. In that same conditions, localisation of the sound source is also a problem.[9]

At this stage, the following recommandations can be made. First of all, reverse alarms should not be pure-tones because they are strongly subject to diffractions and reflections. Even tones modulated between near frequencies are not adequate. The alarm should contain many frequency components in the 500-2000 Hz range. Extending this range above 2000 Hz is useless for those affected by noise-induced hearing loss which can affect hearing above this limit.[10]

Secondly, the position of the alarm on the vehicule should be studied in order to limit the diffraction phenomenon as much as possible. This may raise difficulties if one considers the limited space available as it is the case, for example, on a lorry-truck. It is almost impossible to install the alarm on the lorry door because of the danger of damaging it when the door is rocked and because of the possibility that the door be removed to carry instruments or merchandise.

Thirdly, studies on the habituation phenomenon should be performed in order to find a way to make the alarm audible without creating habituation. Research in the field of obstacle detectors may help solving this problem.[2-4,11] These detectors can be used to sound the alarm only when there are obstacles (humans, other vehicules or static obstacles), thus reducing considerably the number of times workers are exposed to such sounds.

Fourthly, the use of an electronic mirror, allowing to see in the blind spot, could help the driver. This technology is not yet fully developped. Studies should continue in that field with consideration for ergonomic constraints in terms of the driver's workload.

Fifthly, the trucks owners should be informed about the problems associated with back-up alarms and should ask for better reverse alarms to the manufacturers. Very often, trucks owners look for the cheapiest alarm (often the less intense) in order to save few dollars. Prevention of deadly accidents largely justifyies expense for better alarms.

Finally, one must insist on the need to reduce the noise levels on construction sites or anywhere where reverse alarms are used. This action would limit the masking effect of the background noise and would reduce the risk of acquiring noise-induced hearing loss which is highly prevalent among çconstruction workers.[12]

## CONCLUSION

The use of reverse alarms was introduced in order to solve the blind-spot problem and to reassure the vehicule driver while backing-up. At this time, little concern has been expressed about the conditions that need to be met in order to ascertain that reverse alarms are perceived. In most cases, signalmen help the driver to reverse but in many cases, the signalman is the one who is fatally hit. These facts reinforce the need to improve the signals and to pursue studies on safety related to reverse alarms. Studies on optimal acoustical characteristics and position should be continued in conjonction with studies on obstacle detection. This technology could help to overcome the habituation phenomenon and localisation problems reported by many workers which are probable responsible for some of the deadly accidents that have occured.

Acknowledgements

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# PROPOSED STATE-OF-THE-ART CRITERIA FOR OFFICE BUILDING ACOUSTICS


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## Background

Public Works Canada (PWC) is the federal Department responsible for the provision, management, and operation of Canadian Government office space. PWC owns around 400 buildings, and manages leases for about 4000 others. In recent years, PWC has established a Productive Work Environment Committee, to oversee research and development projects dedicated to the improvement of interior environmental conditions in PWC buildings. One of the initiatives of the Committee has been the development of performance based standards for building acoustics, complemented by prescriptive guidance where appropriate, for use by the Department and its suppliers. This paper reports on the proposed criteria which have been collated and/or developed, including targets for: background noise (upper *and* lower limits); reverberation times; attenuation with distance; speech transmission coefficients; and office isolation. These criteria are presented for open and closed office space, and meeting rooms. Legal requirements are reviewed, and the unique requirements of hearing impaired occupants are also addressed. It is most important that PWC's criteria are realistic, and can be achieved within our existing budgets, expertise, and equipment resources. The implications of these criteria in way of office fit-up are discussed, and in particular the impact that these criteria can have on other interior environmental performance indicators including indoor air quality (IAQ), thermal comfort, and illumination. Specific recommendations are given for ceiling systems, screens and partitions, and office equipment noise levels.

For further information on PWC's Acoustic Strategy, see reference 1.

## Legal Requirements

Mandatory requirements for acoustics (noise levels) in federal buildings are defined by the Canada Labour Code and its associated Canadian Occupational Safety and Health Regulations (see reference 2). These were most recently updated in July 91 and include requirements for: investigation of workplace noise levels of 84 dB(A) or over by a qualified individual; noise exposure levels not to exceed prescribed limits; signage to identify high noise areas; training in the correct use of hearing protection devices; and auditory testing of workers who may be exposed to high noise levels with permanent retention of the results on the employee's medical files. Since the primary focus of this paper is on noise levels far below the limits of relevance to this Regulation, no further discussion of these issues is presented herein.

## Performance Criteria

**Background noise** levels for office accommodation have historically been defined by the building industry in terms of maximums not to be exceeded (see reference 3). These limits are appropriate for HVAC system design, but are not able to fully define the background noise levels and spectrum appropriate for office buildings. With the trend towards more open office workstations (PWC's target for the ratio of open to closed offices is 70/30), there is increased attention being given to the subject of speech privacy. This issue is compounded by the additional tendency to smaller workstations: new fit-up typically provides about ten square metres per occupant. PWC is therefore moving towards a background noise *target* (as opposed to only an upper limit), which is based on the spectrum proposed by Dr. L.L. Baranek in reference 4. This type of spectrum is comparable to that proposed by suppliers of sound masking, which is now being included in many of PWC's projects, at the request of our clients. For further information on sound masking see reference 5. With background noise within the target spectrum discussed above, it becomes easy to set targets for the overall level in dB(A): 45 dB(A) in open offices, 40 dB(A) in closed offices; and 35 dB(A) in meeting rooms, executive offices etc. Tolerances of plus minus one dB(A) are typically achievable in buildings with sound masking.

Persons with **hearing impairments** require lower background noise levels for the correct functioning of hearing aids, and thus are better suited to closed offices, or open office areas with lower background noise levels than those listed above. For more information on building acoustics for the hearing impaired, see reference 6.

**Reverberation times** are typically not a problem for PWC in way of standard office fit-up, however, there have been instances where higher ceiling heights and reduced amounts of sound absorbing surfaces (such as with concrete coffered ceilings), where sound absorption in the open office has been less than satisfactory. PWC therefore specifies that reverberation times are to be less than 0.6 seconds.

**Attenuation with distance** is of course directly linked to reverberation times, but may in some circumstances be more easy to measure, depending on the equipment resources available. PWC aims to have at least a 4 dB drop per doubling of distance in open office fit-up.

The determination of **speech transmission coefficients** is yet another way that open office acoustic performance can be quantified. PWC aims to provide speech privacy (RASTI scores below 0.3, see reference 7) at distances beyond one workstation removed.

**Closed office noise isolation** is also becoming an increasingly important issue for our clients, with the trend to smaller offices and lighter weight construction techniques. The basic drywall partition is rated with a Sound Transmission Class (STC) of 35, and thus a Noise Isolation Class (NIC) of at least 25 should be achievable with due care and attention to design and construction details. PWC is moving towards specifying closed office isolation in terms of NIC levels to be achieved: NIC 25 for basic office construction; NIC 35 for enhanced; and NIC 45 for executive offices and other sensitive areas. When levels higher than this are required these are typically dealt with on a case by case basis. For more information on the provision of closed office noise isolation, see reference 8. More detailed definitions of acoustical terms, including STC and NIC, can be found in reference 9.

Good acoustics for **meeting rooms** are particularly important to PWC, in that there is not only the requirement for enhanced acoustic security, but also for the interior finishes to be appropriately selected to provide the acoustics necessary for good speech communication. PWC currently has a field trial underway to develop a concise guideline on how to evaluate existing meeting rooms and determine what improvements can be economically made to improve upon their acoustical performance.

The National Building Code of Canada requires that "in buildings of assembly occupancy, all classrooms, auditoria, meeting rooms and theatres with an area of more than 100 $m^2$ shall be equipped with an assistive listening system encompassing the entire seating area" (see reference 10).

**Prescriptive Criteria**

There is always a danger in specifying prescriptive criteria that a supplier will follow them and fail to meet the required performance criteria. Perhaps the best example of this pitfall is the specification of room isolation in terms of STC instead of NIC. However, it has been found that the provision to suppliers of some indications of fit-up component acoustical performance can be beneficial to guide them in their activities.

PWC uses **carpet** in all office space to dampen footfall and other floor related noises.

**Acoustic screens** typically provide high levels of noise attenuation: PWC uses a minimum of STC 25 and a Noise Reduction (NR) coefficient of 0.6, as rules of thumb. Most screen suppliers provide products well in excess of these figures. Screens should not be higher than 1.6 metres, because of aesthetic issues to do with lines-of-sight in the workplace. Additionally, field measurements made by PWC have shown that when air velocities in the workplace are at the lower limits of ASHRAE recommended values (see reference 11) that there is benefit to air movement, and thus Indoor Air Quality (IAQ) and thermal comfort in the workplace, if the screens have an air gap beneath them, as opposed to resting fully on the carpet.

**Acoustical ceiling tiles** are an important component of office fit-up, in that in combination with the screens they are the major absorbers of noise. For closed offices in particular, if the partition only extends to the false ceiling, the ceiling is invariably the weak acoustic link. Higher STC tiles (over STC 35) are therefore required for closed offices to support the NIC targets given above, although for closed offices the tile NR coefficient is of lesser importance. For open offices the reverse is the case: very low STC tiles can be used with little penalty, although a higher NR coefficient, say 0.8, is most beneficial in terms of the provision of the desired speech privacy between workstations.

The light reflectance characteristics of carpeting, screens and the ceiling system should be carefully considered for their contribution to good illumination in the workplace.

**Noisy office equipment** is perhaps one of the biggest offenders of a good acoustical environment in the open office. Fortunately, with the trend towards laser printers, this problem is becoming less of an issue as time progresses and individual impact type printers are replaced by laser printers shared by a group. Nonetheless, it is important to identify any noisy machines which are required in the open office and group them together in less noise sensitive areas, surrounded by ample screens to absorb their noise as close to the source as possible.

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# The Noise Impact Statement Process in Toronto

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**Noise and the Planning Process**

The major objective of the planning process is to minimize the potential for conflict through the effective planning and design of land uses.

Within the planning process all new development or redevelopment of lands within the City of Toronto is strictly regulated through the provisions of the Planning Act (1983) which prescribes general rules for land use planning within the Province of Ontario.

The City provides comments and recommendations with regard to land use planning on all matters relating to its mandate which is; the protection, conservation and management of the natural environment. Other subjects of concern are Noise and Vibration defined in the Environmental Protection Act as being containments.

**Noise Impact Statements**

The City of Toronto zoning process current policy may request a noise impact analysis of any development in cases requiring an Official Plan Amendment or rezoning. This study is generally referred to as a "Noise Impact Statement" (N.I.S.). It addresses three items as follows:

(a)Impact of the Development on the Neighbourhood -
This is generally a forecast of the noise impact generated by the development, together with an outline of the methods proposed to control those noises from being projected into the adjacent neighbourhood.

(b)Impact of the Neighbourhood on the Development -
An analysis which shows the effects of the existing noise environment on the development and the provisions proposed to control any undesirable effects on the development from a standpoint of noise intrusion. As a further precaution, the developer is asked upon completion of the project, to submit a letter from the acoustical consultant or architect certifying that the project has been constructed in accordance with the N.I.S.

(c)Impact of the Development on Itself -
A detailed proposal for the control of noises generated within the development.

Prior to 1978, all Noise Impact Statement were subject to the same review process. Due to the great volume of development activity under way in the City, ranging from the renovation of a house for professional office use to the building of a huge apartment block, a more streamlined procedure was implemented to reduce the amount of time and data required in dealing with developments having minor noise impacts by dividing the Statements in to three classes:

**Class 1**

Deals with minimum impacts that can be prepared by the proponent without the need for engaging the services of an acoustical consultant.

**Class 2**

Deals with projects sited within a reasonable acoustical environment but where, for example, the impact of noise from an improperly designed air-handling system could generate undesirable noise for the adjacent neighbours. The proponent, with the assistance of his architect, can usually analyze the effects of developments in this class.

**Class 3**

Deals with developments where a problem with the acoustical environment would most certainly occur unless specific steps are taken to counter the noise-related concerns.

The proponent is usually required to engage the services of an acoustical consultant to work with the project architect; the consultant is required to certify that the plans prepared for the project have incorporated the noise attenuation features as required by the Impact Statement. In some instances, the consultant is further required to certify that the noise attenuation features have been properly incorporated in the completed project.

**Procedure**

The City's Planning and Development Department circulates, for comments, an application from a developer for a zoning change, to various other Departments concerned within the zoning process. Subsequently, the applicant is informed as to whether a Noise Impact Statement is required.

Noise Impact Statements are reviewed by the Commissioner of Public Works and the Environment who, upon approval, informs the Commissioner of Planning and Development and the Commissioner of Buildings and Inspections Department as well as the City Solicitor. The approval is conditional upon the applicant agreeing to have the proposed development building plans certified by a qualified acoustical consultant and to construct the building in accordance with those recommendations outlined in the Noise Impact Statement, to the satisfaction to the Commissioner of Public Works and the Environment. This condition and others are secured through a collateral agreement which is registered on title following City Council's approval of the proposed development.

A building permit can be issued at this stage. In order to be consistent with the City's policy of "fast tracking" the issuance of building permits, a computer program has been developed. This program can be accessed by the various Departments involved in the process, thus eliminating any errors or delays that might otherwise be crucial in the process.

**Noise Attenuation in Buildings**

Inherent in the NIS process is the provision for sound transmission between dwellings units, hotels and motels. In the most recent edition of the Ontario Building Code construction of walls requires a Sound Transmission Class rating of 50 and floor a rating of 48. However it is questionable whether these ratings are adequate especially for the floor rating which is not based on impact sound transmission. Proposals for stricter requirements have been submitted however, substantial changes to the Building Code are not expected before the year 2000.

In the case of entertainment establishments and public halls, another potential means of control is though the Metro Licensing Commission. This agency does not presently have jurisdiction to require the setting and meeting of acoustical standards as a condition of issuing a license. In situations where a licenced applicant has had a history of Noise By-law convictions, the Licensing Commission will take this into consideration and may refuse to review a license on that basis. However many of these situations could be prevented by requiring the applicant to provide a suitable building enclosure and by committing to operating the establishment so as to minimize unnecessary noise.

**Further Procedural Developments**

In an effort to improve the pro-active process in dealing with potential noise impacts it has been proposed that not only Official Plan Amendments and Zoning changes, but also projects subject to Development Review be included. This would allow for the review of smaller scale developments not usually considered in terms of noise impact but where, in a number of cases, a problem occurs which must then be dealt with after the fact under the provisions of the Noise By-law. The additional review process is intended to be streamlined so that there is no added burden to the bureaucracy. A simplified application form is being devised so that for the majority of cases, a simple one page (Class 1) application will suffice.

# PRACTICAL ASPECTS OF IMPLEMENTING

# SOUND ISOLATION AND NOISE CONTROL

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A suitable subtitle would be "The Frustrations of an Acoustical Consultant with Contractors". When the acoustical consultant is involved on an architectural project to assist with noise control or sound isolation design, it is because these aspects are critical. Usually a high standard of workmanship is as important or more important than the design itself, in achieving the desired results. It is often very difficult to get the "skilled trades" to properly build even the simplest designs that are specially configured for acoustical purposes. There are a variety of reasons:

- Total lack of understanding (e.g. thermal insulation materials used as sound absorption are often viewed as acting similarly, as "acoustical insulation"; that is, as an adequate sound barrier; small gaps, holes or cracks are often not viewed as important.

- Failure to pay attention to drawings and specifications (trades simply proceed in the normal fashion used on non-critical projects -- after all, they have been doing this for years!)

- Sloppy workmanship (e.g., even in attempting to build what is detailed, there may be inadequate care in fully caulking joints, use of gypsum board sheets with broken corners or edges; poor tolerances in fitting joints).

- Deliberate attempts at short cuts to save time and money.

Often many of important acoustical details will become hidden and inaccessible as construction proceeds. For many contractors "out-of-sight is out-of-mind". If the detail will be invisible or inaccessible, either they assume it is not important or they expect to get away with sloppiness. Therefore, construction errors must be caught at the right time to ensure proper results and resolve problems that might be blamed on poor design.

On projects with special requirements it is important to carefully review shop drawings of critical aspects prior to fabrication. A common occurrence is to submit such shop drawings only days before items are needed on site, putting the consultant and design team under heavy pressure. Even though the contractors may have had many months since contract award, the consultant may be deemed (unfairly) to be holding up the project. The solution, of course, is to take preemptive action and document a submission schedule and warn the owner and contractor when not adhered to. Sometimes shop drawings are submitted for approval after fabrication, or even after delivery to the site. When fabrication errors are then discovered, the construction schedule (which is usually as sacrosanct as the budget) is imperiled. This often leads to compromise solutions to save time.

On-site construction review should be done at strategic times. If too early, the details of concern may not be started; if too late, they may not be visible. Thus, there must be reliance on proper co-ordination and notice by either the contractor, the architect's site person, or the owner's project manager, to allow the consultant to schedule to be on site at the right time.

Example 1:

Gypsum board (GWB) partition between classrooms to extend to fluted deck above. GWB stopped at bottom of flutes and joint carefully caulked. Openings left through wall via flutes.

Solution: Close flutes with surface-applied GWB precut to flute profile.

Example 2:

Back-to-back fan coil units on common (GWB) wall between hotel rooms. Pipe penetrations to be specially sleeved and sealed to maintain acoustical and fire ratings. Contractor simply cut oversized holes and used thermal insulation over pipes. Very little working room between back of units and wall.

Solution: Cut proper pipe holes in GWB; cut in half through pipe holes; pre-caulk all joints; surface mount multiple layers on both sides of wall; seal all edges.

Example 3:

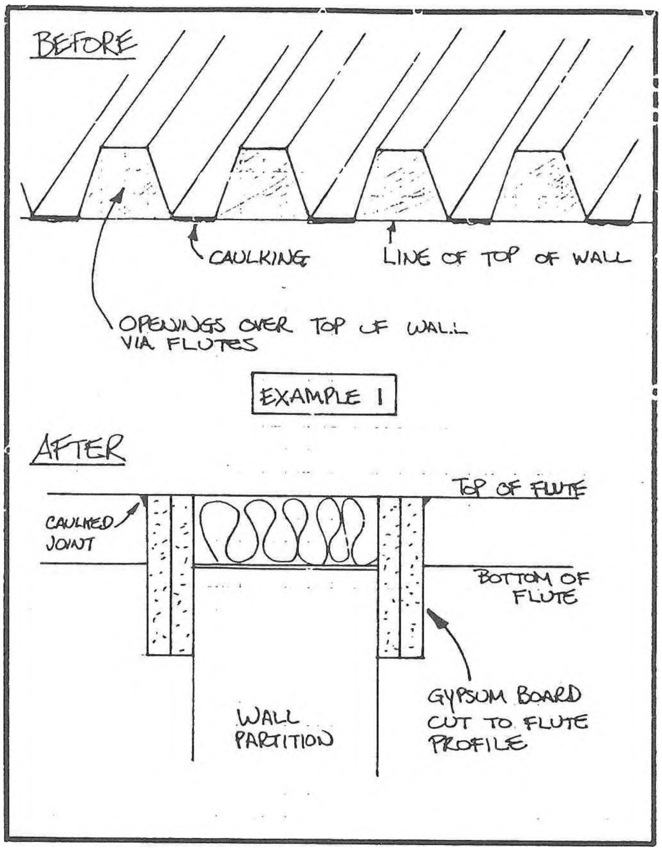
Floating concrete floor for tv studio, on concrete parapets (on isolators) to match existing floor level. Section of isolated parapets poured directly against existing structure, forgetting perimeter isolation board.

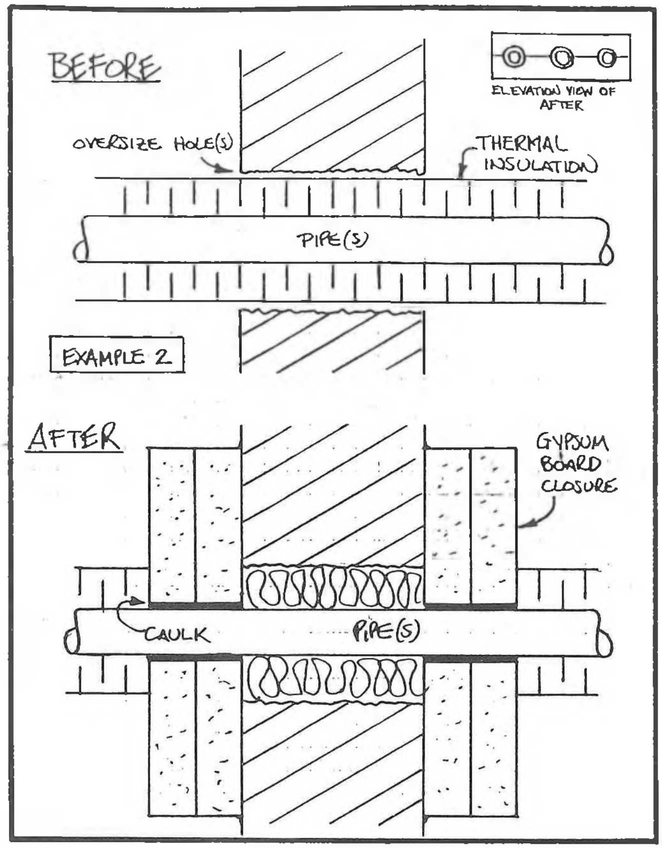
Solution: Saw cut joint; clean out; insert isolation board.

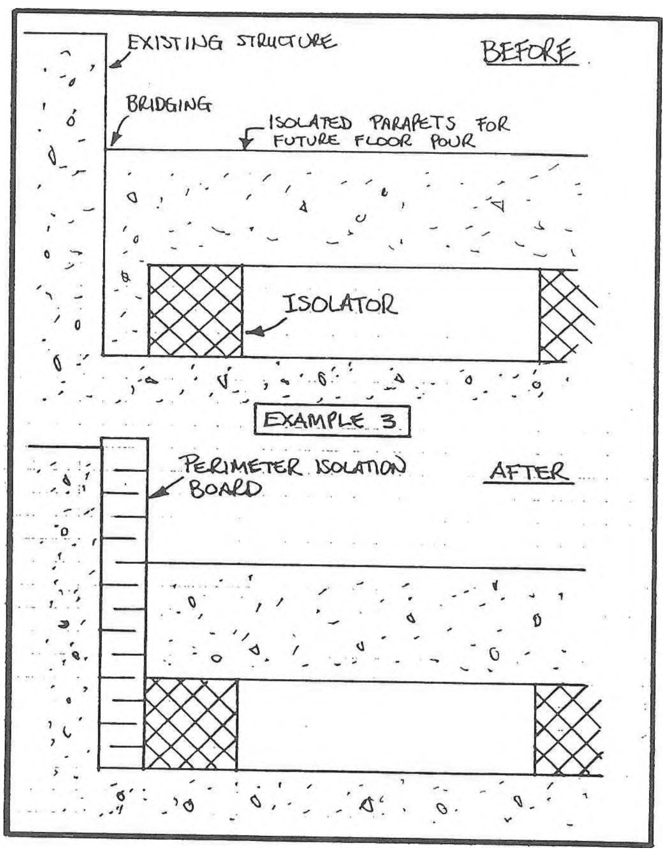
Example 4:

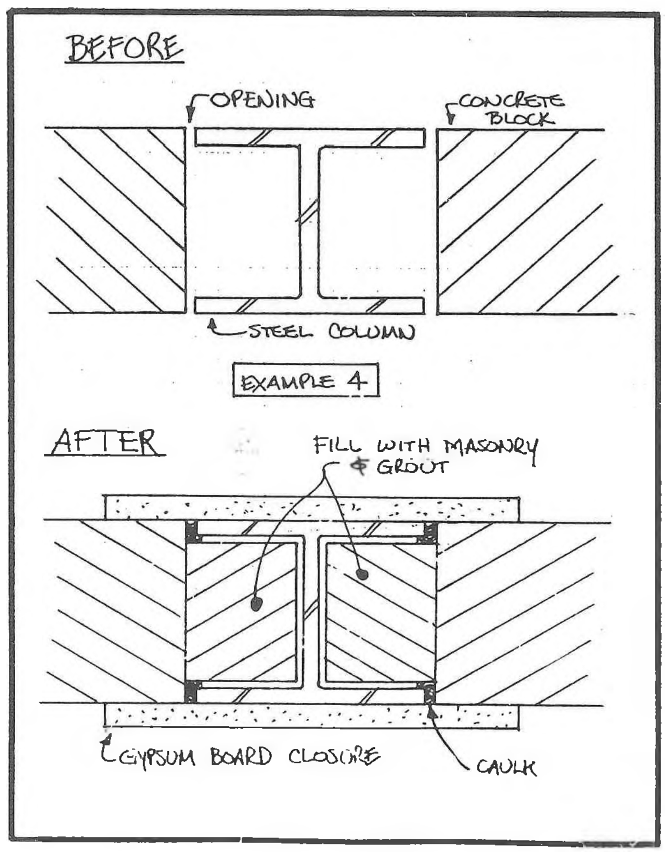
Concrete block wall integrated with steel columns. As built - gap between vertical edges of concrete block and columns, leaving openings through wall around columns.

Solution: Fill cavity with sawcut block, rubble, and grout solid. In some cases, build out column cap of concrete block or GWB.


BEFORE
CAULKING
LINE OF TOP OF WALL
OPENINGS OVER TOP OF WALL VIA FLUTES
EXAMPLE 1
AFTER
TOP OF FLUTE
CAULKED JOINT
BOTTOM OF FLUTE
GYPSUM BOARD CUT TO FLUTE PROFILE
WALL PARTITION


BEFORE
ELEVATION VIEW OF AFTER
OVERSIZE HOLE(S)
THERMAL INSULATION
PIPE(S)
EXAMPLE 2
AFTER
GYPSUM BOARD CLOSURE
CAULK
PIPE(S)


EXISTING STRUCTURE
BEFORE
BRIDGING
ISOLATED PARAPETS FOR FUTURE FLOOR POUR
ISOLATOR
EXAMPLE 3
PERIMETER ISOLATION BOARD
AFTER


BEFORE
OPENING
CONCRETE BLOCK
STEEL COLUMN
EXAMPLE 4
AFTER
FILL WITH MASONRY & GROUT
GYPSUM BOARD CLOSURE
CAULK

# CANADIAN ACOUSTICS ASSOCIATION

# AIRCRAFT NOISE AND LAND USE AT CANADIAN AIRPORTS

**Tom Lowrey**
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**Airports Group, Transport Canada**

## INTRODUCTION

Internationally, in the developed world, noise at airports is being reduced through the introduction of new generation jet aircraft that may be anywhere from 10 to 14 dBa quieter at the same ground location under a flight path than their predecessors. With the passage of a resolution at the special assembly of the International Civil Aviation Organization (ICAO) in the Fall of 1990, the international aviation community took the first step in eliminating the use of noisy second generation jet transport aircraft (those compliant with the noise certification criteria contained in Annex 16 to the International Civil Aviation Convention) from the international fleet.

The principle features of the resolution are that elimination of these jets should occur over a period not less that seven years in length and not begin before 1995. Various countries in the developed world have enacted domestic rules to achieve a phase out of the use of these aircraft from their skies.

The resolution went on to address the issue of individual airports such that where procedures could be enacted to solve noise problems and obviate the need to phase out these aircraft, then they should be introduced. It was recognized that land use in the vicinity of airports should be studied further to attempt to obtain protection from residential encroachment and to consolidate gains in terms of reduced numbers of noise impacted people that would result from the phase out.

## THE CANADIAN SETTING

On March 9, 1992, the Minister of Transport announced that the Canadian air carrier industry would be regulated for the purpose of phasing out the use of the Chapter 2 compliant jet aircraft.

The phase out will be accomplished through a staged reduction, on certain pre-determined dates. Air carriers may comply in one of two different methods, either through a percentage reduction of the number of Chapter 2 jets in their fleets or through an increase in the proportion of quieter Chapter 3 compliant aircraft, to their fleets.

Aircraft that will be phased out include the B737, B727 & DC9.The phasing will begin in 1995 and end in December 2002. Under the proposed regulation, air carriers must;

- on December 31, 1995, reduce the percentage of older aircraft by 25% or increase the percentage of newer aircraft to 55%;
- on December 31, 1997 these percentages must change to 50% and 65% respectively;
- on December 31, 1999, these percentages must change to 25% and 75% respectively;
- after Dec 31, 2002, no Chapter 2 aircraft may operate in Canada.

The only aircraft from this generation that will be left are the gravel kitted Boeing 737 jets that have been modified to operate on northern gravel runways. These aircraft have been exempted from phase out because there is no replacement for them that can operate safely on these runways. They will be permitted to operate at southern airports on their way to or from the north.

The impact of all this is that noise levels are forecast to decrease over the phase out period. Noise predictions are formulated using computer programs that calculate noise level at specified points on the ground. The calculations are based on forecast numbers of aircraft, the type of aircraft expected to be flown, its flight characteristics and noise emission levels, the runways used and the time of day each will be flown at. This last input is used to account for increased disturbance from night flights. A penalty is applied to each flight during the nighttime. The results of the calculations provide a set of contours that are used for land planning purposes. Based on social response studies, public reaction to aircraft noise is predicted and land is developed accordingly.

With the phase out of the noisier Chapter 2 jets, these contours at Canadian airports are expected to shrink over the phase out period with the area impacted by intrusive aircraft noise reduced. It is expected that pressures will be exerted for development on the lands "freed up" from noise. The aviation industry worldwide is looking to land use control authorities for protection from encroachment by non compatible uses. Airports are expecting noise contours to shrink over the period during which Chapter 2 aircraft are being phased out but begin to grow after this period. This growth will be due to traffic growth and the greater proportional use of larger aircraft.

## NOISE CONTOUR SHRINKAGE

The noise contours in place for major airports in Canada, are a land use planning tool depicting noise levels around airports.

Traffic forecasts and noise forecasts presently are predicting a shrinkage in contours up to the final phase out date but thereafter the size of contours are expected to grow again as traffic increases in the next century.

In addition, as new aircraft are modified and enlarged or stretched (to use the vernacular), they will be lifting larger weights and will generate more noise. The 767 is being stretched by 20 feet and strengthened. More weight will be added to the aircraft requiring higher thrust levels and therefore greater noise emissions.

Land use planners and developers are eagerly awaiting the contour shrinkages to take advantage of valuable urban land, close to both cities and airports, that is expected to be freed up for development.

The industry is looking to this shrinkage as a respite from pressure to curtail operations at several airports. Air carriers view relief from pressure for operational restrictions as a payback for the massive investments in new aircraft, airports consider the contour shrinkage as an opportunity to achieve a greater degree of compatibility with their neighbours.

The industry, as a whole, both domestically and internationally, is focusing its efforts on land use planning agencies to promote concerns for careful management of available land adjacent to airports.

Land use planning authorities must be convinced of the need to protect airports from encroachment to avoid non compatible development in adjacent lands and to consolidate the gains made.

## CONCLUSION

In conclusion, aircraft are becoming quieter, the impacts of aircraft noise are being reduced and the public stands to benefit. In order to ensure that these benefits are preserved, land use planning authorities must act now to keep these lands for uses that are compatible with our nation's airports. This country cannot afford to move airports to greenfield locations and waste billions in airport infrastructure investment, not to mention the current positive economic benefits derived from airports, to satisfy the land development lobby.

# IMPLEMENTATION OF NOISE CONTROL MEASURES

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Environmental noise control measures are intended to provide reasonable sound environments indoor and outdoor for residential, and commercial land uses. These measures flow from recommendations outlined in environmental noise studies which are usually required when there is a proposed change in land use. In Ontario, it is normal for noise studies to be:

a) triggered by requirements imposed on the developer at the municipal planning level;
b) conducted by noise consultants;
c) required to conform to generally accepted provincial noise guidelines of the Ministry of the Environment & Energy (MOEE).

Greater emphasis is placed on residential land use and only a few municipalities require noise studies for commercial development.

Noise control measures fall into two categories:

**Indoor measures** address exterior building shell construction - walls, windows, roofs, doors; and mechanical ventilation to allow windows to remain closed for noise control purposes. **Exterior measures** usually address sound barrier construction and location.

Whether and how recommended noise control measures are being implemented into construction is increasingly of concern.

## IMPLEMENTING NOISE CONTROL MEASURES:

Although the processes involved in recommending and implementing noise control measures vary from municipality to municipality, the general approach is:

1) Once the noise study is approved by the reviewing agencies, the study recommendations are incorporated into various agreements such as the subdivision, development and servicing agreements.

2) Prior to obtaining building permit, the building and site plans are reviewed for conformance with the measures recommended within the noise study.

3) Prior to obtaining an occupancy permit or prior to assumption of the development by the municipality, the construction is inspected to ensure the recommended mitigation is in place.

## REVIEW OF ACTUAL MUNICIPAL PROCEDURES:

Municipal staff from a total of 14 Towns/Cities and Regions in Southwestern Ontario participated in a survey to determine how noise control measures were being implemented. Table 1 summarizes the overall results.

### Applicable Standards:

Most municipalities require the use of Ontario Ministry of the Environment and Energy (MOEE) guidelines in noise studies and in determining noise control measures. A few municipalities, however, require their own "standard practice" measures be implemented. Examples include: sound barrier location with respect to municipal or private property; maximum air conditioner sound emission levels; and particular building shell construction.

### Building Design & Site Plan Review:

Most municipalities require a noise consultant to review and certify that building design plans conform to municipal and noise study requirements, prior to issuance of site plan approval or building permits.

### Final Clearance Inspections:

The final clearance inspection is the area of greatest variation and potential shortcoming. About a quarter of the municipal agencies reviewed do not have a requirement for a clearance inspection. Of those which did, the review depended on what is being inspected.

Actual building shell construction and ventilation are normally reviewed by the acoustical consultant.

The inspection procedures for sound barriers vary considerably. Some municipalities require no inspection. Most do, but inspections are conducted by noise consultants or by municipal staff and in one municipality, by the installer. Where barrier inspections are conducted by municipal staff, who does the inspection and what is inspected, varies. Building Departments, Engineering Departments and even Parks Departments get involved; each responsible for a particular aspect. Unfortunately, the relevant aspects of acoustics i.e. barrier densities, gaps, heights, and location relative to source and receivers, are often overlooked. Structural integrity, aesthetics and ensuring barriers are placed on private lands and compliance with applicable fence height by-laws are commonly the focus of the review.

About half of the municipalities which require inspections, indicate that they be done prior to occupancy for each housing unit. The remaining municipalities require inspections prior to assumption of the subdivision. This can lead to problems if physical changes are needed after most or all units are occupied.

### Sound Barriers: Location, Control, Responsibility:

Most municipalities require sound barriers be placed on private lands. Maintenance and upkeep, therefore, become the responsibility of the homeowner. However so does control. Only a few municipalities maintain control by:

a) ensuring all barriers are placed on municipal property, or
b) having restrictive covenants preventing the homeowner from altering or removing noise barriers which are on private lands.

## DEFICIENCIES:

### Coordination:

The lack of coordination between various municipal departments sometimes results in developments being built without either a review of the building plans and/or final inspection. This occurs even within municipalities which require acoustical consultants to do both. As was evident in our review, it is quite common for one department to issue the requirement for building plan review, another the final clearance inspection. Several are often involved in various aspects of the inspection. Coordination and checkoff of items by each department therefore, become difficult.

### Review of Construction Plans:

While the building design is usually reviewed, reviewing the sound barrier (particularly fence design), is often not required.

Wording on Subdivision Agreements often require certification to be in compliance with the Noise Study report. Often there are conflicting requirements resulting from this condition; for example where specific barrier heights have been indicated in the report but grading plans have changed after the report was approved.

The location of air conditioning condenser units, where required, is often indicated on the registered plans as part of the building plan review by the noise consultant. Emphasis is on placement in noise insensitive areas. For requirements to allow provision for future addition of air conditioning, the specific location of condenser units is usually not indicated on building plans and does not get incorporated into the registered plan. The potential exists for units installed in the future to affect neighbours adversely.

**Final Clearance Inspection:**

Inspection provides only a snapshot or sampling of the conditions which exist at the time of the inspection. While many aspects relating to the acoustical inspection cannot be readily verified (if at all), inspectors inherently assume liability for areas reviewed within their expertise. Due to professional liability issues, most noise consultants and municipal inspectors will be careful in restricting the responsibility of the items inspected to those which they can easily verify. For example, an acoustical consultant cannot verify the grading elevations without the aid of a surveyor. Therefore, only fence/wall heights and not top of barrier elevations are "inspected". Similarly, a municipal inspector, who may not be fully knowledgeable in the finer points of acoustics, may restrict inspections to structural integrity.

There are a number of factors and conditions, which depending on the timing of the inspections, make verification of acoustical considerations difficult.

**Grading**: Final grading of the subdivision is difficult to determine on site and the approved grading plans may not be readily available.

**Theft of air conditioning units:** Often results in A/C units being installed at or shortly before occupancy. This results in inspections at short notice and makes scheduling of inspections difficult.

**Access post occupancy:** Many homeowners will not allow access, making post-occupancy inspections difficult.

**Verification of non-visible building shell components**: Interior components of walls such as resilient channels are enclosed at early stages of the construction. Exterior glazing in multiple pane windows is often difficult to measure in multi-storey buildings.

**Sound Barrier Construction:** Inability to assess durability; post inspection settling resulting in gaps; and verifying the species of wood used in acoustical fences; are some of the difficulties inherent to spot inspections.

**Sound Barriers: Control & Responsibility for Upkeep**

Unlike other noise control measures, sound barriers provide mitigation which is shared by multiple residences. Deficiencies in a sound barrier at one location affects adjacent neighbours.

Of major concern in many municipalities is the control and responsibility for upkeep of sound barriers. Barriers placed on private property are often destroyed or damaged, for example, when backyard pools are installed. Decay of older barriers is increasingly a problem.

Maintenance of the sound barriers is usually low on priorities of homeowners who may not have the funds and/or expertise to do proper repairs.

**SUMMARY & RECOMMENDATIONS:**

1) There is too much inconsistency in verifying implementation of noise control measures. A more uniform approach is needed. Guidance could be provided by provincial bodies by issuing model municipal procedures analogous to that of the MOEE model municipal noise by-law.

2) Greater co-ordination between various municipal departments, and a streamlining of the process to ensure plans are certified and inspections are done, are needed.

3) Restrictive covenants placed on title are needed to prevent the destruction of noise barriers placed on private property. Alternatively, barriers can be placed on municipal property, if the municipality will assume maintenance and responsibility.

4) Methods of providing funds which the municipality can use to maintain noise barriers need to be investigated. Most homeowners will not have the funds, expertise or incentive to maintain noise barriers. The municipality should have responsibility and control of barrier maintenance.

5) National and/or industry standards need to be developed to address construction, durability and installation of sound barriers.

6) Spot inspections should be done during construction where non-visible or inaccessible finished components can be viewed. However, this will increase the cost and complexity of the process because of the extra time, and co-ordination required.

7) Ventilation and building shell noise control features should be inspected prior to occupancy to facilitate access and ensure all units affected by noise are inspected. Recognizing that final grading may not take place until after occupancy, noise barriers should be inspected prior to assumption. However, sound barriers should be installed prior to occupancy and sufficient funds withheld (e.g. letters of credit) until assumption, to ensure sound barriers are properly installed.

8) Locations of condenser units should be specified on the building permit certification where required, for both mandatory and provision for adding air-conditioning. Noise by-laws establishing maximum sound emission levels of air-conditioning units should be used to support conditions in the subdivision agreements.

**TABLE I**

**SUMMARY OF MUNICIPAL PROCEDURES**

| Applicable Standards [1] | | Building Design/Site Plan Review | | Final Clearance Inspection [2] | | | |
|---|---|---|---|---|---|---|---|
| | | | | By | | Pre-Occupancy | Assumption |
| MOEE | 9 | Municipal Staff | 0 | Municipal Staff | 0 | | |
| Municipal | 8 | Consultant | 10 | Consultant | 7 | 4 | 3 |
| Rail/Other | 4 | Both Above | 1 | Both Above | 2 | 2 | |
| | | Other | 0 | Other | 1 | | 1 |
| | | Not Required | 3 | Not Required | 4 | | |

Notes:

1. Standards used for noise study and building design/site plan review.
2. All relevant acoustical aspects.

# ENVIRONMENTAL NOISE ASPECTS OF LANDFILL SITE SELECTION

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## Introduction

Finding a suitable landfill site is a complicated process. Many disciplines must interact in selecting a preferred site; Design and Operations, Social Impact, Land Use Planning, Economics, Transportation and Heritage are the main disciplines interacting with Acoustics. As well, public input must be taken into account. Comparing sites from a noise perspective is difficult due to conflicting models for outdoor noise propagation and for community reaction to noise impact.

## The Problems

In Ontario, maximum hourly sound exposures of 55 dBA during the daytime and 45 dBA at night, or the existing ambient, if higher, are applicable for approval of landfills. However, these types of guidelines provide no assistance in evaluating and comparing potential landfill sites. For example, one site may be in a very quiet area (i.e. ambient about 40 dBA) and another may be in a noisier area (i.e. ambient about 50 dBA). Thus, for one site, the increase in daytime sound exposure could be up to 15 dBA while at the other, up to 5 dBA. The site with the lower change to the environment is potentially preferable; even if the applicable guidelines are met at both.

Site selection is a multi-stage process of determining a (long) list of candidates, selecting a short list and finally determining a preferred site, including access routes. The potential zones of real or perceived environmental effects can extend to one kilometre or more around each site and along many kilometres of alternative haul routes. The amount of data gathering required is formidable, as is the time and budget required to study each site in great detail. Meaningful compromises in the technical modelling must be made to allow decision making to proceed with reasonable timing and costs. As the candidate list is shortened, the amount of detail increases.

It is desirable to compare the potential site impacts (i.e. environmental changes) taking into account, qualitatively or quantitatively, the population affected. However, there are no universally agreed-on techniques. One approach is to weight the expected environmental (noise) changes directly by the number of people or family units affected. It may also be relevant to adjust the weighting for sound exposure change, based on where on the absolute scale the resulting sound exposure is. However, there is a paucity of information on the relationship between change and absolute magnitude, and how to formulate the weighting factors; as well as potential disagreement between experts.

Simple population weightings may not be appropriate because landfill operating hours (and times of potential noise impact) tend to correspond to normal working hours. Thus, some people may not be present at their properties to experience the impacts because they leave for work or school.

Aside from comparing the noise impacts between sites, there is also the desire by disciplines such as Social Impact to account for cumulative impacts from unrelated nuisance factors such as dust and smell; or at least be able to integrate and deal with multiple effects in ranking sites. Again, there is little or no guidance in the technical literature as to how noise may interact with the other factors to shape community response.

It may be that the stigma of proximity to a project such as a landfill and the connotation of the noise from site operations or refuse trucks will outweigh the reactions to the characteristics of the sound itself.

Using worst case assumptions is appropriate and commonly done for noise assessments of a selected site. However, this approach is subject to criticism from the public, for site selection. The process could be biased against sites with a large number of proximate receptors even if the potential impacts are low; compared to more remote sites with fewer receptors and much larger individual impacts. The bias may also be due, in part, to worst case conditions occurring for only a small fraction of the site life.

Developing a model which realistically predicts off-site sound exposures involves controversial factors. Atmospheric effects, including wind, temperature inversions/gradients and air absorption will not only vary from day to day, but also very likely from hour to hour. The effects of excess ground attenuation are difficult to apply in practice. The landfill will start below grade and rise up in elevation well above existing grade. Thus, the amount of ground attenuation will vary depending upon the landfill elevation, as well as with wind and vegetation. There are no universally accepted modelling techniques for any of the above mentioned factors.

Using worst case conditions simplifies the analysis. It is thus more practicable because less detailed information about each site is needed. Using more typical or average conditions is subject to criticism by the public because there could be a significant amount of time (e.g. up to 50%) where there could be more noise impact, with no upper bound defined. In any event, adequately detailed site design and operation information to reliably determine typical conditions is generally not available at the site selection stage.

Questions also arise about potential noise effects on wildlife or other animals, on property values, and on businesses. Assessment criteria in these areas are less developed than those relative to impact on people.

## Some Solutions

Noise modelling of site-operations should be based on realistic but somewhat conservative assumptions about equipment, sound emission levels and duty cycles. Activity levels for the peak hour of the "average worst" case day can be used.

To predict off-site sound propagation, the tipping face activity can be modelled as a point source located on the outside embankment of landfill (e.g. half-way up), moving around the site on a locus parallel to the defined operational boundary (toe of embankment).

Contours of sound exposure and sound exposure change based on this locus can be drawn. Clearly, this is representative of worst case conditions since relative to any receptor, this level of sound exposure would not apply for much of the time, when the operations are elsewhere on-site.

The worst hour of the day can be determined for the haul routes based on comparing trucking volumes to ambient traffic. Noise analysis for haul routes is not controversial, using well established traffic noise models such as MOEE ORNAMENT.

Once a preferred site is selected, it will require a detailed environmental noise assessment. In Ontario, the MOEE practice is to not account for excess ground attenuation, regardless of the source elevation. Thus, for consistency, at the site comparison stage, it is desirable to also neglect any ground attenuation effects. Otherwise, there could be significant and undesirable changes in the size of potential impact zones between stages of the study.

Having determined off-site sound exposures, and expected zones of change, the feasibility of various mitigation measures should be considered, together with those designing the landfill and its operations. Generally, at the site selection stage, detailed engineering of cell progression will not be done. However, a preliminary site plan showing the site entrance, primary on-site perimeter haul road, scale house, support facilities and final contours will be available. The purpose of the mitigation is to minimize the zones of change and the number of potential receptors at which the approval guideline may not be consistently met.

With the analyses updated for mitigation, the sites can be compared. Quantitative techniques such as in References 1 and 2 can be used. For each zone or receptor, a composite weighting factor is determined, based on both change and absolute sound exposure. An acoustically weighted population count is then determined to rank sites. Judgements are required on the use of weighting factors. For example, how many people experiencing a change of 2, 3, 4 or 5 dBA are equivalent to individuals with changes of 10, 15 or 20 dBA, and how should the weightings change with increasing sound exposures? In some cases, it may be possible to resolve these issues and how important noise is relative to other factors, by obtaining input from the public to be potentially affected.

Another approach is to qualitatively compare sites using the acoustical data together with lifestyle analyses obtained from detailed interviews of residents in the areas surrounding candidate sites. These letter studies are usually conducted by Social Impact specialist.

Generally, most businesses, wildlife and animals are less sensitive to the type of noise produced by landfills (trucks and engine powered equipment). Potentially noise sensitive situations must be assessed qualitatively on a case-by-case basis.

### Mitigation

Various alternatives may be practicable: choosing quiet equipment where available; perimeter sound barriers/berms to screen internal roads; road network or intersection improvements, particularly at the entrance, to facilitate truck turning movements; operational berms inherent to tipping face design. The operational berm technique starts each cell at the outside edge with a large berm of soil or soil and refuse and then works away from the edge with the trucks and equipment screened by the berm. Technically, off-site mitigation can sometimes be effective, in the form of adding air conditioning, upgraded windows or sound barriers at residential receptors. However, in practice, implementing these methods presents policy and administrative difficulties, as well as potential liability concerns. Compensation for nuisance effects and a property value protection strategy might also be viewed as mitigation. As a last resort, buy-out or expropriation of neighbouring properties that cannot be adequately mitigated can be considered.

### Summary and Conclusions

Current noise guidelines used for approval purposes offer little guidance for site selection. In addition to off-site sound exposure due to the operation, changes to the ambient environment should be considered in comparing sites. As yet, there are no universally agreed-on procedures to compare sites accounting for both factors together. Further research into community reaction to changes at various levels of noise exposure is needed. Nevertheless, there are quantitative and qualitative means to usefully compare and rank sites based on noise. Many of the concerns and techniques would apply to other types of projects besides landfills.

### References

1. "Acoustic Impact Assessment Procedure Used in Industrial Plant Site Selection", F.M.Costlier, Noise Control Engineering, Jan/Feb. 1976.

2. "A Technique For Comparing Alternative Transportation Corridor Alignments Based on Noise Impact", A.D. Lightstone et al, Inter-Noise 92, 947-950, July 1992.

# Uncertainty in Prediction of Environmental Noise Immission due to Ground Effect

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Ontario Ministry of Environment and Energy

## INTRODUCTION

Prediction of sound levels at noise sensitive receptor locations for the purpose of verifying compliance with limits set by an approving agency is a necessary part of the design of a new industrial plant or other noise emitting facility.

One of the main factors influencing outdoor sound propagation, aside from distance attenuation, acoustic shielding, and atmospheric effects is ground absorption loss commonly called "ground effect" which depends in a complicated way on the source-receiver geometry, atmospheric turbulence and their random variations, as well as on the characteristics of the ground surface and the terrain topography.

The effect of ground on acoustic propagation has been thoroughly investigated in the last decades. A large number of theoretical papers have been published on this subject with a broad range of differing results and opinions, and a number of empirical models have been developed to assist in sound level prediction. However, engineering applications are scarce and international standards have not been developed as yet.

Uncertainties in prediction of ground effect along the path from noise sources to distant receptors may have serious implications on the recommended separation distance or the extent of other noise mitigation measures designed to ensure compliance. The results of noise impact assessment for a proposed noisy facility based on a procedure predicting unrealistically high values of ground attenuation may prompt an approval agency to recommend a relatively small separation distance between the facility and noise sensitive receptors which will lead to excessive sound levels and, consequently, to an adverse community reaction. Conversely, an assessment based on an assumption of very low ground absorption loss contribution may result in specification of a large buffer zone and, consequently, the sterilization of land available for development, or to excessive noise abatement recommendations directed at the source to ensure compliance.

## EXISTING STANDARDS

According to (10), Clause 1.2, weather conditions (principally temperature inversions and the following winds) common in temperate climates considerably reduce ground effect and therefore prediction of this sound attenuation component is not part of the standard. However, it is advised in the same clause of the standard, that if ground effect is to be considered, a simple approach using the difference in propagation between hard and soft ground given in Table 3.5 of the CMHC Publication "Road and Rail Noise. Effects on Housing" can be applied.

## MODEL COMPARISON

At present, prediction of attenuation due to ground effect at the design stage is done by a variety of approaches with some similarities. The prediction models developed to include an estimate of ground absorption loss component of sound attenuation are based on a combination of theoretical considerations and empirical experience. They vary in complexity and may provide broadly differing ground effect values.

### Simplified Models

The simplified prediction schemes allow for prediction of ground attenuation as a function of distance and a mean effective height of sound propagation path above the ground along the distance from source to receiver location, and are applied to calculations of the A-weighted sound level due to traffic noise having a predominant frequency component in the spectrum at 500 Hz.

Figure 1 provides a comparison of results obtained using four simplified ground attenuation prediction models (1),(2),(3),(4) for a range of distances from 100m to 300m and one configuration of source and receiver heights of 1.5m. Models in (1) and (2) are well established and widely used, while model in (4) represents a proposal in the draft international standard ISO/DIS 9613. Model in (3) was developed based on experimental results reported in (5).

### Comprehensive Models

These models allow for the calculation of ground effect as a function of frequency. Acoustical characteristics of ground surface between source and receiver, in terms of flow resistivity or a ground factor, is included in the prediction. Although, in general, a homogeneous ground surface is assumed, provision is made in some models for two or three different ground surface characteristics for surface areas close to source, central part and close to receptor, with specific boundary conditions.

A comparison of results for three models (6), (7), and (8) which allow for the calculation of ground effect as a function of frequency is shown in Figure 2. Here, a typical broad band spectrum of a diesel engine was assumed, and ground effect values were calculated in one octave frequency bands. The assumed source spectrum was adjusted for the calculated ground effect values for each model, and A-weighted values calculated for the range of distances from 300m to 1000m. Source and receiver configuration ($Hs=3m$, $Hr=1.5m$) over flat ground was selected to accommodate the range of parameters in the models subject to the comparison, and an acoustically porous grass covered ground surface was assumed.

Results of spectral analysis for three models (7) (8) and (9) are compared in Figure 3 for distances 300m and 800m respectively at a configuration of source and receiver, $Hs=Hr=1.5m$ over flat, grass covered ground surface. S. SPECT. in the legend denotes frequency spectrum of a diesel engine. The remaining three spectra in Figure 3 represent reductions due to ground attenuation calculated from the respective three models (7), (8) and (9).

## CONCLUSIONS

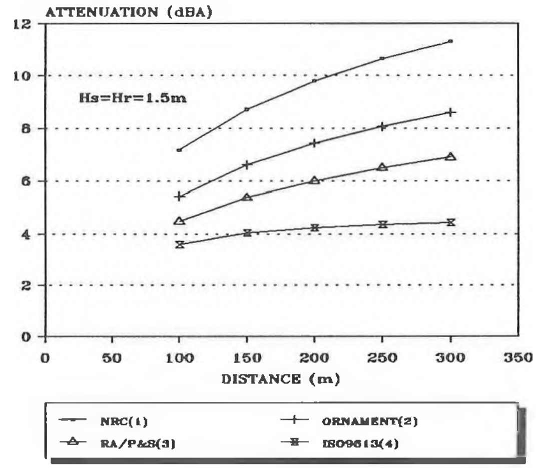
Propagation conditions in terms of atmospheric parameters may be principally responsible for a relatively large variation in results obtained using different models. Model (7) represents theory of interference and ground-wave effects, while (6) and (8) were developed from empirical data collected under specific set of wind and temperature gradient propagation conditions downwind. Model (9) is based on field data obtained under "favourable weather conditions"

A similar conclusion can be drawn from the comparison of results in Figure 1 for the simplified models.

In general, the comparison of models reveals significant variations in the predicted values of ground attenuation, which may seriously compromise the reliability of noise impact assessments. A uniform standarized procedure for assesing the ground effect component of sound attenuation is clearly needed.

The ground attenuation model presented in (8) is currently included in the draft standard ISO/DIS 9613 which is subject to an ongoing review by the ISO Working Group 43.

FIG.1 GROUND ATTENUATION VS. DISTANCE
COMPARISON OF MODELS/TRAFFIC NOISE




FIGU.2 GROUND ATTENUATION VS. DISTANCE
COMPARISON OF MODELS/SPECT.TO dBA

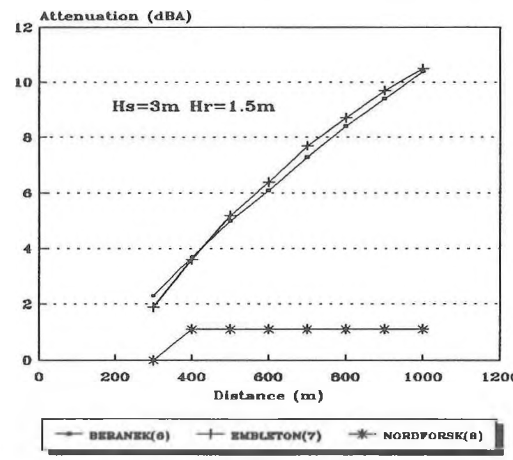
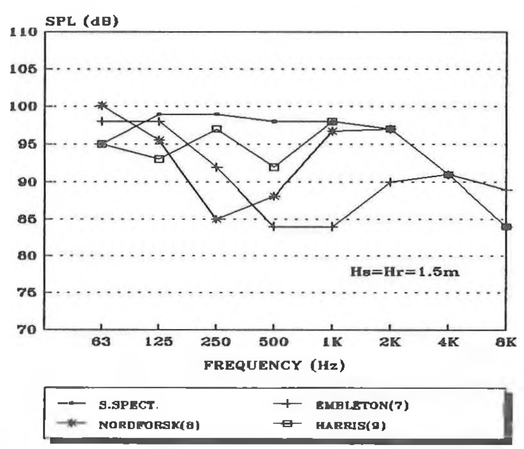



FIG.3 GROUND ATTENUATION VS. FREQUENCY
COMPARISON OF MODELS, D=300m

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## Perception and Production of Syllable-Initial English [r] and [l] by English and Japanese Speakers

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### Introduction.

Various studies have investigated a contribution of multiple acoustic cues to the perceptual distinction of [r-l] phonetic contrast in English (e.g., Dalston, 1975; Underbakke and Polka, 1988). The results of these studies showed that both spectral and temporal properties facilitate a distinction between [r] and [l] sounds. The onset frequency and transition of F3 in relation to F2 is a primary spectral difference needed for differentiation of [r] and [l] by English listeners. For [r], the F3 frequency at onset is low and therefore close to the F2 onset, while in the case of [l], F3 onset is high relative to F2 onset. Moreover, a short F1 transition duration is present at [l] sound and the long F1 transition corresponds to [r] sound in an initial syllable position. Thus, the presence of both spectral and temporal acoustic cues is important for [r] and [l] distinction in prevocalic position. Underbakke and Polka (1988) demonstrated that a trading relation exists between these cues. Thus, in order to enhance the perception of [r] and [l], the perceptual effects of changing one acoustic cue could be offset by changing the other cue in the opposing direction. The trading relation between temporal and spectral cues for the [r] and [l] contrast depends on language-universal phonetic processing constraints, and may be modified in second language acquisition.

As the Japanese language does not have a contrast between [r] and [l] in prevocalic position, these sounds are very difficult to be discriminated, both perceptually and productively, by Japanese adults. Japanese speakers, unlike native English speakers, do not perceive a synthesized [r-l] continuum categorically, and they do not make a distinction between those two sounds productively (Yamada and Tohkura, 1990). This study investigated how native speakers of Japanese, who are living in Canada for many years, perceive and produce Canadian English.

### Method.

**1. Subjects.** Ten native female speakers of Canadian English (age 20-35 years), and ten Japanese female speakers (age 20-53 years), who were residing in Canada, served as subjects. Japanese subjects could be divided into two categories: 2 females who started to acquire English in Canada at an age of 5 years; 8 females, whose first contact with English was in Japanese school at age of 12 years.

**2. Stimuli.** Two synthetic series of nine stimuli each, were generated using parallel/cascade synthesizer KLSYN88a. These series were interpolated in the same steps on the spectral dimension of F2 and F3 onset frequency from "rake" to "lake", but differed on the temporal dimension: one series "r-cue" carrying a temporal pattern typical for [r] sound, and the second series "l-cue" with a temporal pattern typical for [l] sound. Out of these series, the oddity discrimination tests were prepared (Underbakke and Polka, 1988). In each test six repetitions of six stimulus pairs were presented in triads; two stimuli were the same and one was different. All pairs were three steps apart on the spectral dimension. Four types of stimulus comparisons were prepared; a) one cue spectral-"l cue" (varying along spectral dimension with fixed 'l'-temporal pattern), b) one cue spectral-"r cue" (varying along spectral dimension with fixed 'r'-temporal pattern), c) two-cue facilitating (changes in temporal dimension enhanced phonetic discrimination), d) two-cue conflicting (changes in temporal dimension suppressed phonetic discrimination).

**3. Procedure.** Subjects were tested individually on all four oddity discrimination tasks presented in a form of the computer game. Stimuli were presented via loudspeakers at approximately 70 dBA. During a production test, each subject was asked to produce three times the words "rake" and "lake". Recordings were made in an anechoic room, and speech samples were recorded on tape using a microphone B&K 4165, and DAT recorder, SONY DAT-75ES. The recorded speech samples were digitized at a 40 kHz sampling frequency with 16-bit amplitude accuracy. Speech samples were down-sampled to 10 kHz, and the formant frequency trajectories were estimated by an LPC formant tracking method.

### Results and discussion.

The pooled discrimination functions for the English speakers, presented on Figure 1, almost replicate the findings of Underbakke and Polka (1988), as performances in four oddity discrimination tasks are ordered: two-cue facilitating> one cue 'r' = one cue 'l' >two-cue conflicting. Such order of performances reflects the perceptual equivalence of spectral and temporal cues.

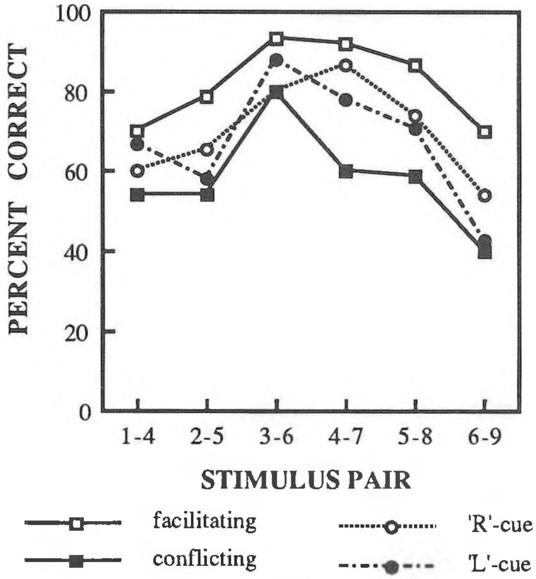


Figure 1.

A different picture emerges, however, from analyzes of the discrimination performance by Japanese, who were exposed to English being over 12 years of age. Their overall performance, for all conditions, was poorer than the English speakers (see Figure 2). Thus, these results might suggest that Japanese speakers did not integrate the spectral and temporal cues. Production results for these two groups of subjects (Figure 3) reflect their perceptual performances. English speakers use both cues, the spectral cue (difference between onset frequencies of F3 and F2), and the temporal cue (duration of F1 transition) in order to distinguish productively [r] and [l] sounds. Figure 3 shows that English [l] sounds (filled squares) are separated from an area of [r] sounds (filled circles).

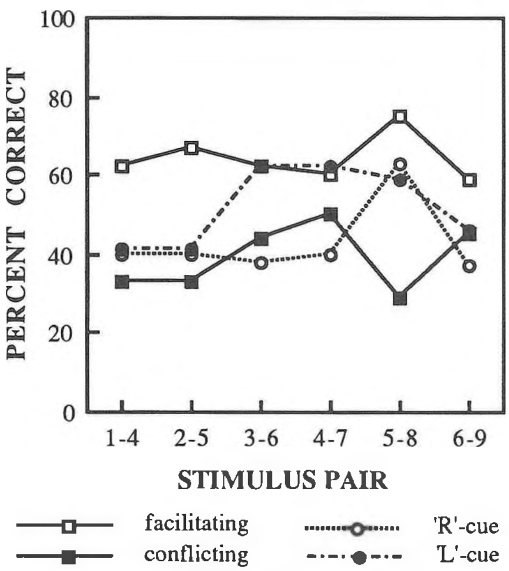


Figure 2.

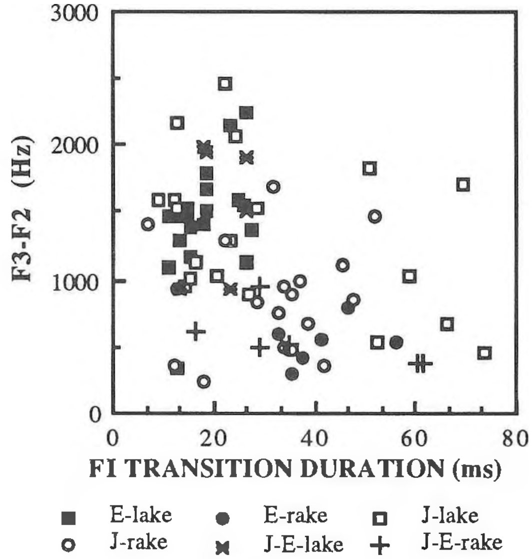


Figure 3.

Japanese speakers, unlike English speakers, use the spectral differences only to a certain extent. The [l] sounds (open squares) produced by Japanese are characterized by slightly larger difference between F3 and F2 formant frequency onsets, than that for the [r] sounds (open circles). However, Japanese speakers do not use the temporal cue. As shown on Figure 3, the areas containing [l] and [r] sounds overlap each other along the 'Transition duration' axis. Performance of Japanese speakers, with early exposure to English (marked on Figure 3 by "*" -[l] sounds, and "+"-[r] sounds), resembles that of English speakers.

The above results indicate, that in an acquisition of phonemes in a second language the perception and the production are strongly related. Secondly, the acquisition of the [r] and [l] phonemic contrast by Japanese after 12 years of age is incomplete, unless a special training is applied.

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## Acknowledgment.

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# The Role of the Auditory Environment in the Development of Speech Production Abilities During Infancy

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University of Calgary

### 1. The Role of the Auditory Environment

The role of the auditory environment in prelinguistic speech development has been controversial since at least 1958 when Brown first proposed the "babbling drift hypothesis" (5). The hypothesis that babble progressively approximates the characteristics of the ambient language has been studied primarily by comparing the characteristics of babble with the presumed characteristics of the adult speech produced in the child's environment, with particular emphasis on sounds which are excluded in the adult language, and sounds which are exclusive to the adult language (in comparison with a second language, usually English). Thus far the results have been somewhat contradictory. Locke (5) reviewed studies covering 15 different language environments, and concluded with confidence that the auditory environment does not influence speech production until the age of 18 months. Other studies strongly suggest that the auditory environment may have a significant impact on speech before the end of the first year.

Recent advances in research regarding the role of the auditory environment in speech development have been made possible by an improved description of the infant's auditory environment, a better understanding of infant speech production abilities, and more sophisticated analyses of both infant and adult produced speech. Often the infant's auditory environment is described by reference to phonetic analyses of adult-produced, adult-directed speech (e.g., radio monologues and telephone conversations). A more complete conceptualization of the infant's auditory environment must take into account the infant's auditory abilities and attentional preferences, aspects of the environment which may mask or modify linguistic input, and characteristics of speech heard by the infant, including self-produced speech, adult-produced infant-directed speech (i.e., motherese), and adult-produced adult-directed speech.

This research has also been hampered by a failure to consider the interaction of the auditory environment with the infant's developing motor abilities. For example, some studies have examined cross-linguistic differences in voice-onset-time and fundamental frequency contours (5). However, there is little variation in these acoustic characteristics either within or between infants, and there is reason to believe that these parameters are particularily difficult for young infants to control.

Finally, the bulk of this research has involved phonetic analyses of both infant and adult speech. Phonetic analysis of infant speech has been criticized on a number of grounds: it is subject to listener biases; similarities among samples are emphasized while differences are obscured; only segmental aspects of speech are described; description of vocalizations which are not grossly speech-like are impossible; and it assumes, unrealistically, that infant vocalizations are composed of the same auditory and articulatory features that characterize adult produced phonemes (2).

The problems listed above have been avoided in more recent research with both cross-linguistic and hearing impaired samples. For example, de Boysson-Bardies et al. (3) described a unique method for determining the phonetic characteristics of the infant's linguistic environment; in this study the adult phone frequencies were taken from the adult models for words produced by 20 month old children, and then compared with phone frequencies in prelinguistic speech samples taken from younger children. This procedure increased the probability that the adult repertoires represented what adults say to infants, rather than the more typical sampling procedures which represent what adults say to each other. In addition, these comparison repertoires are more likely to reflect the infant's hearing abilities and attentional preferences. Comparative analysis of the consonant repertoires of 9 to 16 month old infants learning French, English, Japanese, and Swedish found reasonably good correspondence for place and manner of articulaton, word final consonant use, and vocalization length. de Boysson-Bardies and her colleagues (1) have also found striking differences in vowel formants between samples collected from 10 month old babies learning French, English, Cantonese, and Arabic.

Acoustic analysis has also had a profound impact on the study of speech development in hearing impaired children. Oller (6) cites the traditional belief that the age of onset and quality of babbling produced by hearing impaired infants is the same as for normally hearing babies. However, an acoustically based metaphonological analysis reveals that the onset of canonical babbling is significantly delayed for these children. Kent et al. (4) studied the speech produced by twin boys, one with normal hearing and one with profoundly impaired hearing, during the period 8 through 15 months of age. Acoustic analysis revealed that the hearing impaired baby's vowel space became increasingly restricted with age, while the vowel space of the normal hearing baby changed shape to resemble that of the adult speaker of English.

### 2. Otitis Media and Infant Speech Development

The studies cited above deal only with severe to profound hearing impairment. It is not known whether the more subtle and fluctuating hearing impairment that is associated with otitis media with effusion (OME) has similar effects on speech development during infancy. It is known that OME is especially prevalent during the period 6 through 18 months of age and that children with a history of chronic OME are at risk for speech, language, and learning difficulties. In addition, studies show that the risk for speech and language delay is especially high when the episodes of OME begin during the first year of life (7).

We are currently engaged in research designed to examine the impact of OME on infant speech production abilities. Such studies are necessary if we are to develop: models to explain the relationship between OME and speech and language delay; measures to predict which infants will experience speech and language delay secondary to chronic OME; and treatment programs to prevent speech and language delay in such children.

We are recruiting 6-month old infants with no prior history of hearing impairment or otitis media ("OME-free" group) and 6-month old infants who have had one or more ear infections at or before 6 months of age ("OME" group) through physicians, audiologists, and community health nurses. Each infant in the study has a normal prenatal, perinatal, health, and developmental history and comes from an English-only speaking family with no history of speech, language, or learning disability.

Continued membership in the assigned group (i.e. "OME" or "OME-free") is dependent upon the results of tympanometry at each assessment: infants in the OME-free group will be required to maintain normal tympanometric results throughout the period of study, while infants in the OME group will be required to demonstrate abnormal tympanometric results during at least one of the assessments. Early age of onset is generally associated with increased risk for chronic OME, and consequently it is likely that infants in the "OME" group will demonstrate abnormal middle ear status on more than one occasion between 6 and 18 months of age.

At each of the ages 6, 9, 12, 15, and 18 months the infant recieves a complete standard audiology assessment including otoscopy, assessment of warble-tone and bone conduction thresholds for frequencies between 250 and 6000 Hz using visual reinforcement audiometry, and tympanometry . Following the audiology assessment the infant's vocalizations are recorded for approximately 30 minutes. Phonetic and acoustic analyses are being used to determine a phonetic complexity score and canonical babble ratio for each sample, and the F2/F1 ratio for each vowel contained in each sample. Developmental changes in these measures will be compared across the 2 groups.

The figure below shows some prelimary data for two infants currently enrolled in the study. Baby A, a boy, had normal hearing and middle ear function one week prior to taping and no history of OME. Baby B experienced 3 ear infections, according to parent report, prior to the age of 6 months. He had elevated thresholds and flat tympanograms one week prior to taping but normal hearing and middle ear function one week after taping subsequent to antibiotic treatment. Baby A was taped at age 5 months 2 weeks, while Baby B was taped at age 6 months 3 weeks. The speech samples were analyzed by first segmenting all of the infant's vocalizations into utterances, defining the utterance as a vocalization bounded by adult speech, 1 second of silence, or an audible inspiration. Each utterance was then classified as speech-like or non-speech, and then each speech-like utterance was further classified as canonical or noncanonical (6). The frequency of the first and second formants of each vowel contained in a canonical syllable was determined using autoregression analyses and fast fourier transforms on samples digitized at a sampling frequency of 20 kHz. The figure below plots the resulting F1 and F2 value for each vowel. Baby B is clearly producing a more restricted set of vowels in relation to Baby A, showing a less mature pattern of vowel production ability, despite being almost 5 weeks older than his normal hearing peer.

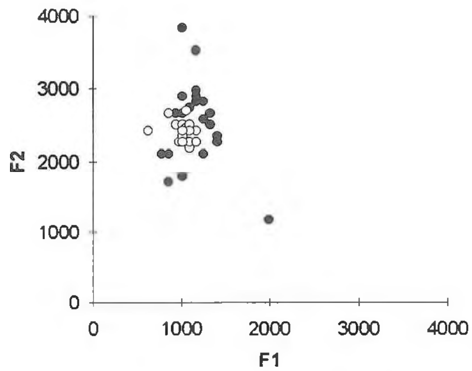


**Figure 1.** F1/F2 plot for vowels produced by Baby A (closed circles) and Baby B (open circles) at age approximately 6 months.

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### 4. Acknowledgements


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# HANDLING FALSE STARTS IN RECOGNITION OF SPONTANEOUS SPEECH

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## 1. INTRODUCTION

Most previous acoustic analysis of speech has examined data from speakers who carefully pronounce their speech, usually by reading prepared texts. Natural spontaneous or conversational speech differs from that of careful or read speech in several ways, the most obvious difference concerning hesitation phenomena. In spontaneous speech, people often start talking and then think along the way. This causes spontaneous speech to have interruptions; the specific interruption phenomena studied in this paper are restarts (or false starts), which are interruptions in the flow of speech, where the speaker (usually after a brief pause) reiterates a portion of the speech immediately preceding, with or without a change. The repetition can range from a portion of a syllable up to several words. In the case of a change, the modification may be either a substitution of a new word (in the place of a fully- or partially-spoken previous word) or an insertion of a word in a word sequence (with the sequence containing the new word being uttered again).

This paper concerns the acoustic analysis of restarts in spontaneous speech, from the point of view of their automatic location via acoustical analysis. A large database of spontaneous speech was analyzed in terms of duration and fundamental frequency measurements, as well as spectral analysis. For recognition purposes, a simple spectral analyzer was used to identify repeated words.

The restarts are described acoustically, with a view toward automatic recognition, to ensure their proper elimination from consideration in speech recognition systems. A primary application of this study lies in improving the performance of automatic speech recognizers, for applications that must accept an input of spontaneous speech (e.g., verbal conversations with computer databases). For such purposes, we wish to eliminate one version of any repeated words (or parts of words), and in the case of changed words, we wish to suppress the original unwanted words, so that the recognizer will operate on only a sequence of desired words. Thus, we examine here the relationship of restarts to intonation, and do so in a fashion that should allow direct exploitation in automatic recognition systems accepting spontaneous, continuous speech.

Within-utterance hesitations can cause significant difficulties for automatic speech recognizers, which usually make no provision for repeated words or parts of words. Automatically determining which words (or parts of words) are being replaced in a speech repair could help automatic recognizers avoid textual errors in the output. In virtually all current recognition systems, words repeated in a false start are either simply fed as word hypotheses to the textual component of the recognizer or cause difficulties in having a proper interpretation in the language-model component (since the language model is invariably trained only on fluent text).

## 2. PREVIOUS STUDIES

Acoustical analyses of disfluencies with a view toward speech recognizers are extremely rare. (To our knowledge, the only such work is recent and found in [1–2].) Previous work on restarts has dwelled almost exclusively on the length of the word-repeat sequences (and occasionally on the pause duration). Most of the work on restarts that has been reported in the literature has treated the phenomena in a general qualitative or overly simple quantitative fashion [3–5]. As far as we know, no reports have previously linked the intonational cues of both F0 (fundamental frequency) and duration to restarts in a way that could be useful to automatic speech recognition. Indeed, very few recognition systems use intonational cues, especially F0, at all. In this paper, we examine how these latter parameters could be exploited directly.

Recently an attempt was made to automatically detect and correct restarts in spontaneous speech [1–2]. Looking at an enlarged version of our own database, the authors examined 10 000 utterances, of which 607 were found to have restarts. In utterances longer than nine words, a significantly high 10% had restarts. 59% of the restarts involved only one word (whose deletion would render the sentence fluent); 24% involved two words (or word fragments).

These authors [1–2] tried to automatically locate and correct these restarts, first using text alone (assuming that a speech recognizer could provide a correct transcription) and then using cues from the speech itself. Based on simple pattern matching of the text alone (e.g., looking for repeated words, cue words, and simple syntactic anomalies), their algorithm had a relatively high error rate for location: missing 23% of the utterances that had restarts and producing false alarms in 38% of the proposed cases. The rate for correcting the restarts (for the properly located ones) was only 57%. After inclusion of a language model, they were able to detect 85% of the restarts, based on text input. They noted that simple repeats usually have a significant pause (mean = 380 ms) between the repeated words, whereas actual (intended) repeated words (e.g., 'flight five one one') have very brief (if any) pause [6a]. They further noted that, for restarts where one word was repeated with a new inserted word prior to it (e.g., '... flight [pause] earliest flight...'): 1) if a pause was present adjacent to the new, inserted word, it was found before the word, and 2) the new word had a higher peak F0 than the preceding word.

The approach of these last authors is similar to ours (and we use the same database), but we address the question of automatic location of restarts, and without prior knowledge of neither the word boundaries nor the identity of the words. In practical recognition applications, one certainly cannot assume such prior knowledge. Thus it is quite practical to examine how such disfluencies can be found directly from an acoustical analysis of the audio (speech) signal.

## 3. SPEECH DATABASE

In this paper, we examine disfluencies in a standard speech database (used by several speech recognition research groups in North America), ranging from simple restarts (involving only the repetition of 1–2 words) to complex restarts (where, instead of simply repeating words, one substitutes a new word for an unwanted one).

In the context of our investigation into voice dialog access to databases, we are currently examining an application involving a simulated travel agent. A naive user (the speaker) is given the task of arranging a trip involving air travel via commerical airlines, by verbally interacting with a "computer travel agent." Thus, the user formulates verbal questions and commands in a spontaneous fashion, as if in conversation with a travel agent. (The current system does not reply verbally, but rather outputs information from a database onto a computer screen.) The spoken data consists of 42 adult male and female speakers, each speaking about 30 different utterances, each ranging in length from a few words to several dozen words (median length of about 12 words).

In the approximately 1000 utterances examined (from many different speakers, each containing an average of about thirteen words), there were 60 occasions where the speaker simply repeated words or portions of words, 30 cases of inserted words, and 25 occurrences of new words substituted for prior spoken words (or word parts). Thus, approximately 10% of the utterances (a percentage consistent with the parallel study of [4–5]) had a restart.

## 4. ANALYSIS METHOD

Hardcopy displays were made of all utterances containing restarts (as determined by listening and transcribing each utterance), in sections of 3-5 seconds at a time. Each display contained a waveform (amplitude vs. time) and a narrowband spectrogram (showing 0-2 kHz). Time resolution in these displays ranged from 44 to 78 mm/s; the frequency axis showed

39 mm/kHz. These displays were manually segmented into words and syllables, and F0 contours were obtained by tracing strong harmonics in the middle of the first or second formant.

## 5. ACOUSTICAL ANALYSIS RESULTS

With false starts, when a word was simply repeated (as is) in a restart, it had virtually the same prosodics (i.e., same duration and pitch) in both its instances in most cases, but there were a number of times where the repeated word had less stress (i.e., shorter duration and lower pitch). When a word was changed (i.e., a substitution or insertion) in the restart, on the other hand, its second instance was virtually always more stressed (i.e., longer duration and higher pitch).

In the case of restarts where the speaker stopped in the middle of a word and simply "backed up" and resumed speaking with no changed or inserted words, the pause lasted 100-400 ms in 85% of the examples (with most of the remaining examples having a pause of about 1 second in duration). About three-fourths of the interrupted words did not have a completion of the vowel in the intended word's first syllable (e.g., the speaker usually stopped after uttering the first consonant). In virtually all examples, the speaker completed at least 100 ms of the word, however, before pausing for at least 100 ms. When the pause occurred at a word boundary, the words repeated after the pause were characterized by two situations: either a straight repetition with little prosodic change (this happened especially when a lengthy pause intervened), or a repetition where the repeated words shortened up to 50%.

In the case of a word being substituted or inserted into the word sequence in the restart, the substituted/inserted word received a large stress (relatively long duration and rise in F0) in examples where the new word added significant semantic information, but did not in examples where the new word was redundant in terms of the prior context (e.g., if the new word was a synonym of an immediately previous word). As for the repeated words (after the pause) prior to the inserted word, function words showed little or no shortening, but usually had lower F0; on the other hand, content words here exhibited significant shortening and lower F0 (the shortening here was about 50% for short words less than 300 ms, and about 100-200 ms for longer words). Such prosodic change only applied to non-prepausal words, because words immediately prior to a pause were often subject to significant prepausal lengthening.

We concentrate now on simple repeat restarts (i.e., those where words or parts of words were simple repeated, with no change in their content), because they were the most promising to recognize automatically. These can be divided into utterance-initial restarts and utterance-medial ones. At the very start of an utterance, speakers often start saying something and immediately stop, having uttered only a few syllables or even a fraction of a syllable (e.g., 'Wh- @ what I want...,' 'I'd @ I'd like to know...', where '@' represents a pause). The pauses in these cases were very variable, ranging from 80 ms to a few seconds, unlike the vast majority of simple repeat pauses in utterance-medial position, which were in the range 100-400 ms. When the pause at the start of an utterance exceeded 400 ms, after speech of fewer than three syllables, the speaker usually restarted his utterance rather than continuing as if no pause intervened.

## 6. RECOGNIZING RESTARTS

Since pauses involved in restarts were generally shorter than other pauses [1], we could suggest a simple rule of "pause < 400 ms -> restart." For our database, such a rule will correctly identify 70% of restarts, but will give 35% false alarms (i.e., incorrectly claiming as restarts those grammatical pauses which are shorter than 400 ms). While this performance is well above chance, it is clear that pause duration alone is not a reliable cue to a simple restart. Also, restart pauses at the very start of utterances were quite variable in duration (the 400 ms rule is more reliable when applied to pauses found after 3 syllables of an utterance). Obviously, the spectral-time detail on either side of a pause must be examined to verify whether a restart is present.

Since most restarts are simple repetitions, looking for identical spectral-time patterns (of up to 3 syllables in length) on either side of a short pause will greatly increase the restart recognition accuracy. For simple repetitions, the scope of spectral analysis is very limited: one need only look at about 2-3 syllables before and after each candidate pause. Very few simple repetitions repeated more than three syllables (a significant portion of complex restarts, on the other hand, involve more than three syllables). If a close spectral match is found and the pause exceeds a low threshold (e.g., 120 ms - to avoid confusion with stop closures), we declare that the pause is a simple restart, and that one version (usually the first) of the matching syllables should be excluded from consideration in any ensuing recognition process. We were very successful in automatically recognizing such simple restarts.

Recognizing restarts with changed words appears to be much more difficult than identifying simple restarts. We look for a short pause (again < 400 ms), followed by a spectral-time pattern containing 1-2 syllables corresponding to a portion of the speech immediately prior to the pause. However, there are many possibilities here and many of them have spectral and prosodic patterns that resemble fluent speech (i.e., speech without repeated or substituted words, but having pauses). For example, after the pause in such a restart, the immediately ensuing word(s) may be the added/substituted ones, or there may be one or two repeated words (from before the pause). The added/substituted words may be as short as one syllable or as long as six syllables. Due to the difficulty of distinguishing complex restarts from fluent pauses, a simple algorithm for identifying such restarts awaits further research.

## 7. CONCLUSION

This paper has detailed the extent of prosodic phenomena in speech restarts in a multi-speaker database of spontaneous, continuous speech, and has given intuitive explanations for them, based on a theory of using prosodics to cue semantic information to a listener. Based on the acoustic data, ways have been described as to how to attempt to recognize these phenomena in the context of an automatic speech recognizer.

Simple restarts can be distinguished acoustically, via an analysis of duration, F0 and spectral detail in the neighborhood of a pause. Restarts with changed words may be distinguishable, but the required analysis will need to be much more complex. It will require a detailed examination of the pitch and durations of the pauses and adjacent words, along with acoustic recognition of words or syllables. Unfortunately, the wide variety of possibilities seen in this study for restarts with a modification does not suggest a simple recognition algorithm at this time.

## ACKNOWLEDGMENTS

This work was supported in part by grants from the Natural Sciences and Engineering Research Council of Canada, the Fonds pour la Formation de Chercheurs et l'Aide a la Recherche (Quebec), and the Canadian Networks of Centres of Excellence program (IRIS).

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Discrimination of Frequency Transitions: or, Can You Distinguish the Different Birds Chirping by the Waterfall?

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### Overview

One method which has become increasingly popular to study complex phenomena such as speech perception is the use of non-speech analogs (eg. frequency transitions). In this study, the discrimination of frequency transitions was examined as a function of frequency region, trajectory direction, and presence of background noise.

### Methods

Subjects:

Subjects were 5 young adults aged 19-30 years (mean 24 yrs) audiologically screened for normal hearing.

Stimuli:

Eight continua of 17 signals (60 msec in duration including 5 msec rise/fall times) were sampled at 20 kHz, 12 bit resolution, low passed at 5 kHz and output through a Macintosh II computer. Four continua were centered at 1 kHz; four at 3 kHz. At each frequency region, for both upward and downward trajectories, signals converged on a common offset frequency or diverged to varying offset frequencies. Increments were 10 Hz steps at 1 kHz; 30 Hz steps at 3 kHz.

Procedure:

A two-alternative forced-choice paradigm designed to determine the just noticeable differences (jnds) was used. Trials consisted of 2 stimuli with an ISI of 500 ms and the required response was 'same' or 'different'. Stimuli were delivered monaurally via AKG headphones at 65 dB SPL. For converging signals, speech spectrum noise was added and presented at an S/N ratio of +5 dB.

### Results and Discussion

Logit transformations were computed on percentage correct scores as a function of frequency separation. Average psychometric functions were calculated with logistic analysis and are shown in Figures 1 and 2.

(NOTE: FOR ALL FIGURES: D = Diverging; C= Converging; DN = Down; N = Noise)

At 1 kHz, the functions for signals with a downward trajectory show a displacement to the right relative to the other series. The reverse pattern is seen for the 3 kHz signals where the signals which sweep downward are shifted to the left of those which have an upward trajectory.

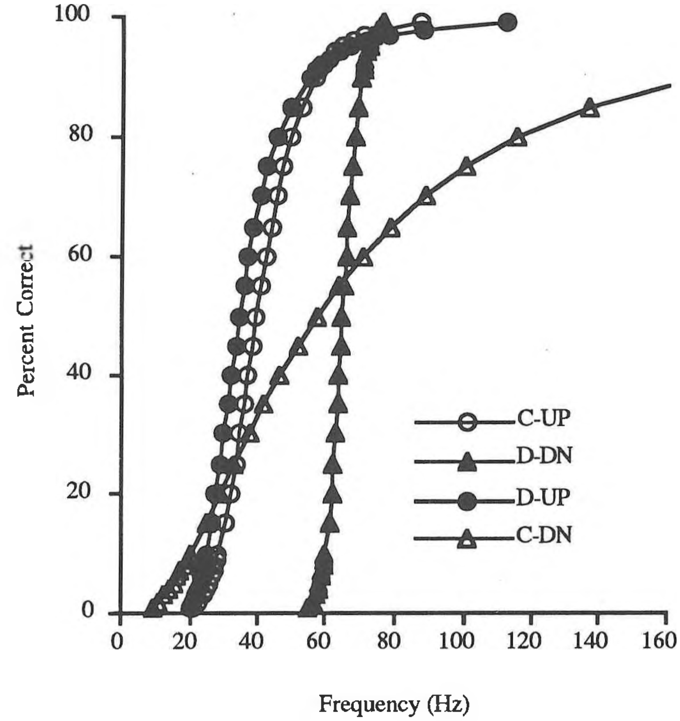


**Figure 1. Functions at 1 kHz**

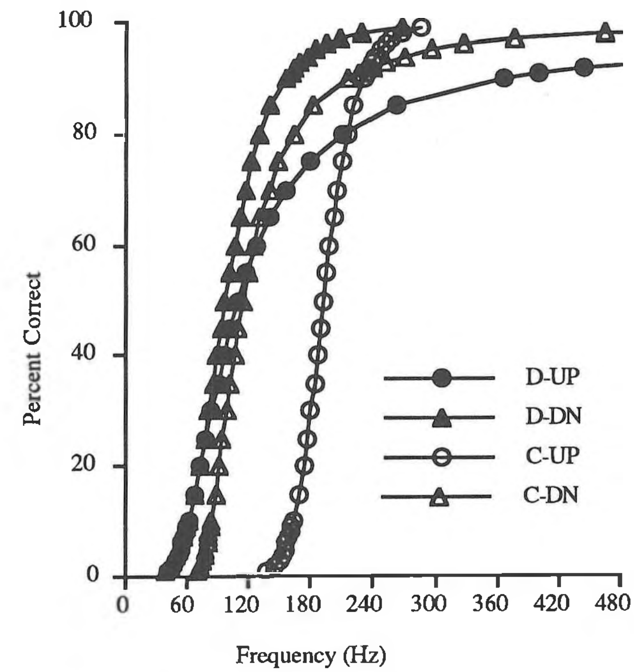


**Figure 2. Functions at 3 kHz**

Mean jnds (defined as the 70% correct position) across signal type are summarized in Figures 3 and 4.

A 2-factor within-subjects analysis of variance (transition direction: up or down, and end frequency: diverging or converging) was conducted separately for the data at each frequency region. At 1 kHz, there were significant main effects of direction [$F(1,4) = 101.1$, $p < .001$] and end frequency [$F(1,4) = 35.22$, $p < .001$]. At 3 kHz, the effect for direction was significant, [$F(1,4) = 22.15$, $p < .01$; however, neither the effect for end frequency, nor the interaction term were significant. At 1 kHz, diverging offset frequency signals were easier to discriminate than converging signals. Decreases in frequency over time were more difficult than increases in frequency over time. At 3 kHz, signals which decreased in frequency over time, regardless of whether they converged or diverged to differing offset frequencies, were easier to discriminate then signals which swept in an upward trajectory. There was no significant effect for 1 kHz signals embedded in a background of noise. At 3 kHz, only signals which converged upward to a common frequency were adversely affected by noise.

There are two major findings from this study: a minimal effect for noise and frequency dependent directional sensitivity. Directional preference for different frequency regions is consistent with other findings involving both human (Porter et al, 1991) and animal studies (Heil et al; 1992). Asymmetries in tuning curves where above 1 kHz, the high frequency slopes are very steep, might also account for the finding that signals in this region which decrease in frequency over time may provide 'clearer' cues for discrimination.

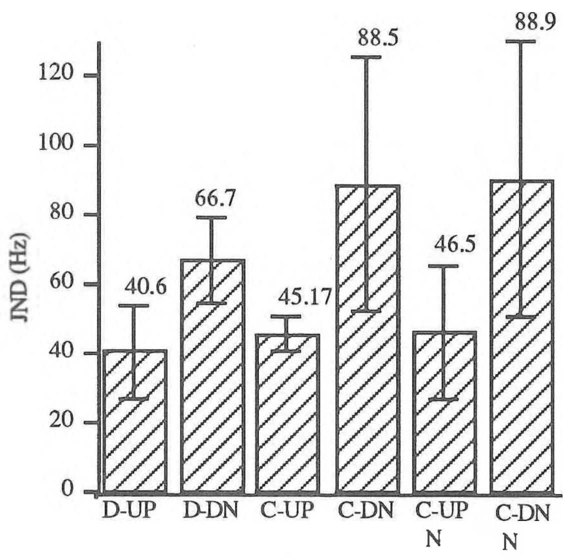


**Figure 3. Thresholds at 1 kHz**

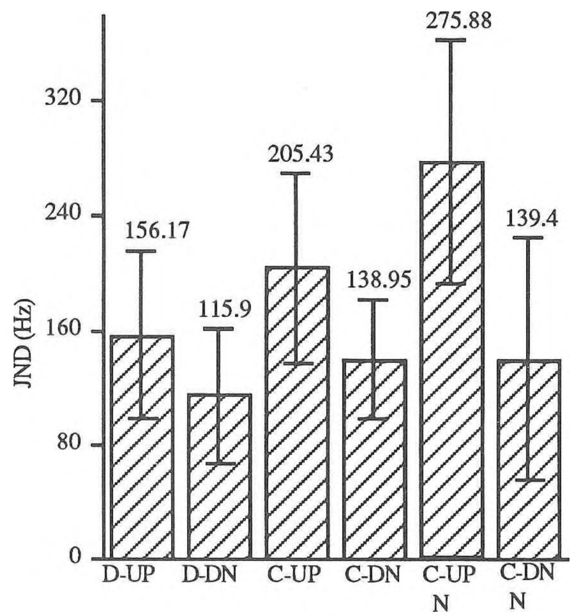


**Figure 4. Thresholds at 3 kHz.**

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# KINEMATIC ANALYSIS OF COARTICULATION AND SPEAKING RATE

by

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## INTRODUCTION

During speech production, the acoustic and kinematic properties of certain sounds and articulatory configurations are changed under the influence of adjacent sounds. This is termed "coarticulation." Previous studies have resulted in two competing theories of anticipatory coarticulation: 1) look-ahead or feature spreading models and 2) coproduction or frame models. The look-ahead model suggests that the anticipatory activity begins as soon as there are no competing requirements of the articulators, and is therefore relatively free to vary in its onset time (e.g., Henke, 1967). The coproduction model suggests that an anticipatory activity begins at a fixed time before the actual target (e.g., Bell-Bert and Harris, 1981). Recently, Perkell and Matthies (1992) proposed a hybrid model of coarticulation, which combines components of these two models. They suggested that there may be two phases to a lip protrusion gesture. The first phase involves a slow, gradual onset of protrusion that begins as early as permitted by the competing requirements of the articulators. The second phase is a more rapid protrusion movement, which is time-locked to the onset of the rounded vowel. The authors suggest that the maximum acceleration of the lip signal separates the two phases of the protrusion gesture.

Previous studies have examined coarticulation by manipulating intervocalic consonant durations before a rounded vowel, typically by changing the number of intervocalic consonants. The timing relationship of select movement, electromyographic, or acoustic events related to lip rounding are then analyzed. In the current study, intervocalic consonant durations were manipulated by having speakers produce utterances at systematically varied rates of speech. Stetson (1951) suggested that systematic modifications of rate may reveal the underlying control mechanisms which operate during speech production. Investigations into the manipulation of rate may therefore provide us with information regarding how the motor system is organized to coordinate the articulators for the complex movements required for speech.

## METHODS

Six normal speakers served as subjects; 3 were considered "Young Normal" (YN) and ranged in age from 20 to 25 years; 3 subjects were considered "Older Normal" (ON) and ranged in age from 50 to 78 years. All subjects were native English speakers. Subjects produced the sentence "I see two and I see tea again," fifteen times at six different rates.

Rate was manipulated using a magnitude production task (c.f. Adams et al., 1993). Subjects produced the sentence at a self-selected normal rate of speech. Subjects were then asked to produce the sentence at two and four times their normal rate, one half and one quarter their normal rate, and maximally fast. Rate conditions were randomized for all subjects. Upper lip protrusion, transduced using strain gauges (Barlow et al., 1983), and the acoustic signal were digitized.

Upper lip protrusion was clearly detected as an upward movement of the signal during the production of "I see two," while the movement trace did not deflect upward during the production of "and I see tea again." This indicates that the observed lip protrusion was not inherent to the production of the consonant /t/, as has been proposed by Gelfer, Bell-Berti and Harris (1989). Rather, the trace deflection can be attributed to upper lip protrusion for the /u/ in "two."

The duration from the onset of upper lip protrusion to the acoustic onset of /u/ was measured and labelled "Movement Interval." Similarly, the duration from maximum upper lip acceleration to the acoustic onset of /u/ was measured and labelled "Acceleration Interval". "Consonant Duration" was measured as the time from the last glottal pulse of V1 (that is, /i/ in "see") to the first glottal pulse of V2 (that is, /u/ in "two").

## RESULTS

Qualitatively, four distinct patterns of upper lip protrusion were observed, with patterns varying both within and between subjects. For some subjects, the pattern appeared to be dependent on rate, while for other subjects, the pattern was clearly not related to rate.

The most frequently occurring pattern was characterized by a smooth onset of upper lip protrusion, and a single velocity peak. A second pattern was characterized by double peaks in the velocity profile, with one velocity zero crossing. A third pattern was similar to this; however, there were two zero crossings in the velocity profile. The fourth, and final pattern was characterized by a very slow, irregular increase in velocity, that often hovered just above zero for an extended period of time. A weak trend indicated an increase in the multiple velocity peak patterns with slower rates of speech. However, this trend was not observed for all subjects. A slightly stronger trend indicated that Young Normal subjects exhibited fewer double zero crossings and a higher number of single-peaked productions than the Older Normal subjects. However, due to the small number of subjects in each age group, only limited conclusions can be drawn from such findings.

Based on quantitative results, the onset of upper lip protrusion was similar to that observed by Perkell and Matthies (1992). That is, movement onset typically began early, during or even before /i/ production in "I see", and appeared to be independent of rate. Perkell and Matthies hypothesized that various regression lines would be expected for pairs of values of acceleration interval and movement interval versus consonant duration, according to the three models of coarticulation discussed earlier. According to these authors, the look-ahead model predicts a positive slope for both plots, suggesting that both the acceleration interval and the movement interval vary with consonant duration. The coproduction or frame model predicts slopes of zero for both plots, suggesting that neither the acceleration interval nor the movement interval change with changes in consonant duration. The hybrid model proposes that the acceleration interval is constant across consonant durations, with a slope of 0, while the movement interval changes with consonant duration, with a positive slope.

In the current investigation, the relationship of both acceleration and movement intervals to consonant duration was examined using regression analysis. The $R^2$ values, provided in Table 1, indicate that regression accounted for between 0.7% and 78.2% of the variance. Because the variability cannot be accounted for with a low $R^2$, those values below 45% were not analyzed further.

The slope of movement interval versus consonant duration was substantially greater than 1 for all three Young Normal subjects and for Older Normal 2. This indicates that the onset of the protrusion gesture began earlier relative to the acoustic onset of /u/ as consonant duration increased. This finding does not support the coproduction model, in which protrusion onset would be time-locked to the /u/.

Subjects Young Normal 1 and Older Normal 2 demonstrated fairly high $R^2$'s for both acceleration and movement intervals. Protrusion onset began earlier as consonant duration increased, as demonstrated by the slopes of the movement interval. Additionally, the slopes of the acceleration interval versus consonant duration were 0.952 and 1.250, for YN1 and ON2, respectively. This indicates that the maximum acceleration also occurred earlier relative to /u/ as the consonant duration increases. These findings, particularly for these two latter subjects, support the look-ahead model, which predicts that both onset and maximum acceleration of lip protrusion will begin earlier relative to /u/ as consonant duration increases.

## DISCUSSION

These data indicate that the time of lip protrusion onset varies with the intervocalic consonant duration. As consonant duration increased, lip protrusion began earlier relative to the acoustic onset of /u/. This pattern was distinctly observed in four of the six subjects. This finding lends support to the look-ahead or feature spreading model, which suggests that movement for an upcoming segment may be initiated as soon as its onset does not interfere with the immediate articulatory requirements. It was also observed that the time of maximum acceleration of the protrusion gesture covaries with consonant duration, rather than remaining time-locked to the onset of /u/. Perkell and Matthies have proposed that maximum acceleration may be temporally related to the acoustic offset of the preceding vowel. This will be examined in further analyses of the data.

These data demonstrate that there are changes in the pattern of coarticulation with alterations in rate. It appears that the motor system allows for flexibility in timing lip protrusion as we manipulate speaking rate. Previous research on coarticulation, in both normal and disordered speakers, has typically not controlled for speaking rate. Therefore, we need to interpret the literature with caution. Future studies should take into account the effect of rate on the timing of coarticulation.

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## ACKNOWLEDGEMENTS

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Table 1.

| | | Slope | $R^2$ |
|---|---|---|---|
| YN1 | Acceleration Interval | 0.952 | 69.9%* |
| | Movement Interval | 2.350 | 59.8%* |
| YN2 | Acceleration Interval | 0.390 | 9.2%* |
| | Movement Interval | 1.090 | 56.1%* |
| YN3 | Acceleration Interval | 0.238 | 2.2% |
| | Movement Interval | 2.640 | 78.2%* |
| ON1 | Acceleration Interval | 0.548 | 0.7% |
| | Movement Interval | 1.480 | 12.4%* |
| ON2 | Acceleration Interval | 1.250 | 45.0%* |
| | Movement Interval | 1.650 | 73.3%* |
| ON3 | Acceleration Interval | 0.446 | 4.3% |
| | Movement Interval | 1.170 | 12.6%* |

* $p < .05$

# Voice Onset Time and Vowel Duration Across Multiple Speech Rates in Normal and Parkinsonian Speakers

by

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Individuals with Parkinson's disease (PD) present with motor control deficits evident in decreased speech intelligibility[1]. Additionally, poor control of speech rate has been implicated in the speech deficits of PD[2,3]. Subsequently, rate manipulation has been used extensively as a clinical treatment of choice with this population[4]. However, it remains to be determined whether the disordered speech of PD speakers is a consequence of impaired speech motor control or of the abnormal speech rates they produce. That is, are slow speech rates inherently different from normal rates? This question holds for both neurogenically normal and disordered speakers.

Studies employing speech rate manipulation allow examination of speakers' motor control systems under varying conditions. These studies offer unique opportunities for drawing conclusions about the organization of speech motor control. Studies have revealed differences between the speech of fast and slow talkers. For example, Crystal and House[6] reported that there was greater variability for their three slow speakers than for their three fast speakers suggesting there are differences in motor control at various speech rates.

This paper reports initial analysis of speech at six self-selected speech rates, by neurologically normal and PD speakers. The questions addressed include: (1) Are there essential differences between the speech of normal and PD speakers at different speech rates? and, (2) Do normal and PD speakers vary segment durations differently at different speech rates?

## METHODS

**Subjects:** Four PD and five neurologically normal speakers between 50 and 78 years old served as subjects.

**Stimuli:** Subjects produced three different utterances, 15 times each, at six different speech rates. The six rates and three utterances were elicited in random order. Data were audio recorded and later digitized and analyzed using the Kay Elemetrics Computerized Speech Lab. For the present study analysis was conducted on one of the three stimuli, the utterance "Buy Bobby a poppy." The following measures are presented: total utterance duration, voice onset time (VOT) of the initial /p/ in poppy, and vowel duration of the /a/ in poppy.

**Magnitude Production:** Eliciting speech at different rates has been accomplished through a variety of procedures such as modelling and pacing. However, these rigid techniques do not easily allow the elicitation of a wide range of rates within a subject, and they force the speaker to attempt motoric adjustments that may not be natural for that speaker. Therefore, this investigation used a magnitude production task to elicit speech at six different rates. This procedure involved asking subjects to produce an utterance at their normal speech rate. Following practice subjects were told to think of this rate, their normal rate, as being equal to the number '10.' They were then asked to speak twice as fast as their normal rate. Subjects were given the opportunity to practice and the number '20' was then assigned to this rate. Similarly the number '40', was assigned to a rate four times as fast as their normal rate, '5' to half their normal rate, and '2.5' to one-quarter their normal rate. Finally, subjects were asked to speak as fast as they could, this rate was termed 'MAX'. The numbers and the word MAX were always present in front of the subjects when they were speaking. Before each rate condition subjects were reminded of what the number meant (e.g., 20 = 2xs as fast as normal). Subjects were instructed to produce each repetition on a single breath and to not use increased interword pause time to lengthen their total utterance durations. This was practised and monitored on-line, and when it occurred that token was repeated. Thus subjects autophonically scaled the six different rates at which they spoke.

## RESULTS

**Total Utterance Duration:** All subjects, both neurologically normal and PD, were able to produce speech at six different rates. Within subjects, the means of adjacent rates (e.g., MAX vs 40 or 10 vs 20) often did not differ greatly. However, total utterance duration differences were present both within and across all subjects across the six target rates. In general, the total utterance durations produced by PD speakers were slower than the normal speakers at each rate. Furthermore, the quantity of rate change, relative to the self-selected normal rate, was less for the PD than for the normal speakers. This was true for both faster and slower rates (see Table 1).

**Table 1. Total Utterance Duration (ms): Buy Bobby a poppy**

| RATE | NORMAL | PD |
|---|---|---|
| Maximum | 820 | 730 |
| 40 = 4 x Normal | 872 | 759 |
| 20 = 2 x Normal | 1007 | 823 |
| 10 = Normal | 1193 | 982 |
| 5 =  x Normal | 1483 | 1212 |
| 2.5 =  x Normal | 2399 | 1493 |

**Voice Onset Time:** Both subject groups generally increased VOT as the target speech rate decreased. However, when looking at the individual data it appears as if two distinct patterns of speech production were present (see Figure 1). For four of the normal subjects and one of the PD subjects VOT increased rapidly as total utterance duration increased. The remaining subjects presented with minimal changes in VOT across speech rate changes. $R^2$ values ranged from 3.5 to 70.6 and 4.0 to 32.3 for the normal and PD subjects, respectively. Three of the normal and one of the PD subjects had $R^2$ values over 30%. Additionally, the overall quantity of VOT change accompanying changes in total utterance duration was less for the PD speakers than the normal speakers.

**Vowel Duration:** Inspection of Figure 2 reveals that all normal and two PD subjects greatly increased mean vowel duration as they increased total utterance duration. The remaining two PD subjects produced relatively stable mean vowel durations as they increased total utterance duration. $R^2$ values ranged between 20.3 and 88.5 and 0.1 and 71.9 for the normal and PD subjects, respectively. However, $R^2$ values for four normal but only one PD subject were over 30%.

## CONCLUSIONS

Across the different rates PD speakers produced shorter durations than did the normal speakers; they spoke faster. More importantly, they did not vary their speech rate, relative to their self-selected normal rate, as much as the normal subjects. This may be related to reduced range-of-motion reported in PD. It also appears that some PD subjects varied their segment durations systematically in manners similar to most normal speakers, while others did not. Analysis of additional subjects will reveal whether these are truly patterns of motor control or simply artifactual. Finally, low $R^2$ values for both VOT and vowel duration for the PD subjects indicates that their speech was highly variable. It may be that it is this high degree of within speaker variability that contributes to reduced speech intelligibility in PD speakers.

**Figure 1a. VOT vs Total Utterance Duration, Normal Ss**

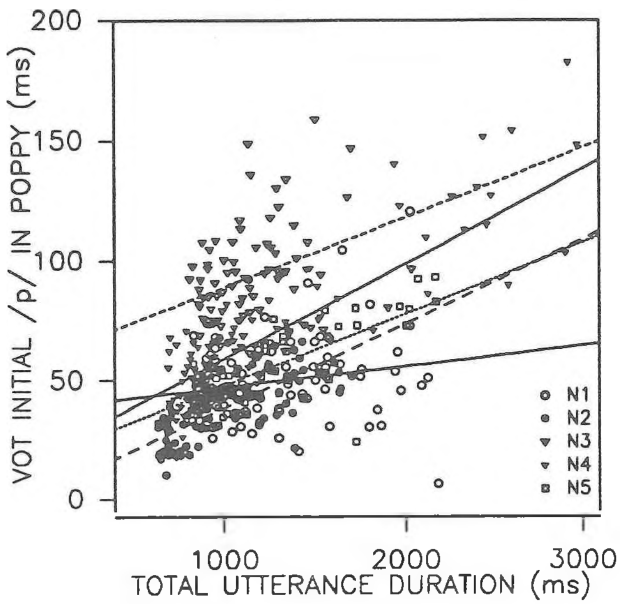



**Figure 1b. VOT vs Total Utterance Duration, PD Subjects**

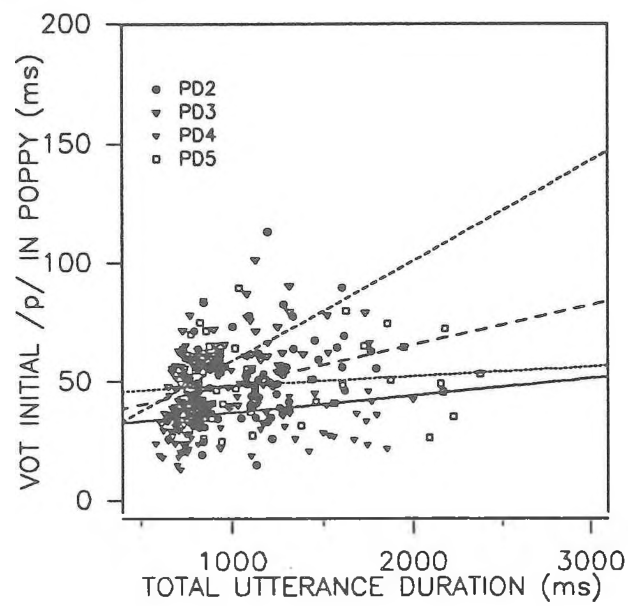



**Figure 2a. Vowel vs Total Utterance Duration, Normal Ss**

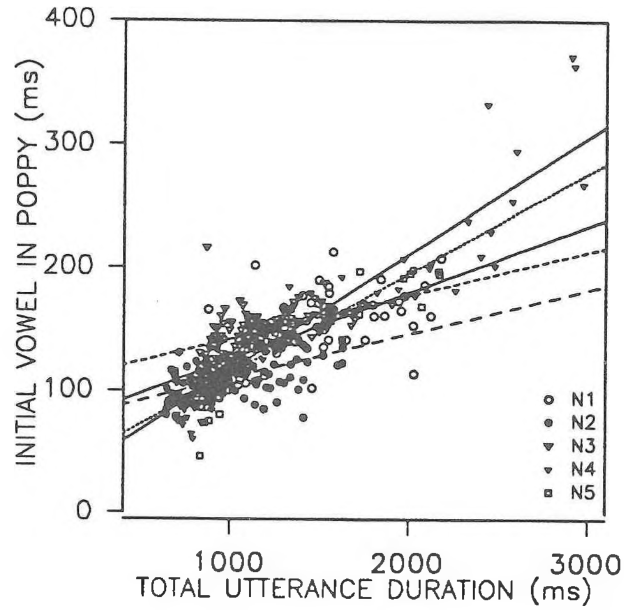
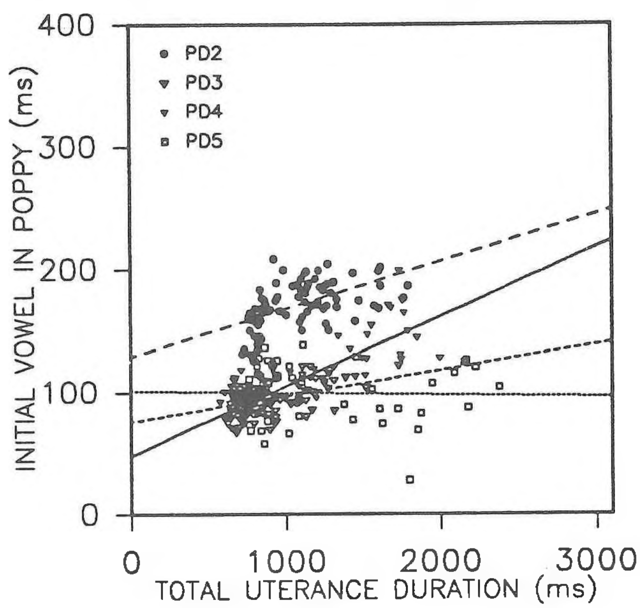



**Figure 2b. Vowel vs Total Utterance Duration, PD Ss**

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# Acoustic design for noise reduction

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## 1. Introduction

One of the main challenges that mechanical engineers will have to take up in the next years will be to design lighter but quieter structures. Actually, engineers have essentially three main types of approaches for tackling noise problems:

(i) Classical control tools such as muffler, barriers, absorbing materials, enclosures which are still useful but not very convenient in terms of added weight and cost and very often not feasible in regards of the production constraints.

(ii) Control by an appropriate structural acoustic and vibration design which consist essentially in modifying mechanical parameters of the structure: change the dimensions, the material characteristics, add stiffeners, add local masses, add damping internally et externally, or modify the fixations conditions, decrease the energy transmission between coupled structures or systems [1].

(iii) Active noise control which may be divided in two main categories. Active noise control through anti-noise which consist essentially in finding the appropriate signal to feed to speakers to cancel the noise in specific positions where control microphones have been placed. It is actually the preferred approach for controlling the noise in internal volumes such as cavities and ducts. The other way, is to actively control the "bad" vibration which means the vibration which causes the noise. This is a more challenging problem which implies that you must know in great details the link between the acoustic and the vibratory energy [2]. This method uses essentially piezos sensors and actuators for sensing and acting on the structure.

In this paper, we will concentrate on the second approach the one dealing with modifying or creating an appropriate design which minimizes the radiated noise.

## 2. Theoretical predictions

Structural acoustic and vibration design cannot be made by trial and error methods. It has been demonstrated many times in the past that it is too complex to be handled by chance. The most classical example is addition of stiffeners. Doing so, is beneficent for the vibration level, however often times the radiated noise in not reduced but may even increase.

To predict the acoustical power radiated by a structure one has to solve a mechanical integro-differential system which governs the equation of movement of the structure as well as the wave equation for the surrounding fluid.

These fundamental equations can be solved (i) either by discretizing the displacement in terms of a modal basis or any convenient functional, (ii) or by discretizing the structure itself and/or the volume which involve the use of finite element or boundary element approaches. This is not a simple task. More details on these methods can be found in the following references [3,4].

To fix approximately the ideas these two methods have some advantages (+) and drawbacks (–) which are described in the following tables.

Table 1

| Analytical approach |
|---|
| The + <br> • This method which is more physical, authorizes a physical insight of how and why structures radiate noise. <br> • This method increases the difficulties in terms of analytical calculations but reduces the computation time (about 10 to 100 times quicker than a numerical approach). <br> • This method allows for the study of low and even medium frequencies. <br> • The programs based on analytical methods are easy to use and are especially efficient for parametric studies. |
| The – <br> • This method is limited to simply shaped structures and volumes (plate, cylinder). <br> • This method looses its advantages when the structures become complex (holes, variable boundary conditions) <br> • The method becomes heavy when dealing with coupled substructures. |

Table 2

| Numerical approach |
|---|
| The + <br> • In principal, the method is valid for structures or volume of any shape, with weak or strong fluid coupling. <br> • The method benefits regularly from the increasing capacity of computers (computer time and memory). <br> • Once validated, this method can be used instead of experiments when measures become too costly. <br> • Commercial codes are available. <br> • The method allows inter connection with other fields, such as fluids mechanics and heat transfer, using also numerical approaches. <br> • It is undoubtedly the method of the future. |
| The – <br> • the method is limited to low frequencies for the moment. <br> • The method tends to be used as a black box with the risks its implies. The user of commercial codes based on numerical approaches is better to be quite knowledgeable in structural acoustic and vibration. <br> • The method necessitates access to efficient computers and cannot be performed decently on a P.C. |

## 3. Acoustic design

In order to find design ways of controlling the noise, the authors have been deeply involved in developing the previous approaches for various mechanical configurations going from simple to more complex structures. These models have been developed to solve real problems submitted by several industries. It must be mentioned, here, that noise control at the source by design modification is a VERY COMPLEX and TEDIOUS task. There are absolutely no simple solutions, and research and development involve years and not weeks or months. There are no general rules neither. Each industrial problem is a particular one, with its own type of excitation, geometry (so its own model response and its own radiation factor), and type of coupling between its subsystems. However, in order to help clarify the situation, we have tried to summarize in Table 3 the great tendencies that engineers have to keep in mind when they want to deal with noise control by design at the source.

## 4. Conclusions

To face noise control at the source in the future, engineers will have to work closely with researchers to profit from the new knowledge and the new tools available. Analytical and numerical tools can be used in conjunction to get a better understing of the phenomena and to help find a better design.

The general tendencies given here and obtained from parametric studies have to be confronted with practical cases to get more confidence in their value. The challenge is huge since, although general tendencies are now well mastered, each industrial products reveals a very particular case with a very high degree of complexity, not handled at present by the different prediction models.

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Table 3

| Mechanical modifications | Vibratory response | Radiation efficiency | Sound power |
|---|---|---|---|
| Increase of stiffness (measures, stiffeners) | ↘ especially in low frequencies | ↗ especially in low frequencies | • ↘ in low frequencies<br>• even or ↗ in medium frequencies<br>NOTA: always better to increase the added stiffness than the number of stiffness |
| Increase mass (local point mass) | ↘ especially in high frequencies | ⟶ | ↘ for high frequencies |
| Increase mechanical impedance mismatch at the junction between excitation and structure | ↘ broad band effect | ⟶ | ↘ broad band effect |
| Increase damping | ↘ at the resonances | ⟶ | ↘ only at resonance frequencies and above the critical frequency |
| Dimension: tend towards square surface | ↘ less modal density | ↘ | ↘ |

# A formulation for the vibro-acoustic behavior of a rectangular plate with constrained-layer damping.

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## 1. Introduction

In an attempt to reduce the resonant bending of a structure, intensive research on plates vibrations have been done over the years. It is well known that flexural vibrations in a plate can be damped by application of a viscoelastic layer constrained by a thin, elastic layer. Furthermore, the addition of a spacer between the plate to be damped and the viscoelastic layer have been found to enhance the damping performance. Kerwin [1] has shown that energy from such a structure is dissipated when shear deformation is induced in the viscoelastic layer. The spacer and the constraining layer enhance the damping effectiveness by inducing additional shear deformation in the viscoelastic layer.

The purpose of this work is to develop a rapid but rigorous tool to help acoustics engineers understand and predict the vibro-acoustic behavior of a constrained-layer damping of a plate. A rectangular four layered simply supported baffled plate is considered. In addition, the plate is assumed to be semi-complex in the sense that it can support added masses, stiffeners and several types of excitation (i.e point, line, surface forces and moment). The problem is formulated using a variational approach and solved by the Rayleigh-Ritz method. The modeling of the stiffeners is based on an equivalent orthotropic layer. Since the plate is assumed to radiate in air, added mass due to fluid loading is ignored. However, possible cross modal coupling due to stiffening or the type of the excitation is accounted for. This is done using a novel method for evaluating the radiation impedance matrix based on multipoles expansions of Green's kernel [2]. The numerical evaluation of the radiated power is done easily from the radiation impedance matrix.

## 2. Theoretical model

Several studies have been devoted to modeling plates constrained damping [3,4,5]. The most comprehensive is based on the Reissner-Mindlin's hypothesis which assumes that each layer could have pure bending, shear deformation and traction-compression effects [5]. These models yield an accurate representation of constrained damping but need long computational time. In addition, most existing studies have been limited to the vibrational problem. The model developed herein is inspired from the work done by M.R. Garrison et al. [3]. It is a simplification in comparison with the Reissner-Mindlin's model in the sense that it uses the appropriate assumptions for each layer. The displacement field of the elastic outer layer is considered to allow for pure bending (Love-Kirchhoff's assumptions) and traction-compression effects.

$$
\begin{aligned}
u_i(x,y,z,t) &= u_i^0(x,y,t) - z\frac{\partial w(x,y,t)}{\partial x} \\
v_i(x,y,z,t) &= v_i^0(x,y,t) - z\frac{\partial w(x,y,t)}{\partial y} \\
w_i(x,y,z,t) &= w(x,y,t)
\end{aligned}
\tag{1}
$$

where $u_i$ and $v_i$ are the transverse displacements of the outer layers ($i = 1$ for the plate to be damped and $i = 3$ for the constraining layer). $w$ is the normal displacement of the plate.

For the viscoelastic inner layer it is assumed that it can support both bending, traction-compression and shear deformation. The displacement field in the viscoelastic layer is completly defined by the continuity of the displacement at its junction with the constraining layers. Consequently the displacement field in the vicoelastic layer is obtained by linear interpolation between the two outer layers displacements and hence it is expressed in term of the assumed displacement of the outer layers .

$$
\begin{aligned}
u(x,y,z,t) &= \frac{z}{h_2}\left[u_3^0 + \frac{1}{2}\frac{\partial w}{\partial x}(h_3 + h_1 + 2h_s) - u_1^0\right] \\
&\quad + \frac{1}{2}\left[u_3^0 + \frac{1}{2}\frac{\partial w}{\partial y}(h_3 - h_1 - 2h_s) + u_1^0\right] \\
v(x,y,z,t) &= \frac{z}{h_2}\left[v_3^0 + \frac{1}{2}\frac{\partial w}{\partial x}(h_3 + h_1 + 2h_s) - v_1^0\right] \\
&\quad + \frac{1}{2}\left[v_3^0 + \frac{1}{2}\frac{\partial w}{\partial y}(h_3 - h_1 - 2h_s) + v_1^0\right] \\
w(x,y,z,t) &= w(x,y,t)
\end{aligned}
\tag{2}
$$

$h_1$, $h_2$, $h_3$ and $h_s$ represent respectively the thicknesses of the plate to be damped, the viscoelastic layer, the constraining layer and the spacer.

Since the spacer is assumed to be rigid in shear and to have no bending stiffness, its motion follows the motion of the plate to be damped. Furthermore, in order to allow for the equivalent orthotropic modeling of stiffeners, the four layers are assumed to be orthotropic thus the stress field is deduced from the displacement field using an orthotropic matrix of elasticity, which satisfies the plane stresses hypothesis for each layer.

Once the displacement field is defined, the problem is formulated using a variational approach. The functional of Hamilton is written as

$$
H(\vec{u}) = \int_{t_0}^{t_1} (T - V + W)\,dt \tag{3}
$$

where $T$ is the kinetic energy (rotary inertia is neglected), $V$ the potential energy, $W$ is the work of the external forces and $\vec{u}$ represents the unknown displacements.

In order to study pure bending, shear deformation and traction-compression effects, five degrees of freedom are needed, hence there are five unknown displacements. These unknown displacements are written in terms of series of sine functions that allow for simply supported boundary conditions. The coefficients of the sine functions are solved for using the Rayleigh-Ritz approach which leads to the classical system of linear equations

$$
[M]\{\ddot{a}\} + [K]\{a\} = \{f\} \tag{4}
$$

where $[M]$ and $[K]$ represent respectively the mass and stiffness matrices which are deduced from the kinetic and potential energy. $\{f\}$ is the generalized vector of forces and $\{a\}$ is the vector of the coefficients of the unknown displacement field.

## 3. Results

The developed model has been validated by comparison with a model based on Reissner-Mindlin's theory for the three layers[6], and good agreement has been found for the different vibro-acoustic indicators.

In order to show the effect of the constrained-layer damping, the results (mean square velocity, acoustic ratiated power, radiation coefficient) of the present model (without a spacer) are compared with the predicted behavior of an undamped plate. All of the results are obtained using a primary layer which is 0.533 m long, 0.203 m wide, 1.9 mm thick and whose density is 2762 kg/m$^3$ (aluminum). The Young's modulus is 68.9 GPa and the loss factor is 0.005. The thickness of the viscoelastic layer is 0.1 mm, the Young's modulus is 3.45 MPa, the loss factor is 0.1 and the density is 1024 kg/m$^3$. The constrained layer is made with aluminum whose properties are the same as the primary layer and a thickness of 0.25 mm.

Figure 1 shows the effect of constrained damping on the mean square velocity. This effect is more pronounced at lower frequencies and around the resonances. The radiation efficiency is shown in Fig. 2. As expected it is unchanged by the treatment, because it only characterizes the primary layer. Finally, the radiated power decreases mainly due to the decrease of the mean square velocity as seen in Fig. 3.

## 4. Conclusion

The proposed model allows for an accurate modeling of the physics with minor computational effort. This is important since the main objective of the study is the development of a simple and accurate model for viscoelastic damping. Furthermore, the model combines well with a novel integral approach for the calculation of the radiated field thus allowing for the investigation of the effect of viscoelastic damping on the radiated field.

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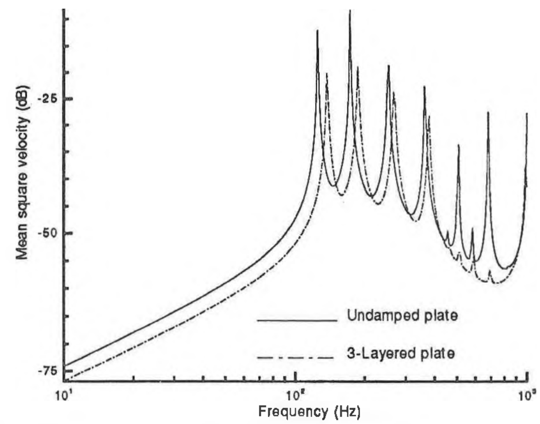


*fig. 1 Mean square velocity, comparison between an undamped plate and the same plate with constrained layer damping.*

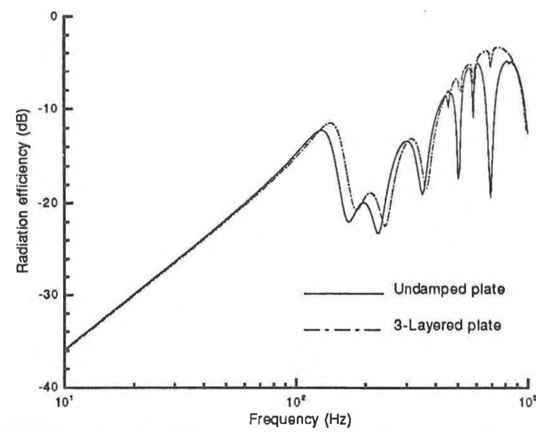


*fig. 2 Radiated coefficient, comparison between an undamped plate and the same plate with constrained layer damping.*

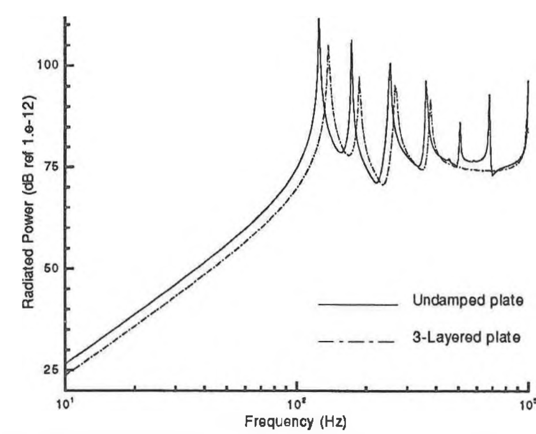


*fig. 3 Radiated power, comparison between an undamped plate and the same plate with constrained layer damping.*

# ACTIVE NOISE CONTROL SIMULATIONS IN A CAVITY-BACKED FLEXIBLE PLATE SYSTEM

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## Introduction

Manufacturers today are facing international competition and complete customer satisfaction is quickly becoming the focus of all the different echelons within an organisation. The transport industry is a good example of a competitive market where a product's acceptance is primordial and the stakes considerable. Among attributes of importance to customers, studies have found that acoustic discretion is among the first three. Most transport means are plagued by a high levels of noise in the passenger environment. In order to achieve reasonable noise reductions, manufacturers are confronted with choices which usually requires non-negligible space, adds weight, increases cost and complexity of manufacturing tasks (the more so if in their interest lie in the low frequency range). Recently, with the advent of technological advances in computing power, digital data aquisition and signal processing techniques, the possibility of attacking the noise problematic with counter noise measures has developed and commercial systems are being tested on aircraft [1,2] and automobiles [3] to eliminate undesirable tonal components. In transport vehicles, noise is produced by vibration and acoustic sources. The excitation produced by such sources eventually reaches the passenger compartment through many types of paths. One of these paths (structureborne) leads to noise radiation inside the passenger cavity by flexible plate type structures. Analytical and experimental investigations have been initiated on a cavity-backed simply supported plate system which has for principal goal the comprehension of the fundamental principles and the determination of the applicability of active noise control techniques for structureborne noise. The present work aims at the development of a rapid and complete tool to simulate the efficiency of active control schemes for low to mid range frequencies.

In the 1980's, publications began to appear on the active suppression of enclosed sound fields. Rigorous work has been conducted by Nelson et al. [4] on the active control of sound inside a rigid paralellepipedic enclosure. Their work was stimulated by propeller aircraft application and culminated by the actual design and testing of an active control system. Fuller et al. [2] have published extensively on similar applications but their studies concentrated on flexible cylinders and plates with control of interior sound field generated acoustically or structurally by point force actuators, acoustic drivers and piezoelectric devices. Pan et al. [5] have furthered the concepts to a cavity-backed flexible plate system. They have simulated and experimentaly studied the control of the cavity sound field due to a primary excitation (acoustic drivers exciting the flexible plate) by secondary point force actuators.

The research conducted by the authors is concerned with the simulation of active noise control within a plate-cavity system by the action of primary vibrational inputs acting on the flexible plate. Comparisons between vibration and acoustic control strategy as well as the influence of sensor and actuator numbers are pursued. Simplicity of the geometric model as well as the understanding of the nature and characteristics of modal behaviour account for the choice of the physical model. Simulated control schemes include the use of vibration control inputs (point, line, surface forces and line moments) acting on the flexible plate and acoustic control acting within the cavity (speakers). The following discussion will be limited to the development of the analytical basis and the comparison of results obtained in a case studied by Pan et al. [5].

## I. Model development

The analytical model used in this study was developed using a modal summation approach similar to the ones used by Nelson et al. [4] and Pan et al. [5]. The model to be considered is shown in Figure 1 and consists of a rigid-walled paralellepipedic cavity having at one face (z=0) a simply supported flexible vibrating plate. Also shown are the elements which are modelled analytically, they are point, line and surface forces, line distributed moments and surface mounted acoustic velocity sources (speakers). By using the modal summation approach, the pressure contribution due to the plate *in vacuo* and cavity modal responses are evaluated for both controlled and uncontrolled conditions. The goal is to obtain an expression for the pressure within the cavity (where control is desired) in terms of primary (known) and control excitations. Minimization of the quadratic pressure is the method used to control the sound field inside the cavity. The analytical development is summarized in the following lines. Initially, the plate's movement equation is obtained in terms of its displacement W and is given by (a list of symbols used is given in Appendix I):

$$\mu(\partial^2 W(x,y)/\partial t^2) + D\nabla^4 W(x,y) = F^{tot}(x,y) \quad (1)$$

where $F^{tot}(x,y) = \sum F^p(x,y) + \sum F^c(x,y) + P^p(x,y,0) + P^c(x,y,0)$

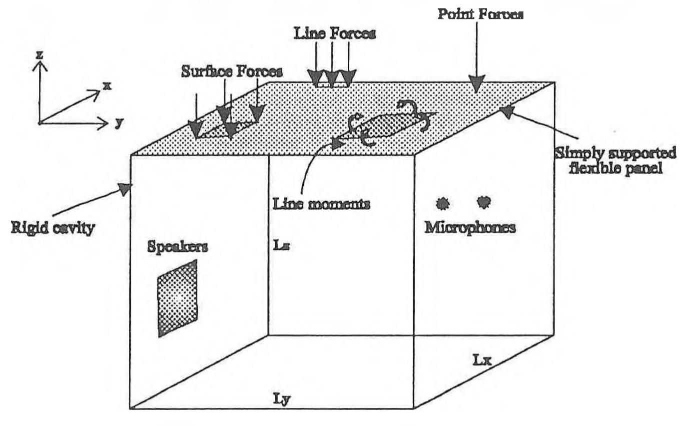


Figure 1. Sketch of plate-cavity system.

Equation (1) is expressed as a function of the total forced excitation (primary and control). The excitation is due to the action of vibration forces and acoustic pressure loading due to the presence of the rigid cavity (the outside pressure field influence is not taken into account since the mechanical excitation will dominate the plate response). If time dependancy of the form $e^{j\Omega t}$ is assumed then equation (1) becomes:

$$-\mu\Omega^2 W(x,y) + D\nabla^4 W(x,y) = F^{tot}(x,y) \quad (2)$$

Expressing the plate displacement in terms of a modal summation basis:

$$W(x,y) = \sum_{m=1}^{\infty}\sum_{n=1}^{\infty} A_{mn} w_{mn}(x,y) \quad (3)$$

where $w_{mn}(x,y) = \sin(m\pi x/Lx)\sin(n\pi y/Ly)$

is the modal shape function for simply supported conditions. Substituting equation (3) in (2), integrating the result over the plate surface and using the orthogonality property of *in vacuo* plate modes we obtain an expression for the coupled plate-cavity system:

$$A_{mn}(\omega_{mn}^2-\Omega^2)(\mu LxLy/4)=(\sum F^p(x,y)+\sum F^c(x,y))w_{mn}(x,y)$$
$$+\rho_o\Omega^2\sum_{m=1}^{\infty}\sum_{n=1}^{\infty}\int_0^{Lx}\int_0^{Ly}(P^p(x,y,0)+P^c(x,y,0))\,dydx \qquad (4)$$

The pressure terms inside equation (4) can be expressed in all points as a function of the plate displacement and a Green function developed for the modelled cavity by using the Kirchoff-Helmoltz Theorem. The pressure is given by:

$$P(M)=-\iint_{S_c}\frac{\partial P(M_d)}{\partial \vec{n}}.G(M,M_d)dS_c-\iint_{S_p}\frac{\partial P(x,y,0)}{\partial \vec{z}}.G(M,(x,y,0))\,dS_p \qquad (5)$$

The first integral corresponds to the contribution of acoustic velocity sources and the partial derivative represents a velocity distribution. The second surface integral corresponds to the contribution of the plate movement and the partial derivative represents the acoustic/mechanic velocity continuity condition on the plate-cavity interface, which is given by the following expression: $\rho_o\Omega^2W(x,y)$. The Green function is developed for a rigid walled paralellepipedic cavity. It's development brings:

$$G(M,M_d)=\sum_{p=0}^{\infty}\sum_{q=0}^{\infty}\sum_{r=0}^{\infty}\frac{\Phi_{pqr}(M).\Phi_{pqr}(M_d)}{\alpha_{pqr}(\omega_{pqr}^2-\Omega^2)} \qquad (6)$$

where $\Phi_{pqr}(x,y,z)=\cos(p\pi x/Lx)\cos(q\pi y/Ly)\cos(r\pi z/Lz)$

is the modal shape function representation for a rigid cavity. The $A_{mn}$ terms can be found by substituting equations (5) and (6) in (4), evaluating the surface integral at the plate-cavity interface and solving the resulting linear system. Using a viscous damping model for both plate and cavity, the pressure expression for an arbitrary point within the cavity space is found to have the following form for the case of surface force excitation and control (no acoustic velocity terms):

$$P(x,y,z)=-\rho_o c_o^2\Omega^2.\frac{LxLy}{\pi^2}\sum_{mn=1}^{\infty}\sum_{pqr=1}^{\infty}\left[\frac{(A_{mn}^p(\omega)+A_{mn}^c(\omega)).\Phi_{pqr}(x,y,z)}{\alpha_{pqr}(\omega_{pqr}^2-\Omega^2+j\eta_c\omega_{pqr}^2)}\right].$$
$$\left(\frac{1}{(m+p)_{odd}}+\frac{1}{(m-p)_{odd}}\right)\left(\frac{1}{(n+q)_{odd}}+\frac{1}{(n-q)_{odd}}\right) \qquad (7)$$

## II. Control scheme and numerical results

Evaluation of the pressure equation at one point composed of the contribution of primary and control excitations can then be performed. Multiplying equation (7) with its complex conjugate form (separated into its primary and control constituents) and integrating the result over the cavity volume yields the average volumetric squared pressure as shown in equation (8):

$$\langle P_{ave}^2\rangle=1/V_c.\iiint_{V_c}(P^p(M).P^p(M)^*+P^p(M).P^c(M)^*+$$
$$P^c(M).P^p(M)^*+P^c(M).P^c(M)^*)dV_c \qquad (8)$$

The expansion and rearrangement of this cost function into a quadratic form (having for unknown the point force actuation amplitude and phase) allows its minimization by using the unique minimum property of quadratic functions. A comparison has been attempted with a case found in Pan et al. [5] where the primary excitation was due to a plane acoustic wave acting perpendicular to the plate surface and where the control was achieved with a centered point force. The present model only accounts for vibration forces, thus the acoustic plane wave is replaced by an equivalent surface force. Qualitative comparison of the results obtained in figure 2 and the literature shows that the two sets of curves agree extremely well. This was somewhat predictable since very similar methods were used. Figure 2 shows the noise reduction achieved (NR is defined as the logarithmic ratio of the average volumetric quadratic pressure within the cavity to the quadratic surface force applied on the plate). The plane surface force and the point force actuator destructively interfere optimally in this particular case where control is applied in the plate center (both primary and control forces excite odd-odd plate modes only).

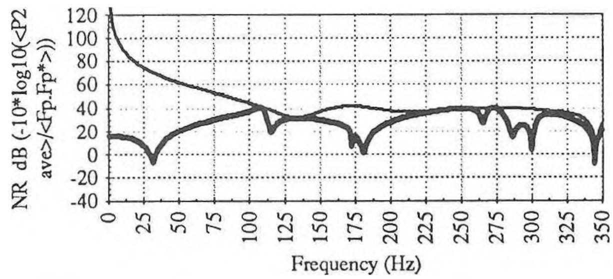


Figure 2. Noise reduction achieved with point force vibration control (___ with control, ____ without control).

As can be observed plate modes dominate the spectrum. Vibration control at the plate is found to be beneficial and the mechanism responsible for the reduction observed is termed "modal suppression". This phenomena is characterized by the reduction in modal vibration amplitudes and consequently the noise radiated. The first plate mode (efficient radiator) is well attenuated at 30 Hz while higher order less efficient radiator odd-odd modes are relatively less controlled. The comparison validates the exactness of the model and will allow further studies to be conducted with the use of multiple point forces for control and excitation. Investigation of control mechanisms and control optimization as well as other developments concerning the acoustic control aspects will be presented in the conference.

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## Appendix I

| | | | |
|---|---|---|---|
| $A_{mn}$ | plate modal amplitude | $\alpha_{pqr}$ | modal coupling coef. |
| c | speed of sound (m/s) | $\eta$ | damping coef. |
| D | flexural rigidity (N.m) | $\mu$ | surface mass (kg/m$^2$) |
| L | plate dimensions (m) | $\rho$ | volume density (kg/m$^3$) |
| F | force (N) | $\omega_{mn}$ | plate nat. angular freq. |
| m,n | plate modal indices | $\omega_{pqr}$ | cavity nat. angular freq. |
| p,q,r | cavity modal indices | $\Omega$ | angular frequency |
| P | int. cavity pressure (Pa) | $\nabla^4$ | 2$^{nd}$ order Laplace oper. |
| S | surface (m$^2$) | $\vec{n}$ | normal vector to surface |
| V | volume (m$^3$) | M | point of evaluation |
| x,y,z | general system coordinates | $M_d$ | disturbance point |
| | **Subscripts** | | **Superscript** |
| c | cavity | p | primary excitation |
| p | plate | c | control excitation |
| o | air characteristics | | |

# AN ACOUSTIC DESIGN METHOD FOR LIGHT WEIGHT STRUCTURES SUPPORTING A VIBRATING SOURCE

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## QUALITY IN ACOUSTIC

Today, acoustics play an important part in the commercial success of new manufactured articles. In fact, acoustic is part of the product quality and the consumers are interested in quality. Furthermore, the consumers sensitivity to quiet products has increased over the past few years because manufacturers have promoted the acoustical quality of their products.

However, there is a general problem with the integration of the acoustic in the design process. The development of a new product often neglects the acoustical aspect at the design stage to consider it only when a noise problem has appeared. The acoustical aspect has to be integrated in the design process by the development of an acoustical design method.

## THE PROBLEM OF A LIGHT WEIGHT STRUCTURE SUPPORTING A VIBRATING SOURCE

Ventilation equipments are typical devices in which the noise level must be controlled to satisfy the acoustical comfort of the consumers. This type of devices generally include a light weight sheet-metal enveloping structure (a box-like structure) on which an electrical motor is mounted with a metal bracket (figure 1).

The acoustic problem with these ventilation devices is a very high noise level at 120 Hz and a variability of the level at this frequency from one device to another. The 120 Hz frequency is due to electromagnetic forces generated by the electrical motor and transmitted to the enveloping structure through the metal bracket.

There is a need to develop a systematic approach to obtain a design that is robust design and minimise the radiation of this type of light weight structure excited by a low frequency source. The design method should meet two objectives:

1) Reduce the noise level at 120 Hz
2) Control the variability at 120 Hz

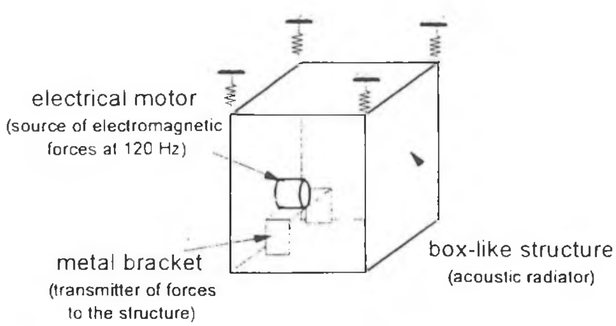


Figure 1: Basic vibro-acoustical model of a ventilation device

## MOTIVATIONS

This research has two motivations. First, a scientific one of understanding the vibroacoustical behaviour of the structure as a function of geometrical and manufacturing parameters. Second, a technological motivation of improving the design approach to the development of quieter products.

## DESIGN PROCESS

The proposed design method starts with the identification of basic design concepts based on acoustical engineering judgement. Then, two complementary approaches are used to analyse the effectiveness of these basic design concepts as well as the effect of various manufacturing and assembly parameters on the noise level. The first approach is to conduct statistically designed experiments to identify key parameters, their effects and their interactions. The second approach is a finite element modelisation including a sensitivity analysis of the quadratic velocity as a function of key parameters.
The design process is illustrated in figure 2.

## BASIC CONCEPTS

The basic concepts are based on the interpretation of the loads transmission between the connecting structure (bracket) and the receiving structure (box-like structure). The loads can be divided in forces and moments.

In the transmission of forces principle, normal forces are reduced by creating a bracket attached on the edges of the box. In this case any force transmitted to the box give rise to two in plane forces. In the transmission of moments principle, moments are reduced by using a very stiff bracket. These two principles are illustrated in figure 3.

Many other cases are possible and were derived and analysed using the same design principles.

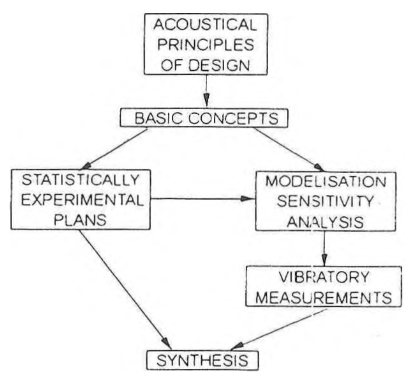


Figure 2: Design process

## EXPERIMENTATION

Seven two-level variables were defined to characterize the response and the variability of the structure (figure 4). A $2^{7-3}$ fractionnary factorial plan was realised with two replications. The sound pressure level in the exhaust duct was the measured response.

The results for the mean analysis are shown in figure 5. It is interesting to note that the motor mount has a big influence on the response and that the elastic motor mount is very effective. This type of mount is effective because it eliminates the moment induced by the electrical motor. It is also interesting to note that the flexible bracket is very effective. At this stage, it is not possible to explain exactly why the stiff bracket is not effective but this will be done in the future with the modelisation. Also, the "at plate junctions bracket" is better than the "conventionel bracket" as it was supposed in the basic concepts. The difference may seem to be small, but this is because a first plan was realised to determine the optimal position of the conventional bracket.

The results for the variability analysis is given in figure 6. The motor mount, motor position and the number of assembling bolts variables have a big effects in comparison with the other factors. The problem is that although these effects are large, it is not possible to assign the variability of the response to any of these variables because the confidence level is lower than 90%. In conclusion of this analysis, the presence of an uncontrolled factor that influences the variability may be suspected.

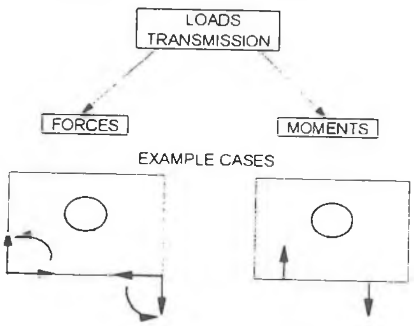


Figure 3: Basic concepts

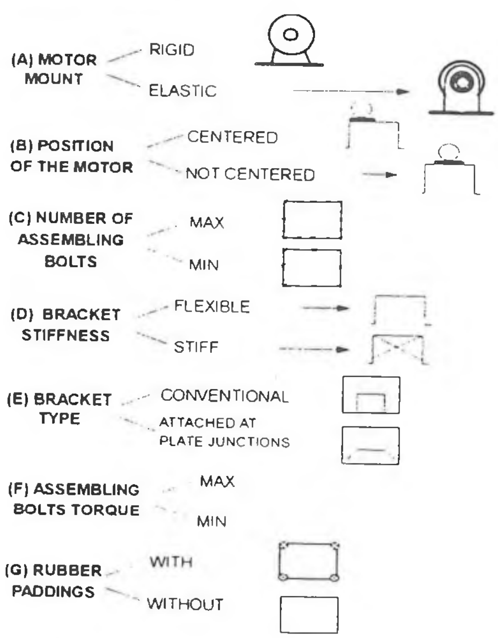


Figure 4: Variables for the experiments

## UNCONTROLLED FACTOR

The structure is very flexible and the assembly is not a precision mechanical one. Then, variable internal stresses and deformations may be induced by assembly. The problem is that it is not possible to make exactly the same configuration twice. This fact implies a shift of the natural frequencies for two identical setups made at two different times. This little shift in frequency induces a very high difference in amplitude.

## MODELISATION

In the future, a finite element modelisation will be realised to understand of the results of the experimentation. Also, a sensitivity analysis of the quadratic velocity as a function of various parameters will be made

## CONCLUSION

For the noise reduction objective, an analysis of the mean and the determination of the main effects lead to the conclusion that the motor mount variable is very important. These results will have to be verified with the modelisation in the future.

For the variability reduction objective, an analysis of the variance was performed and lead to the uncontrolled factor hypothesis. In the future, one way to better assess the effect of each factor would be to perform the same analysis with a new design of the structure in which the uncontrolled factor is eliminated.

| FACTOR | A motor mount | B motor position | C # bolts | D bracket stiffness | E bracket type | F torque | G rubber paddings |
|---|---|---|---|---|---|---|---|
| EFFECT (db) | 10.40 | -5.30 | -0.71 | 7.02 | 1.78 | -5.30 | 4.00 |
| SIGN. (%) | 99 | 99 | <90 | 95 | 95 | 99 | 99 |
| MEAN (db) | 76.02 | | | | | | |

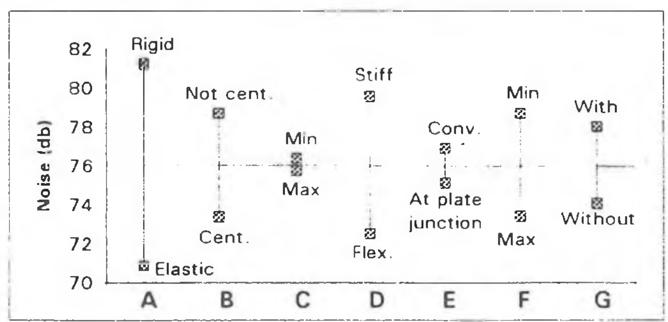


Figure 5: Main effects on the mean

| FACTOR | A motor mount | B motor position | C # bolts | D bracket stiffness | E bracket type | F torque | G rubber paddings |
|---|---|---|---|---|---|---|---|
| EFFECT (db) | -0.96 | -1.08 | -0.73 | -0.10 | 0.35 | 0.29 | 0.05 |
| SIGN.(%) | <90 | <90 | <90 | <90 | <90 | <90 | <90 |

Figure 6: Main effects on the variance

# NOISE MANAGEMENT AT HYDRO-QUÉBEC CONSTRUCTION SITES

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## 1. INTRODUCTION

Noise generated by construction site activities is likely to become a major source of nuisance if inadequately managed. This prompts the need for criteria to control the nuisance resulting from such activities. Prescribed controls must take into account the specific nature of construction noise, i.e., temporary and fluctuating. Hydro-Québec conducted an exhaustive review of the statutes, regulations, criteria and procedures used by different agencies and organizations to manage this type of noise. That review resulted in the 1993 publication of guidelines for noise management at Hydro-Québec construction sites *(Guide relatif à la gestion du bruit des chantiers de construction d'Hydro-Québec).*

## 2. NOISE CRITERIA

Two types of criteria are used to manage noise at construction sites. The first refers to individual items of equipment likely to generate noise. It prescribes the permissible noise level at a given distance from the apparatus, generally 15 metres, when only that one apparatus is in operation. This criterion varies with the item of equipment involved.

The second type of criterion sets a maximum noise level for construction site activities as a whole, at one or more specific spots. This approach has the advantage of reckoning with the worksite environment, as the criteria are established in light of milieu sensitivity.

Having weighed both options, Hydro-Québec decided to base its noise criteria (NC) on the noise emanating from the site as a whole, given that the magnitude of the noise impact of a construction site is determined by all the noises heard in the sensitive areas of the site, not the noise originating from individual items of equipment.

Noise criteria were thus established in terms of host milieux, i.e., the types of areas in which the noises are heard. Table 1 gives permissible maximum noise levels. Note that noise emanating from trucks operating off-site is not subject to the levels prescribed in this table. It was deemed more appropriate to establish maximum noise levels for different categories of trucks (see Table 2).

**Table 1: Recommended Noise Criteria for Construction Sites**

| Type of host milieu | Permissible daytime level (dBA) | Permissible nighttime level (dBA) |
|---|---|---|
| Institutional | 65 | 50 (≤1month)<br>45 (>1month) |
| Inhabited | 75 (≤1month)<br>70 (1 to 6 months)<br>65 (> 6 months) | 50 (≤1month)<br>45 (>1month) |
| Commercial | 75 | 60 |
| Industrial | 80 | 80 |
| Vacant | No NC | No NC |

**Table 2: Recommended Noise Criteria for Heavy Trucks**

| Truck horsepower (HP) | Measuring distance (m) | Permissible noise level |
|---|---|---|
| under 275 | 15 | 84 |
| 275 and over | 15 | 87 |

## 3. NOISE MEASUREMENT

The guidelines mentioned earlier describe the procedure for evaluating construction site noise.

Needless to say, meteorological conditions, e.g., temperature, humidity, wind velocity and direction, soil type and cloud cover, must be determined and logged. It is unadvisable to make measurements when the temperature is below -10°C or the wind velocity above 6 m/s.

### 3.1 Measuring Construction Site Noise

The procedure for measuring construction site noise is derived from Standard SAE J1075 MAR87 "Sound Measurement-Construction Site." However, that standard was modified somewhat for our purposes. First, the measuring points are to be located in areas most suceptible to site noise. Second, the measuring process shall last a minimum of 30 consecutive minutes (Leq 30 min) and shall be scheduled to include the times of greatest impact.

### 3.2 Measuring Truck Noise

In situations that require measuring the noise generated by trucks, Standard SAE J1096 FEB87 "Measurement of Exterior Sound Levels for Heavy Trucks under Stationary Conditions" shall be used.

### 3.3 Measuring Noise from Trucking Activities

Most impact-assessment studies for construction sites require measurement of the noise from trucks on the surrounding roads. Although no criterion has been prescribed, this is necessary to determine and, where appropriate, minimize the impact of truck traffic. The recommended measuring procedure is derived from the document "Sound Procedures for Measuring Highway Noise," prepared for the US Federal Highway Administration.

The recommended minimum sampling time is 10 minutes, and the vehicles are counted by category for the duration of the procedure

## 4. NOISE IMPACT-ASSESSMENT STUDY

Hydro-Québec's guidelines describe the stages of a noise impact-assessment study for construction sites, as well as some of the sampling tools to be used.

The stages are as follows:
- Verify the need for the study
- Gather the relevant information
- Establish applicable noise criteria
- Measure background noise
- Forecast noise levels
- Validate project conformity and impact
- Propose mitigative measures
- Design monitoring program

Project impact is determined by the gain in noise levels together with the duration of the work. It is evaluated by means of tables 3 and 4.

**Table 3: Gain in Noise Levels**

| Gain in noise level (dBA) | Description of gain |
|---|---|
| Less than 0 | Negative |
| 0 to 3 | Insignificant |
| 3 to less than 5 | Small |
| 5 to less than 10 | Medium |
| 10 to less than 15 | Strong |
| 15 and over | Very strong |

**Table 4: Impact Evaluation**

| Descrip-tion of gain | Duration | | |
|---|---|---|---|
| | 1 month or less | 1 to 6 months | more than 6 months |
| Negative | Positive | Positive | Positive |
| Insigni-ficant | Insigni-ficant | Insigni-ficant | Insigni-ficant |
| Small | Insigni-ficant | Small | Small |
| Medium | Small | Small | Medium |
| Strong | Small | Medium | Strong |
| Very strong | Medium | Strong | Very strong |

## 5. CONCLUSION

Hydro-Québec's guidelines for the management of construction site noise provide a standardized approach to the problem of noise pollution and a means of clearly determining the associated impacts.

# Noise Control of Coal Pulverizers
# An Implementation of User Friendly Design

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**Process Description**

Five of Ontario Hydro's generating stations are fuelled by coal. In order to extract maximum energy from this fuel, while minimizing emissions, the coal must be pulverized before it enters the boiler combustion zone. At our largest fossil station each boiler is served by five pulverizers which grind coal to fine dust which is then transported to the boiler burners, entrained in high pressure air flow.

These vertically oriented coal mills have 14 ft. diameter cylindrical housings fabricated of heavy steel plate. The housing is supported on a cylindrical plate steel base such that the bottom plate of the housing is about 7 ft. above grade level.

The grinding elements of the pulverizers can be considered as giant ball bearings. Raw coal is crushed between ten 30 in. diameter balls and a stationary upper grinding ring, and a rotating lower grinding ring. The lower ring is driven by a 500 hp motor through a robust right angle gearbox located within the housing support base. A pressurized labyrinth seal at the drive shaft entry prevents blow out of coal dust and pressurized heated primary air from inside the housing.

**Noise Sources**

The brute force required to break down the bituminous coal results in high vibration levels throughout the pulverizer, and consequential emission of high level noise. The grinding noise is broad band in character and distributed over frequencies below about 1 kHz. The gearbox is also a source of lower frequency noise, but at a lower level than that of the process noise.

In addition to that from the grinding process, significant noise originates from the seal assemblies. This emission, comprising both broad band and tonal components, is generated by high pressure air leaking through the seal gap. Because there is inevitably some eccentricity in the seal fit, the tonal components are modulated at the 37 rpm mill drive speed. Depending on the seal eccentricity and air pressure, the tonal peaks may be found in any of the octave bands from 1 kHz to 16 kHz.

**Noise Exposure**

Mills must be kept under close surveillance to maintain their adjustment and detect potential failures such as oil leaks before they can cause a forced outage. One member of the crew operating each pair of generating units is responsible for inspection and adjustment of ten mills several times per shift. Operators are careful to wear hearing protection during their surveillance tours as noise levels in the vicinity of the mills can exceed 100 dBA. A typical surveillance visit could require the operator to spend up to 2 minutes inspecting the gearbox and motor. On some rounds, a more extended visit is required as the operator increases the load on the grinding elements by stroking a lever actuated hydraulic pump at a panel adjacent to the mill.

**Project Scope**

As part of a pilot project to demonstrate the practicability of retrofit of noise controls to power station equipment, a package of controls was developed and implemented on a single mill.

Because high level, lower frequency noise induced by the grinding process was beyond control by passive means, effort was concentrated on controlling the annoying seal air squeal.

The challenge in retrofit installations is to design cost effective controls which do not interfere with surveillance and maintenance, and which do not compromise the performance and durability of the equipment being controlled. Review of all recommendations by a project team comprising participants from design engineering, construction, operations, and maintenance, insured that optimal controls would be implemented.

**Design Requirements**

The following design requirements were agreed to by the project team:

- reduce noise emission by 8 dBA or more
- protect operators from high level sound even when under the mill housing
- not obstruct visual observation of gearbox
- not result in area under housing being designated a confined space
- minimize additional heat load under housing
- not restrict access to gearbox for housekeeping, inspection, maintenance or removal

Extensive noise surveys identified and quantified the contributing sources and propagation paths. Seal air leakage at the shaft entry into the housing was the dominant source under the mill housing and also propagated into the surrounding area through 3 large openings in the housing base. Because the operator is required to carefully inspect the gear box during each visit, it was necessary to control the noise as close as possible to the source, rather than simply contain it within the area under the housing.

**Noise Control Design**

The first stage of control is a cylindrical acoustic shroud (Photo 1.) and associated baffle plate surrounding the bottom seal area; replacing the existing coupling safety guard. This is 1.5 in. thick, fabricated in appropriately sized segments to facilitate installation and removal, and incorporating hatches to provide access for measuring and setting the coupling and seal.

Acoustic panels lining the support base interior ceiling and much of the walls absorb seal air and gearbox noise and also reduce the heat load under the housing which is of benefit both to the operators and to gearbox durability.

The 40 ft.$^2$ opening at the front of the support base is closed with an acoustic panel (Photo 2.) incorporating double doors hung from heavy duty cam-rise hinges. Detents hold the doors open at 120° so that operator access to clean up leaked oil and absorbent is unrestricted. Vision panels in each door and internal illumination facilitate gearbox surveillance from outside.

Two openings, each of 9 ft.$^2$, at the back corners of the gearbox provide access to the hold-down bolts which require frequent retightening. One of these openings is closed off by a removable steel panel with a ventilation opening fitted with a simple acoustic trap.

Heated air and fumes are extracted from the housing base by a fan attached to the mill housing and joined to the other opening by an acoustically lined duct. Both side walls of the duct are hung from vertical hinges. Access to the hold-down bolts is accomplished by releasing a set of draw latches to fully open the duct (Photo 3.). Visual inspection of the gearbox and bolts is accommodated by a vision panel in the outside duct wall.

Replacement air is provided by an additional acoustically controlled opening located low on the housing support base.

### Results

As expected, implementation of the controls is effective primarily in controlling noise above about 500 Hz. The spectra presented in Fig. 1 display the noise measured 1 m. from the front of the mill, with and without controls. At this location the overall noise reduction is about 13 dBA. At a location just inside the support base, the reduction is about 6 dBA.

Operator feedback regarding the trial installation has been positive and there has been no significant increase in gearbox temperature. Similar controls are now being fabricated for the 4 other mills on the Unit. Noise reductions on some of these may exceed that achieved on the trial unit because the seal air is a more significant contributor to their overall emission.

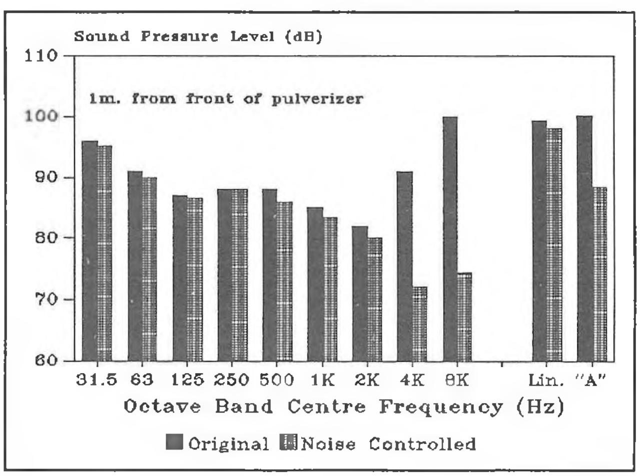


Fig. 1. Pulverizer Noise Reduction

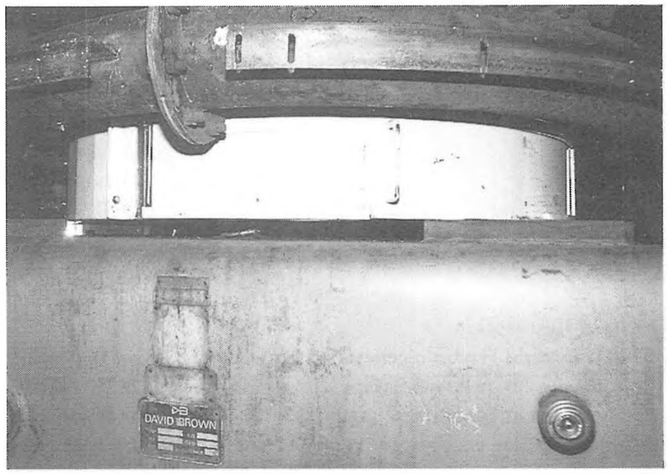


Photo 1. Acoustic Shroud around Coupling


Photo 2. Front Closure Panel - Gearbox visible through door.

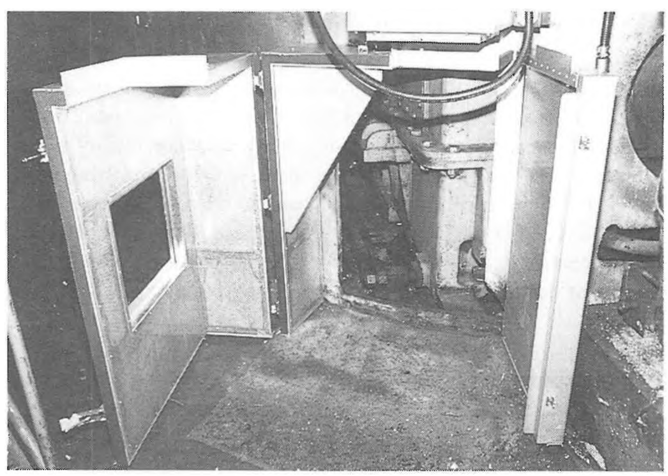

Photo 3. Lined Extraction Duct - Fully opened to access gearbox bolts.

# Errors Incurred Using a Dummy Head to Make Omni-directional Room Acoustics Measurements

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## INTRODUCTION

Most auditorium acoustics measurements require the use of an omni-directional microphone; others are binaural and require the use of a dummy head. This paper presents some aspects of investigations using a dummy head to make omni-directional measurements. Data from three different concert halls, using both techniques, were used to examine the differences between the two. The halls used were Mechanics Hall in Worcester, Massachusetts, Massey Hall in Toronto, and two configurations of the John Aird Centre Recital Hall in Waterloo, Ontario.

## EXPERIMENTAL METHOD

The omni-directional microphone impulses were obtained using RAMSoft II [1], while the binaural impulses were obtained using BRAM (Binaural Room Acoustics Measurement software). Both are computer based measurement systems developed at the National Research Council of Canada. Both produce impulse responses for a maximum length sequence signal, using a Fast Hadamard transform process. The integrated impulses from the dummy were added on a simple energy basis. The RAMSoft II software calculates 12 acoustical quantities including: reverberation time, RT (-5 to -30 dB), early decay time, EDT (0 to -10 dB), clarity-early/late energy ratio, C80 (80 ms early time period), and relative level, G (level re. free field level at 10 m). All the measures were obtained in the six octaves from 125 to 4000 Hz. The omni-directional microphone was a half-inch B&K microphone and the dummy head was a B&K type 4128 head and torso simulator with internal microphones.

## RESULTS

Consider first the differences between hall average measurement results from the dummy head and the omni-directional microphone system. The average differences, dummy head - omni, for EDT are plotted versus octave band frequencies for the three halls and are shown in Figure 1. The differences tend to be small (within 0.1 s) which could be due to errors in accurately re-positioning the source and receiver. Bradley [2] showed that moving the receiver by only 30 cm can induce a variation of over 0.1 s at low frequencies and 0.05 s at high frequencies. Since the two measurements were made at different times and with different receivers, significant variations in the positions are quite plausible. At the higher frequencies, where the width of the head becomes a significant fraction of the wavelength, the directionality of the head will become a factor and thus contribute to the differences. Another problem is that the dummy head measurements averaged two positions that are 15 cm apart, i.e. the spacing of the ears. This will also contribute to the observed differences. On a seat-by-seat basis, the differences are larger. Differences of up to 0.4 s at mid frequencies and 0.8 s at low frequencies were found at individual seat locations in one hall.

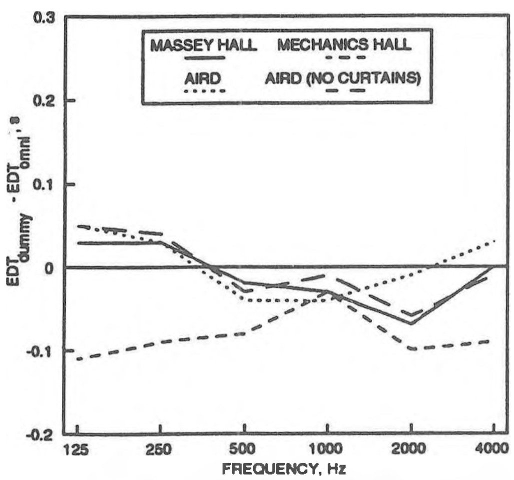


*Figure 1. Average differences over entire hall of $EDT_{dummy}$ - $EDT_{omni}$.*

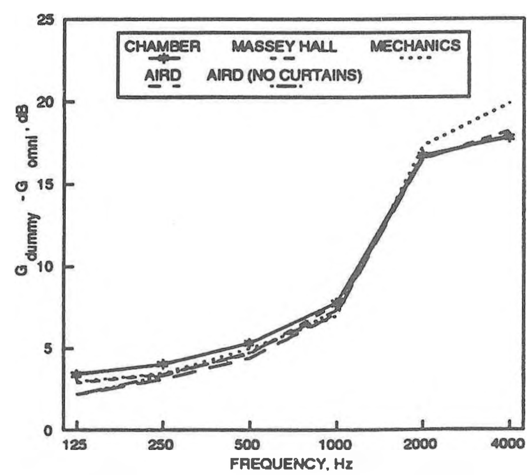


*Figure 2. Average differences over entire hall of $G_{dummy}$ - $G_{omni}$.*

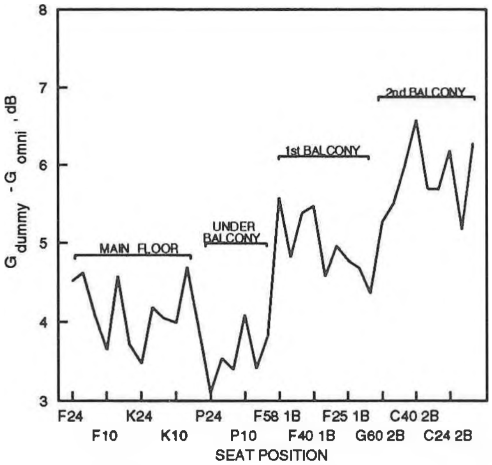


*Figure 3. Seat-by-seat differences of G for Massey Hall at 500 Hz.*

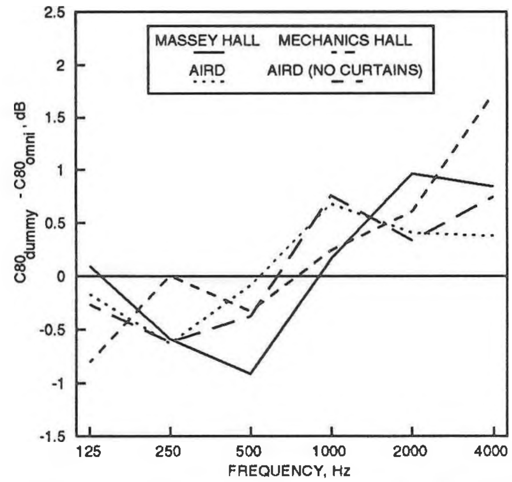


*Figure 4. Average differences over entire hall of C80 dummy -$C80_{omni}$.*

A graph of the average differences of G versus frequency is shown in Figure 2, and the effect of the directionality of the head is quite apparent. Along with the hall differences, the measured differences of the dummy head and the omni-directional microphone in a 250 $m^3$ reverberation chamber at the National Research Council are shown. In the 125 Hz octave, the difference is about 3 dB, which corresponds to a doubling of energy. The energies of the two ears were added together, and therefore at lower frequencies where the head is less directional, the energy should be twice that of the omni-directional microphone. At higher frequencies, the directionality of the head is a larger factor and is not a simple doubling of energy. The differences from the hall data results follow the form of the reverberation chamber measurements. The differences range from 3 dB in the low octave to between 15 and 20 dB in the 4000 Hz octave. There is also a noticeable spread of differences between the halls, about 1 dB in the lower octaves and up to 2 dB in the highest octave.

A plot of the differences in G versus seat position for the 500 Hz octave for Massey Hall is shown in Figure 3. The curve is shifted above zero due to the average directionality effects of the dummy head at 500 Hz, shown in Figure 2. The major variations are shown to be related to specific areas in the hall that would change the significance of the dummy head directionality. The dummy head would be more sensitive to sound arriving from the side, thus these locations might make these more prominent. These areas are under the balcony and in the balconies where the receiver is significantly above the source. On a seat-by-seat basis, Figure 3 shows that these differences are as large as 3 dB.

The differences of C80 versus frequency for the two measuring techniques are shown in Figure 4. At the low frequencies, the differences are in accordance with the expected re-positioning errors [2] which are about 1 dB for source/receiver differences of 30 cm. At the higher frequencies, differences of about 0.5 dB would be expected [2], but once again the directionality of the head will also play a role. The seat-by-seat differences are large - from 1 dB to over 7 dB at 4000 Hz in one hall. These differences are probably due to directionality effects of the head but it is not clear that this is the only effect.

**CONCLUSIONS**

The directionality of the dummy head plays a very important role and causes differences in all of the above measured quantities. On an average basis for an entire hall, the differences are not very large, but are significant. It may be possible to apply an average correction to account for some of the systematic differences. This is obvious with G values (as seen in Figure 2) but for the other measures it will require further work. However, the individual differences can be much larger. At some locations these differences would be greater than typical differences between halls. These differences can vary as much as 3 dB, as seen in Figure 3. Therefore, the two techniques cannot be used to give precisely the same results.

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# Objective Comparisons of Massey Hall and Boston Symphony Hall

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## Introduction

Massey Hall (MH) and Boston Symphony Hall (BSH) are of about the same age with generally similar acoustical properties, but with a number of distinct differences. This paper summarises comparisons of extensive objective acoustical measurements in each hall. These comparisons are intended to demonstrate how various newer auditorium acoustics measures can be used to better understand the acoustical properties of large halls and to more reliably diagnose acoustical problems in halls.

BSH is a long, rectangular or "shoe-box" shaped hall with a volume of 18,746 $m^3$, a width of 23 m, and with seating for 2631 people. MH has an approximately square floor plan with a volume of 14,190 $m^3$, a width of 34 m, and with seating for 2500 people. Both halls were measured in a completely unoccupied condition and therefore the differences in the acoustical properties of the seats in the two halls will significantly influence the comparisons. BSH has seats that are leather covered while MH has more absorbent cloth-covered seats on the main floor and 1st balcony seats. The second balcony seats in MH are molded plywood.

In both halls, measurements were made using our RAMSoft-II computer based measurement system that calculates 12 different quantities in 6 octave bands from impulses in a few seconds while at each location in the hall. Responses were obtained for the combinations of three source positions and 12 receiver positions in MH, and 14 receiver positions in BSH (i.e. 36 measurement combinations in MH and 42 in BSH). Quantities measured included: reverberation times (RT), early decay time (EDT), early/late sound ratio (C80), lateral energy fraction (LF5), and relative sound level (G). In addition, the relative level of the early arriving sound energy, G80, the late arriving sound energy, G(late), and the early lateral energy, GEL, were also calculated.

## Hall Average Results

Averages of all measurements in each hall can be used to compare the mean properties of the two halls in their unoccupied conditions. (Differences could be quite different when occupied.) The two halls had quite similar average RT, EDT, and G values at low and mid-frequencies (125 to 1000 Hz). At higher frequencies (2000 and 4000 Hz), RT, EDT, and G values were lower in MH, probably due to the more absorptive seats. C80 values indicated that there would be higher clarity in MH. The LF5 values were generally higher in MH, thus providing an increased sense of envelopment and spaciousness.

## Within Hall Variations of Measures

There are many ways of looking at the variation of acoustical measures within halls. Examining these variations helps one to better understand the acoustical properties of a hall. Figure 1 compares plots of 2000 Hz G values versus source-receiver distance. At closer seats, 10 m from the source,

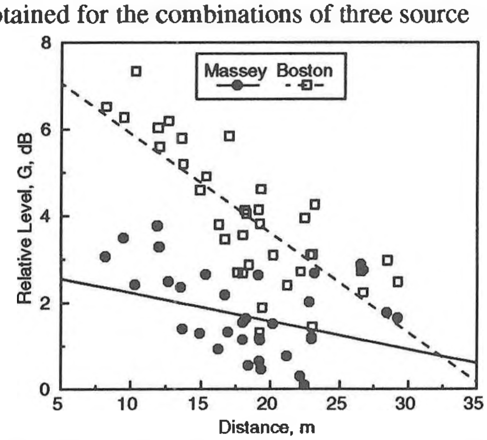


*Figure 1: G values at 2000 Hz versus source-receiver distance in both halls.*

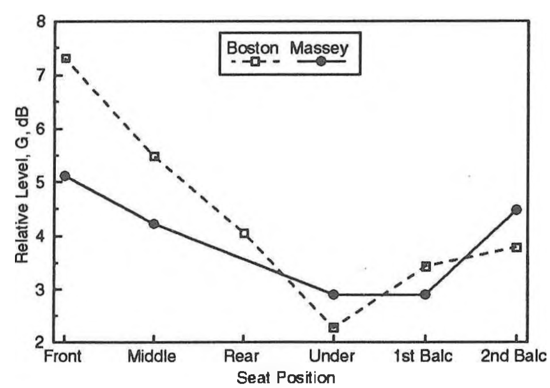


*Figure 2: Position average 1000 Hz G values versus position in the hall.*

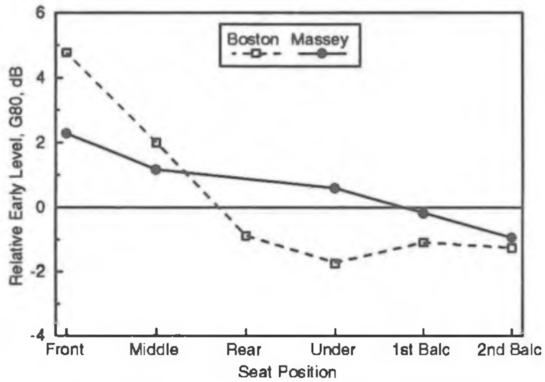


*Figure 3: Position average 1000 Hz G80 values versus position in the hall.*

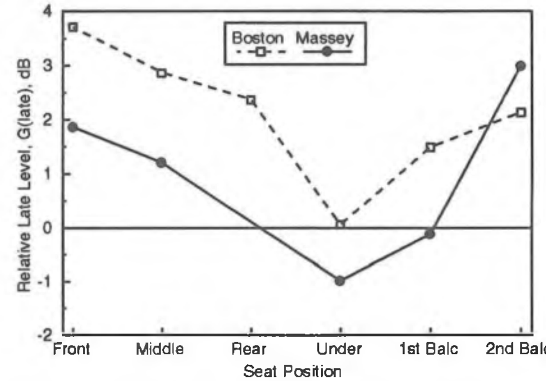


*Figure 4: Position average 1000 Hz G(late) values versus position in the hall.*

levels in BSH are approximately 4 dB higher than at the same distance in MH. However, in BSH levels decrease more rapidly with distance. Because BSH is much longer than MH, this decreasing level trend leads to similar levels at the rear of BSH and MH. There is considerable scatter about these trends that increases at lower frequencies.

To eliminate the scatter and to relate variations to the geometry of the halls, average values of quantities were calculated for combinations of seat groups and all three source positions. Thus, these results were the average of measurements for the combinations of three source positions and two or three receiver positions. Figure 2 plots 1000 Hz average G values versus position in each hall. Because MH is shorter, there were no "Rear" positions in MH and the distances to each seating group would be different in each hall. These results again show higher levels in BSH close to the source that decrease more rapidly towards the rear of the hall. These average levels tend to be lowest at seats under the balcony. In MH, levels were also low in the 1st balcony at seats under the second balcony.

Figures 3 and 4 compare the area average values of 1000 Hz G80 and G(late) versus position in the halls. In BSH, early sound levels, G80, are higher closer to the source but are lower than in MH towards the rear of the hall and at seats in the balconies. The late sound energy, G(late), shown in Figure 4 was higher in BSH than in MH at all locations except in the second balcony.

It is thought that the higher levels close to the source are due to the narrower width of BSH and also to the shape of the orchestra enclosure that directs reflected sound to the closer seats in the hall. The wider MH has a more even distribution of early energy (from this position average data), but a generally lower level of late arriving sound energy.

While sound levels seemed more reduced at seats under the balcony in BSH, C80 and EDT values were more changed at under-balcony seats in MH. In MH, 1000 Hz EDT values were less than other seating areas by 0.4 seconds or more. Clarity, as measured by C80, was highest at under-balcony seats in MH.

The position average results hide the observed variation of sound levels across the width of MH. Each position shown on Figures 2,3, and 4 was an average of the results obtained for the combinations of three source positions and two receiver positions. One of these receiver positions was closer to the centre line of the room and the other was closer to the side of the room. Sound levels decreased by 1 to 3 dB from the centre to the side seats in MH. This effect contributed to the generally larger spatial variation of measured results in MH.

**Conclusions**

Massey Hall and Boston Symphony hall have similar average unoccupied RT, EDT and G values at lower and mid-frequencies. The two different plans of these halls causes differences in the spatial variation of acoustical quantities. In the wider MH, levels varied more across the hall. In the longer and narrower BSH, levels varied more from the front to the rear of the hall. At seats under the balcony in MH, early sound and early lateral sound were stronger than in BSH.

# SUBJECTIVE COMPARISONS OF MASSEY HALL AND BOSTON SYMPHONY HALL

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## 1. Introduction

Massey Hall and Boston Symphony Hall date from the same period but are quite different architecturally. Whereas Boston Symphony Hall is a classic narrow shoebox, Massey Hall is shaped more like a square. One might expect that, given these architectural differences, the two halls should sound significantly different. In this paper we describe a detailed subjective comparison of the two halls and correlate the results with objective measurements within the halls.

## 2. Experimental Procedure

Sound fields were produced by convolving anechoic music (Mozart, "Le Nozze di Figaro") with measured binaural impulse responses taken in the two halls [1]. The resultant binaural sound fields were played back to listeners over a pair of loudspeakers with appropriate steps taken to eliminate cross-talk. The validity of the system for such subjective testing was demonstrated in an earlier study [2] and a detailed description of the system can be found in [3].

The subjective testing was in the form of double-blind paired comparison tests. A computer provided random playback of pairs of sound fields and subjects could switch back and forth between the two sound fields until they had made their decision. Subjects conducted eight sets of tests and were asked to rank the two halls in terms of *reverberance*, *clarity*, *loudness*, *spaciousness*, *treble*, *bass*, *apparent source width*, and *overall preference*. This list of parameters is based on the one used by Barron in his subjective study of British concert halls [4]. Prior to each set of tests, a description of the parameter under test was read to the subject. Also, to ensure that they fully understood their task, subjects were given a brief training sequence before each test. All of the subjects used in the study had previous experience in critical listening tests and most had extensive musical training.

Eight sound fields were used in the study. These were produced from binaural impulse responses taken at four seats in each hall. The seats were chosen to represent a reasonable cross-section of the acoustical characteristics of each hall.

## 3. Results

*Loudness*

Figure 1 shows the perceived loudness of the various sound fields versus measured G values. A higher value of loudness implies a subjectively louder sound field. The values of G consist of an average value of the 500Hz and 1kHz octave bands. It is not well understood how to add G values across frequency and therefore an average mid-frequency value seems like a reasonable first guess. There is very good correlation between the subjective and objective results. Also, the two halls are similar in both their measured and perceived loudness. It is interesting to note that the subjects were able to accurately resolve very small differences in G.

*Clarity*

The perceived level of clarity is plotted against measured values of C80 in Figure 2. Again these values are derived by taking the average of the mid-frequency octaves. There is a fairly strong correlation between the subjective and objective measures of clarity. There is however one seat where the objective measure does not correctly predict the subjectively perceived level of clarity. The point on the graph corresponding to this seat has been circled. This seat was located on the main floor (Massey) under the balcony. Inspection of the impulse response for this seat revealed two very strong and distinct reflections arriving at about 27msec and 47msec after the direct sound. It may be that these reflections are in some way reducing the perceived clarity at this seat, although further investigation is certainly necessary before any conclusions can be drawn. We see also that the two halls share a similar range of C80 values and that neither hall dominates in perceived clarity.

*Reverberance*

Figure 3 shows the subjectively perceived level of reverberance versus the average of the mid-frequency EDT's. Again, there is reasonable agreement between the subjective and objective results. It can be seen that Massey Hall has a much wider range of EDT values than Boston Symphony Hall. Mid-frequency EDT's in Massey Hall vary by as much as a second.

*Treble*

The perceived level of the treble frequencies is plotted against the average high frequency (2kHz and 4kHz) values of G in Figure 4. The results show that the high-frequency values of G act as a good predictor of the amount of treble perceived by a listener. It is interesting to note that all of the sound fields measured in Boston Symphony Hall have greater high-frequency G values than any of the sound fields measured in Massey Hall. This is the only measured acoustical parameter for which this is true.

## 4. Conclusions

We have compared subjective and objective measures of several acoustical parameters in two concert halls. The results indicate that the objective measures are able to predict subjective opinion reasonably well. Also, except for the high-frequency G values, the two halls are quite similar in their measured acoustical performance. Results for the remainder of the acoustical parameters, including overall preference, will be presented at the annual meeting.

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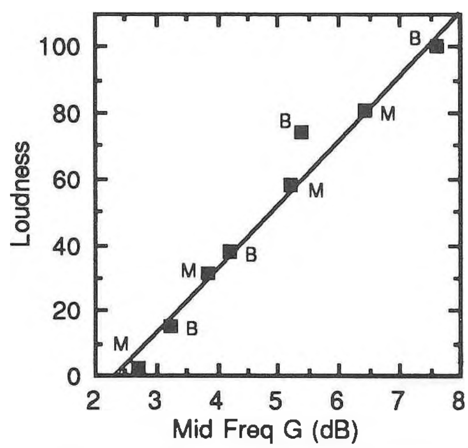


Fig. 1 Perceived loudness vs. mid-frequency G. M-Massey Hall, B-Boston Symphony Hall

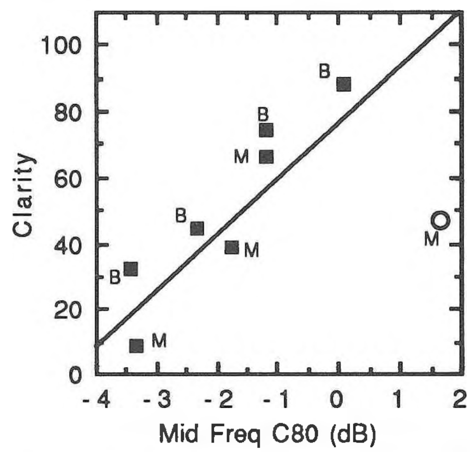


Fig. 2 Perceived clarity vs. mid-frequency C80. M-Massey Hall, B-Boston Symphony Hall

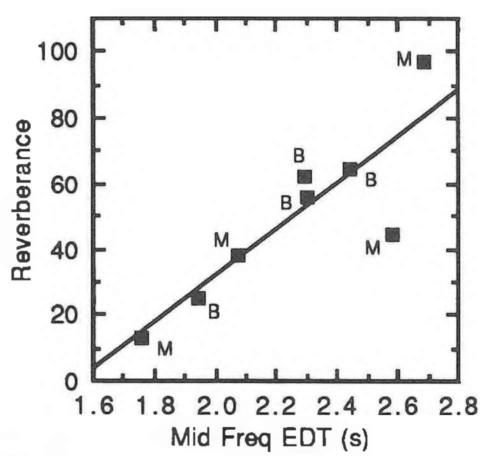


Fig. 3 Perceived reverberance vs. mid-frequency EDT.

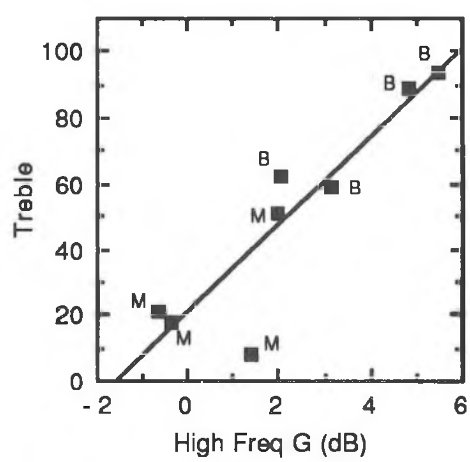


Fig. 4 Perceived treble vs. high-frequency G.

# An Apparatus for Measuring the Dynamic Stiffness of Acoustical Materials

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## 1. Introduction

A common practice to improve both the impact-noise isolation and the airborne sound transmission of a structural floor is to add a floating floor. It consists of a hard rigid floating slab supported on a resilient layer that dynamically decouples it from the structural slab. The impact-noise isolation of certain types of floating floors have been investigated by Cremer[1] and Ver[2]. One of the parameters that determines the performance of these floors is the dynamic stiffness of the resilient material used. A draft international standard, ISO/DIS 9052-1, exists that specifies a procedure for measuring the dynamic stiffness of the resilient material. This draft standard allows measurements to be made either with a shaker or an impact hammer.

In this report, a description of the test methods will be presented together with a comparison of the results obtained by both methods.

## 2. Test set-up and measurement procedure

The procedure specified in the draft standard applies to the determination of dynamic stiffness per unit area of resilient materials with smooth surfaces used in a continuous layer under floating floors. A specimen of dimensions 200 mm x 200 mm was placed between two horizontal surfaces, i.e. the baseplate and the load plate, as shown in Fig. 1. The load plate was square, with dimensions 200 mm x 200 mm, and was made of steel. The combined weight of the load plate, the accelerometer, and the force transducer was 8.1 kg. The baseplate was a 100 mm thick marble slab with dimensions 700 mm x 950 mm. It weighed about 150 kg. The baseplate rested on three resilient mounts. The system consisting only of the marble slab and the resilient mounts had a resonance frequency of about 7 Hz which would be below that of the system formed by the load plate and the test specimen.

Figure 1 also shows the other equipment used in the shaker test. The driving signal for the shaker was a 4095 point m-sequence generated by the computer at a clock frequency of 1250 Hz. The exciting force was measured by a force transducer and the response of the system was measured by an accelerometer. Both the force transducer and the accelerometer signals were low-pass filtered at 315 Hz with a dual channel anti-aliasing filters before they were digitized and processed. Since the driving signal is deterministic, synchronous averaging is possible. In this measurement, ten averages have been used. Individual signal was first cross-correlated with the m-sequence via the fast Hadamard transform[3] before they were converted into the frequency domain by FFT. The system frequency response was subsequently determined as the ratio of the cross-spectral density between the motion response and the excitation force divided by the auto-spectral density of the excitation force. Since the motion response was measured by an accelerometer instead of a displacement transducer,

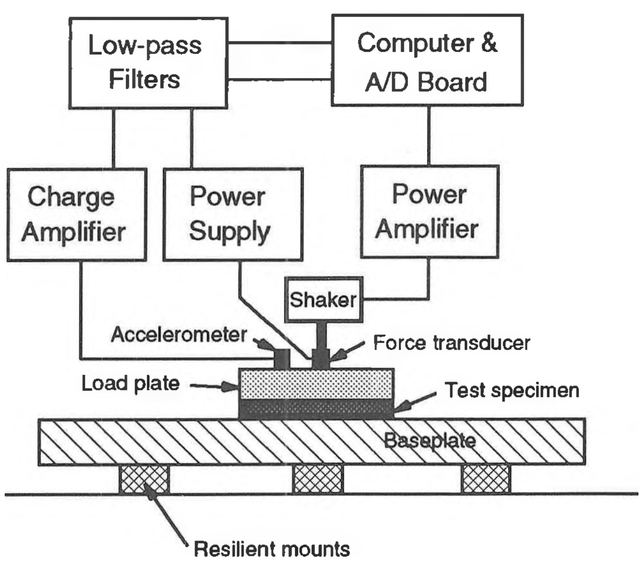



*Fig. 1: Diagram of equipment used in the shaker test.*

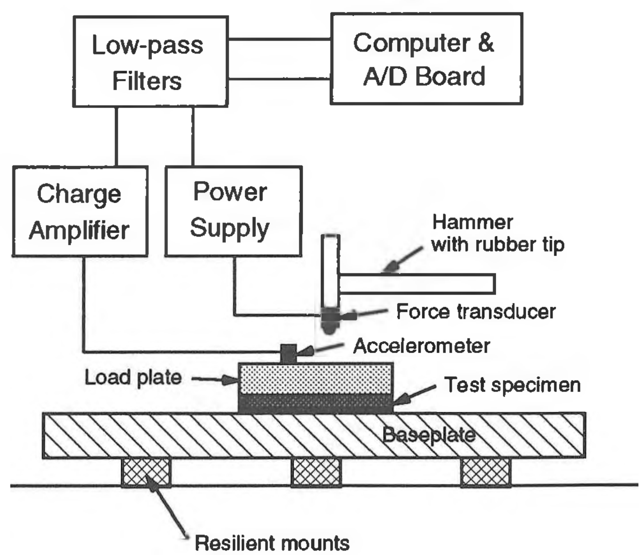
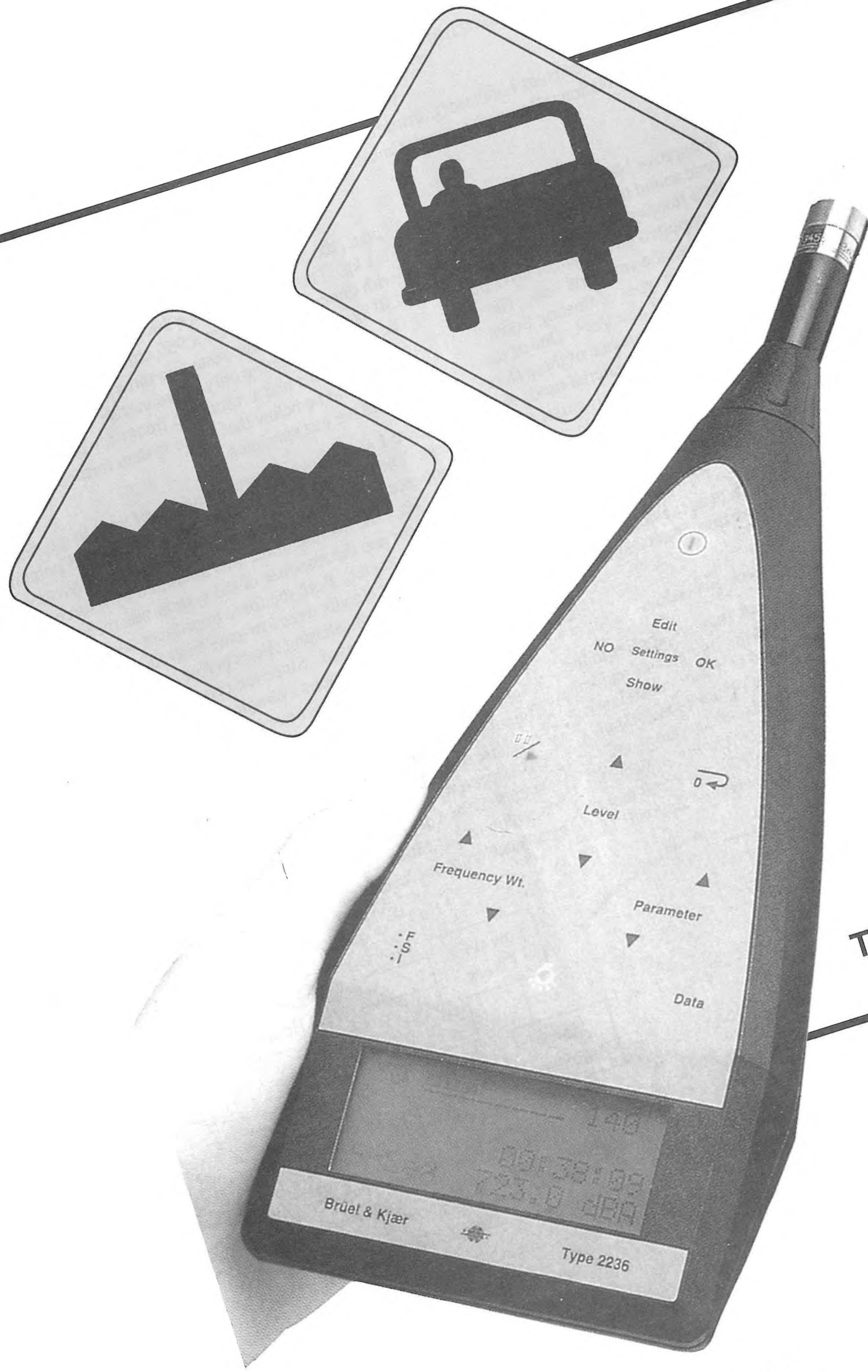



*Fig. 2: Diagram of equipment used in the hammer test.*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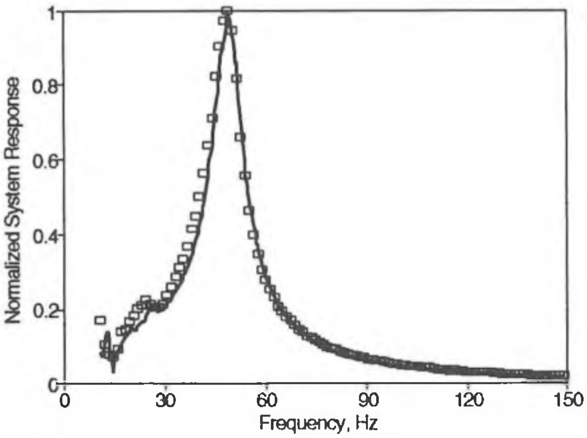


*Fig. 3: Comparison of the system frequency responses measured by the two methods: line - shaker method; open square - hammer method.*

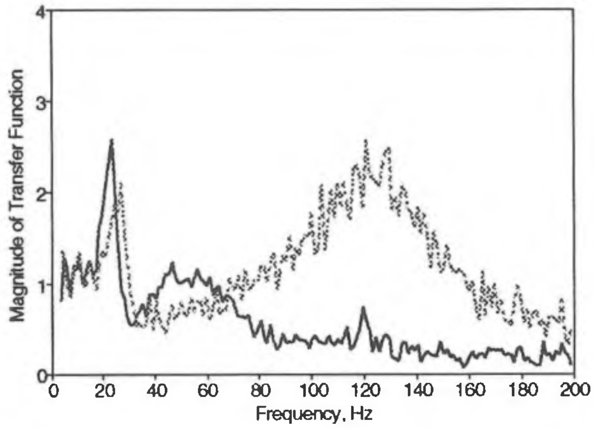


*Fig 4: Comparison of the magnitudes of the transfer functions between the vertical acceleration of the baseplate and that of the load plate coupled by resilient channels from two different suppliers.*

an additional division by the square of frequency was applied. The dynamic stiffness of the material is given by the resonance frequency of the fundamental vertical vibration of the spring and mass system.

Figure 2 shows a similar set-up when measurements were made using an instrumented impact hammer. The sampling frequency used was 1250 Hz. A direct 1024 point FFT was applied to both the force transducer and the accelerometer signals. Similar data analysis was used to obtain the system frequency response.

## 3. Results and Discussion

The sample tested consisted of two layers of 10 mm thick rubber pads. Figure 3 compares the system frequency responses measured by the shaker method and the hammer method without showing the resonance frequency of the baseplate assembly. The response curves have been normalized by the magnitude at resonance. Although the hammer test consisted of a single run, the results show good agreement with those obtained by the other method. Because of its simplicity and ease of operation, the impact method seems to be a better procedure to use.

## 4. Other Applications

Recent studies of the sound transmission loss of party walls of identical configuration using resilient channels supplied by different manufacturers show significant differences in results. An attempt has been made to see if the methods presented here can be applied to study the transmissibility of the resilient channels used in the wall assemblies. As a preliminary investigation, the test specimen shown in Fig. 2 was replaced by two pieces of resilient channels screwed to the bottom of the load plate near two opposite edges. The assembly was attached to the baseplate using double sided adhesive tape. Another accelerometer was used to measure the acceleration of the baseplate. The hammer was used to impact the baseplate and the accelerations of the baseplate and the load plate were measured simultaneously. A transfer function between the acceleration of the load plate and that of the baseplate was calculated. Figure 4 shows significant differences in the magnitudes of the transfer functions obtained for the resilient channels supplied by two different manufacturers. Although no quantitative results can be derived from these preliminary data, the results shown in Fig. 4 seemed to correlate well qualitatively with the performance of the party wall employing the two different types of resilient channels. The one characterized by the solid curve of Fig. 4 gave the party wall a better transmission loss performance. A more detailed study of this application is currently in progress.

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# Reverberation Time?

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### 1.0 Introduction

Reverberation Time ($RT_{60}$) defines the rate of decay of sound energy in a room which in turn is directly related to the overall sound absorption in the room. $RT_{60}$ is the transient response (the statistical time domain analysis) of the room which is assumed to be convertible to the steady state decay (the statistical frequency response) of sound in the room with distance. The concept is simple and suitable as a descriptor for room acoustics. The detailed evaluation and measurement of $RT_{60}$ however is full of tricks and traps. The problems associated with the evaluation of $RT_{60}$ are described first. The difficulties associated with the field measurement of $RT_{60}$ are highlighted and the methods usually applied to overcome these are also discussed. Several methods and instrumentation systems for reverberation time measurements are compared and a cost effective procedure is presented.

### 2.0 Theoretical Evaluation

Attempts have been made in the recent past, with limited success, to calculate $RT_{60}$ by detailed evaluation of the sound decay from a specific source location using methods such as ray acoustics, finite element schemes etc. However, the concept of $RT_{60}$ is more commonly derived from the theoretical model of a room with statistically uniform sound energy density. The room is assumed to be large enough, box like in shape and the surfaces treated relatively uniformly. The sound field is then diffuse when a sound source is turned on and decays uniformly when the source is turned off. Sabine [1] derived a simple formula to calculate the reverberation time which is:

$$RT_{60} = 0.161 \cdot \frac{V}{\alpha S} \tag{1}$$

where V is the volume of the room in cu.m., S is the total room surface area in sq.m. and $\alpha$ is the average absorption coefficient. Norris-Eyring [1] modified Sabine's formula to account for extreme cases of absorption:

$$RT_{60} = 0.161 \; \frac{V}{-S.\ln(1-\alpha)} \tag{2}$$

Real life situations of course are never that simple. Two problems are immediately obvious in the above formulations: the proper absorption coefficients of material used in the space [2] and the actual distribution of the absorptive material in the room. Even if one assumes that the correct values of absorption coefficients are known, the actual distribution of the material in the room can make a lie of the $RT_{60}$ evaluations. Some attempts have been made by methods such as Fitzroy's [3], where the separate calculations for 3 pairs of opposite surfaces are combined together to provide one value of $RT_{60}$. The Fitzroy formula is:

$$RT_{60} = 0.161 \frac{V}{S} \left[ \frac{S_v}{\alpha_v} + \frac{S_t}{\alpha_t} + \frac{S_l}{\alpha_l} \right] \tag{3}$$

where $S_v$ and $\alpha_v$ represent the area of a pair of opposite surfaces and the absorption coefficient and similarly for the other two surfaces. A significant difference in the $RT_{60}$ values from Eqs. (1) and (3) probably is an indicator that neither equation provides the correct answer, since the measured values are likely to be dependant on the source - receiver location combinations.

Other field conditions such as dominant echoes, flutter echoes or slow decay of vibration induced on lightly damped surfaces tend to produce measured $RT_{60}$ values that are larger than the theoretically evaluated values. The theoretical under estimation of $RT_{60}$ values is particularly acute in relatively soft rooms.

Similarly, problems are also encountered where spaces are either too small or the aspect ratios, the ratios of length to width to height, are large. In small rooms, statistical energy analysis is applicable only if sufficient number of room modes are present in the frequency bandwidth of interest. Most measurement standards consider the magic number of 10 (ten room modes) to be suitable. The problem is equally complex in large aspect ratio rooms especially if some of the surfaces are highly absorptive. In the extreme cases one can think of open plan offices with highly absorptive surfaces or manufacturing plants with acoustic roof decks. For these rooms, steady state measurements (in the frequency domain) have often indicated decay of sound larger than 6 dB per doubling of distance from the source. There is no equivalence in $RT_{60}$.

In spite of all the problems highlighted above, no other simple enough descriptor is available at present to replace $RT_{60}$. One still talks of "a concert hall with $RT_{60}$ of 1.8 secs," as if true for all frequencies and true for all source-receiver combinations in the hall. A number of other design indices such as AI (Articulation Index), STI (Speech Transmission Index) and a whole list of design curves for acoustically sensitive spaces are based on $RT_{60}$. Some day $RT_{60}$ may be replaced but not until theoretical calculation methods, measurement techniques and design indices are readily available for use.

### 3.0 Measurement Methods of $RT_{60}$

The traditional method of exciting the room is with an impulse source, gunshot or otherwise, and a recording of the decay of the sound pressure level (SPL). Alternatively, a loudspeaker providing a stationary random noise source is cut-off and the decay recorded. More recently the source has also been a pseudo-random noise (maximum length sequence) played through the loudspeaker. In the last case the cross-correlation between input and response is calculated from which the decay curve can be calculated. Fourthly, a source of sound power can be initiated and the SPL rise curve recorded. This method is rarely used and will not be further considered.

The time for 60 dB of decay can be calculated from the SPL/time curve. A portion of the curve is used because usually 60 dB of decay is not available, the first 5-10 dB is usually not straight and the curve will often contain two slopes. Lately the initial portion of the curve, the first 5 to 15 dB, is often used because of its importance in assessing room acoustics. The "real" $RT_{60}$ can only be determined as accurately as the measured slope of the curve. An alternate method, developed by Schroeder [4], reverse integrates the SPL/time curve from a suitable point above the background to T=0. This provides a more accurate slope.

### 3.1 Comparison of Methods

The impulse noise method is the least expensive and can be used with the simplest measuring instruments (Sound Level Meter + Level Recorder). A very short learning curve and very portable instrumentation make this easy for casual measurements. However, the source, even a gun shot, is non-repeatable and often provides low energy in the low frequencies resulting in a low signal to noise ratio. Often only the first few dB of the decay curve are generated at the lowest frequencies. Many measurements are usually required for averaging and for octave bands. The shock wave from the impulse source requires measuring at a distance. The lowest measurable $RT_{60}$ is determined by sound energy in the impulse, filter ring and instrument response.

The use of a random noise source and a large loud speaker provides more energy in the low frequencies and can provide a larger length of decay curve at all frequencies. The same measuring instrumentation is used. However, for low frequency measurements large and/or multiple loud speakers and amplifiers are required. This has traditionally been the most reliable method.

Both of the first two methods can be used in conjunction with microprocessor based equipment such as Rion NA-29 or Larson Davis 820. These expedite the process by simultaneous and software driven calculations of $RT_{60}$ in all octave or 1/3 octave bands.

Maximum length sequence methods are very attractive in that a single measurement can yield a large amount of other data in addition to $RT_{60}$. The $RT_{60}$ can be calculated from any portion or length of the SPL/time curve and digital filters allow calculations for any octave or 1/3 octave band from the single test. Multiple test for averaging purposes can be easily conducted. However, the learning curve is quite long. The method requires a portable computer and the software is either proprietary (eg. MLSSA), and expensive or the software must be developed (very expensive). As with the random noise source large or multiple speakers and amplifiers are required. The capability and flexibility will appeal to the serious user willing to bear the cost and complexity.

### 3.2 Measurement Results

A large room (15000 $m^3$ ) with average amount of absorption characteristics was used as a test room. The room had the following surfaces: concrete floor; metal siding walls with perforated facing and fibreglass batts; and metal roof deck. The three procedures applied to measure $RT_{60}$ were: (a) Pink noise source with RION NA-29 Sound level meter; (b) MLSSA source, B&K 2215 sound level meter and MLSSA software; and (c) Pink noise source with B & K 2230 sound level meter and B & K 2306 graphic level recorder.

The $RT_{60}$ results are the average of measurements at four to five microphone locations in the test room. The results are shown in Table 1. It is seen that the three methods produce comparable results and one can therefore conclude that any one of the three methods could be used for the test room.

The variability of the $RT_{60}$ with source and microphone location combinations is indicated in the results of Table 2. Procedure (a) was used to obtain the reverberation results. This large (30 x 20 x 25 m) room had absorptive material uniformly distributed over four of the six surfaces and contained several large diffusing elements. Since it satisfies the assumptions of Eq. (1) reasonably well, it is all the more remarkable that the reverberation time showed such a range.

### 3.3 Examples of Measurement Difficulties

There are a number of difficulties associated with reverberation time measurements in the field. Some of them have been mentioned earlier, but only two of the unusual difficulties will be discussed here. The two problems are associated with a highly absorbent small room such as a studio control room. The problems are due to overdriving an acoustically dead room with a loud source and due to the filter characteristics of the measuring equipment.

The steady pink noise source generally used in the measurements is set to produce relatively high levels of sound so that the signal-to-noise ratio is high. The high levels would usually excite various components in the room such as thin panels, air diffusers etc. The vibrations of these components could be sustained for a longer time than the actual reverberation time of the room. Such a scenario is common when air diffuser rings in a room with a low reverberation time. It is easy to overlook the secondary excitations since the resulting sound would be well below the overall steady state sound level. An example was a small dead room with design reverberation of 0.1 sec or less across the spectrum. The measured $RT_{60}$ at 125 Hz was 0.19 sec and was 0.13 sec at 500 Hz. When damping material was added to the air diffuser to reduce the secondary vibrations, $RT_{60}$ reduced to 0.14 sec and 0.09 sec at the two frequencies respectively. If secondary excitations are suspected, source generated by methods such as MLSSA may be better, since the room sound pressure levels could be kept low.

A second problem in a small room with $RT_{60}$ of 0.1 sec or less is due to the transient decay rate of the octave or 1/3 octave band filter of the sound level meter. The filter decay rate for 60 dB of decay is usually more than 0.2 sec. The rise time of the filters is much smaller and the trick then is to use the rise time of the signal instead of the decay. However, the sudden 'switching on' of a speaker is impossible. The same is true for a gun shot. An alternative method is to reverse the recorded signal so that the decay signal appears as a rise signal when connected to the filters. The reverberation time (values as low as 0.02 sec) can thus be evaluated.

## 4.0 Conclusions

The general concept of reverberation time and the difficulties associated with its evaluation (both theoretical and measurements) were discussed in this paper. The different measurement methods commonly used in the field were described and compared.

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**Table 1. $RT_{60}$ Results in secs**

| | $RT_{60}$, secs in Octave Band | |
|---|---|---|
| Measurement Method | 125 Hz Band | 1000 Hz Band |
| MLSSA | 1.21 | 1.51 |
| PINK Noise & RION | 1.43 | 1.68 |
| PINK Noise & SLM + Chart | 1.44 | 1.65 |

**Table 2. Variability in $RT_{60}$ Results**

| | $RT_{60}$, secs in Octave Band | |
|---|---|---|
| Measurement Location | 125 Hz Band | 1000 Hz Band |
| 1 | 1.54 | 1.40 |
| 2 | 1.52 | 1.50 |
| 3 | 1.50 | 1.92 |
| 4 | 1.32 | 1.67 |
| 5 | 1.26 | 1.89 |

# High Temperature Effects on the Insertion Loss of Absorptive Duct Silencers

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## 1 INTRODUCTION

Conventional passive rectangular duct silencers, which are in widespread use in commercial and industrial applications, employ a bulk absorptive material such as fibreglass, mineral wool or foam to attenuate noise. The majority of test data and calculation schemes [1], [2], [3], used to predict silencer attenuation consider only the acoustic performance at or near room temperature. While such predictions are useful for silencers used in heating and ventilating duct systems, many industrial applications require an estimate of the acoustical performance of the silencer at high temperatures. The use of absorptive silencers to control noise from gas turbine generators is an example of an application in which the extreme operating temperatures (250 to 700 °C) can have a significant effect upon silencer insertion loss (IL) performance.

## 2 THEORETICAL BACKGROUND

The IL of an absorptive silencer is affected by two temperature dependant phenomena. First, the flow resistivity of the porous material changes with temperature. This occurs because the resistance to gas flow through the pores of the fibrous material varies with the dynamic viscosity of the gas in the pores (which is usually primarily air). The dynamic viscosity is temperature dependant. Furthermore, the speed of sound in the gas and the gas density vary with temperature. This means that the ratios of silencer dimensions to wavelength, $d/\lambda$ and $2h/\lambda$, (refer to Figure 1) change, as does the ratio of the acoustic resistance of the porous fill to the characteristic acoustic impedance of the gas, $Rd/\rho c$.

The change in gas properties with temperature can be modelled using well established thermodynamic/fluid-dynamic relationships:

$$c = c_0(T/T_0)^{1/2} \tag{1}$$

$$\rho = \rho_0(T_0/T) \tag{2}$$

$$\mu = \mu_0(T/T_0)^{0.7} \tag{3}$$

where, $c$, $\rho$, $\mu$ are the speed of sound, density and viscosity of the gas, respectively and T is the absolute temperature. The '0' subscript denotes normal room temperature values.

The flow resistivity, R, of the porous silencer fill varies with the dynamic viscosity [4], [5]. Hence,

$$\begin{aligned} R &= R_0(\mu/\mu_0) \\ &= R_0(T/T_0)^{0.7} \end{aligned} \tag{4}$$

For the present study, silencer IL predictions were made using a finite element code based on a cubic Galerkin formulation [6]. The algorithm inherently addresses the changes in $d/\lambda$, $2h/\lambda$ and $Rd/\rho c$, since $\rho$, c, R, d and 2h are entered as parameters into the program at runtime. The algorithm calculates the characteristic acoustic impedance, $Z_C$ and the complex propagation constant, $k_C$, using regression equations by Mechel [5], at any number of desired frequencies. For the purposes of the present analysis, IL predictions were made for six silencer geometries at the standard 24 one-third octave band centre frequencies ranging from 50 Hz to 10 kHz, for each of five operating temperatures.

## 3 RESULTS AND DISCUSSION

Figures 2 and 3 show the variation in IL with temperature for two of the six silencer geometries modelled. Several trends are readily apparent.

First, the frequency at which peak attenuation occurs increases with increasing temperature. This effect is accompanied by a corresponding decrease in attenuation in the lower frequencies as temperature rises. Additionally the bandwidth of the attenuation peak changes as T varies, although a definite trend is not obvious. Table I presents a summary of the frequencies at which peak attenuation occurs and the maximum insertion loss per unit length achieved at this peak.

The observed changes in silencer IL spectrum are consistent with an intuitive examination of the changes in fluid properties and flow resistance quantified in equations (1) through (4). As temperature increases, the wavelength, $\lambda$ for any given frequency, f, also increases (i.e., since $\lambda = c/f$). Thus, the critical silencer dimensions, d and 2h are, in effect, becoming smaller with respect to wavelength as temperature increases. The frequency of peak attenuation depends on d and 2h, so it is no surprising that longer wavelengths result in higher peak IL frequencies.

The reduction in low frequency attenuation observed with increasing T, also can be explained in part by the fact that the frequency of peak attenuation is shifting upward. As well, the increased resistivity of the porous fill, R, at higher T contributes to decreased low frequency attenuation. This occurs because an increase in R results in an increase in the magnitude of the reactive part of the characteristic impedance ($X_C = \mathrm{Im}\{Z_C\}$) of the fill material at low frequencies [5], [7]; greater reactivity implies a more reflective (i.e., less absorptive) fill.

## 4 CONCLUSION

Based on the six absorptive silencer models analyzed thus far, it is apparent that an increase in temperature causes an increase in the frequency at which peak attenuation occurs, and a decrease in low frequency attenuation. The results of this preliminary study suggest that scaling factors can be derived through a regression analysis to predict the high or low temperature performance of an absorptive silencer knowing its room temperature performance and the anticipated operating temperature. An investigation to this end is ongoing, in order to generate sufficient data to develop such scaling factors.

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**Table I**

| | Temperature °C | | | | |
|---|---|---|---|---|---|
| | -100 | 20 | 120 | 250 | 500 |
| Silencer #1: d = 0.06 m, 2h = 0.183 m, R0 = 20000 Rayl/m | | | | | |
| $f_{PEAK}$ [Hz] | 800 | 1250 | 1600 | 2500 | 3150 |
| $IL_{PEAK}$ [dB/m] | 47 | 36 | 34 | 35 | 35 |
| Silencer #2: d = 0.299 m, 2h = .152 m, R0 = 15000 Rayl/m | | | | | |
| $f_{PEAK}$ [Hz] | 800 | 1250 | 2000 | 2500 | 3150 |
| $IL_{PEAK}$ [dB/m] | 39 | 41 | 41 | 42 | 42 |

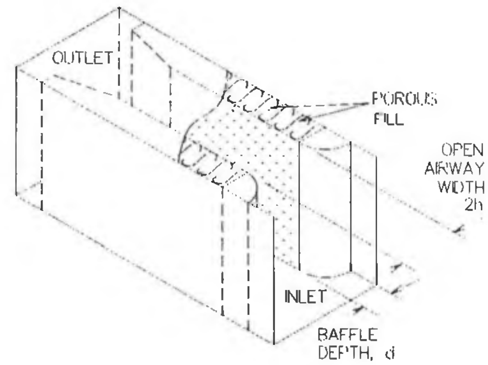


**Figure 1**

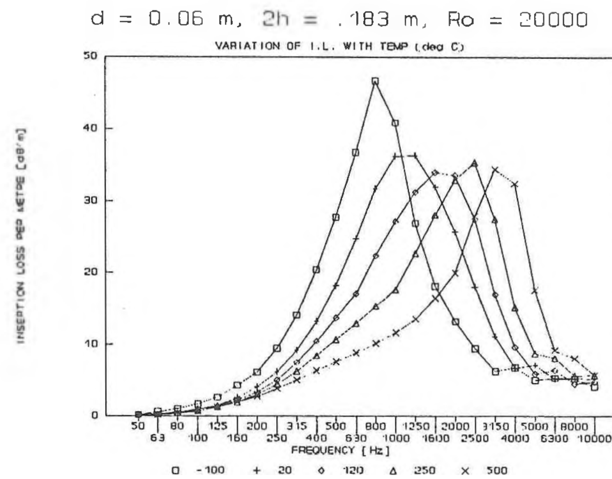


**Figure 2**

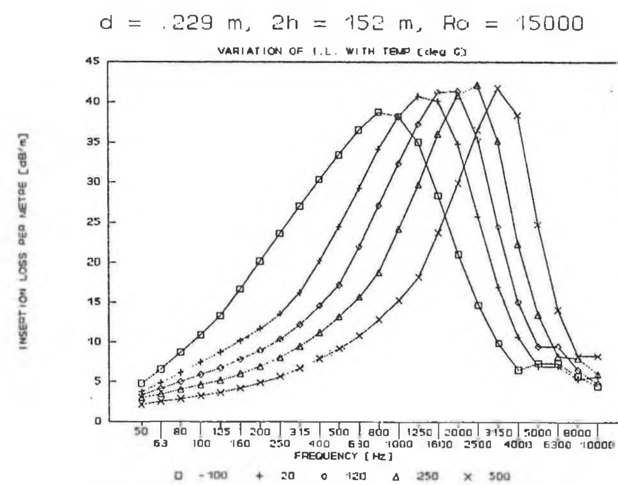


**Figure 3**

# Musical influences on the perception of time

by

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The rate at which time appears to pass is not uniform, but may vary depending on a number of factors. Time perception may be influenced both by the attentional state of the perceiver and by the characteristics of stimuli to which a person is attending (Fraisse, 1963; Ornstein, 1969). Several researchers have suggested that time perception is most generally related to the amount of processing activity (Michon, 1972; Ornstein, 1969). Others have attempted to develop a theory of time perception based on the concept of an internal clock (Kristofferson, 1980; Luce, 1972).

This paper concerns the perception of time during music listening. Although many studies of musical time have focussed on sensitivity to rhythm and meter (e.g., Longuet-Higgins & Lee, 1982), few have examined the ability of listeners to judge duration over several bars. In a study by Clarke and Krumhansl (1990), listeners were presented excerpts from pieces by Stockhausen (Experiment 3) and by Mozart (Experiment 6). The excerpts all differed in duration, but had an average duration of about 30 seconds. Listeners were asked to judge the duration of each excerpt, and were also asked to rate each excerpt on a number of subjective scales, such as complexity, variedness, and completeness. Regression analyses suggested that judgements of duration were highly veridical, and were apparently not influenced by the subjective characteristics of the excerpts.

Our investigation was an extension of the research reported by Clarke and Krumhansl (1990). For purposes of comparison, we used the same music used by Clarke and Krumhansl (1990, Experiment 6) -- Mozart's fantasie in C minor, K. 475. Our experimental method, however, differed from that used by Clarke and Krumhansl. First, listeners in our study were not asked to judge excerpts of differing durations. Thus, we removed that source of variation from duration judgements. Second, the listeners in our study were explicitly instructed to estimate *subjective* time, rather than veridical time. Third, each excerpt was presented at each of three tempi. Finally, both objective and subjective characteristics of excerpts were used as predictors of perceived duration. There were two parts to the study. In Part 1, listeners judged the duration of excerpts. In Part 2, other listeners provided subjective judgements of excerpts.

## Method

**Subjects**. Sixteen musically-trained adult listeners participated in Part 1. Ten different trained listeners participated in Part 2. Listeners had a minimum of five years of musical training, and all reported normal hearing.

**Musical materials**. Eight musical excerpts from Mozart's fantasie in C minor, K. 475 were used. The starting locations of these excerpts were: bar 12, bar 28, bar 42, bar 84, bar 99, bar 135, bar 160, and bar 171. Each of the eight excerpts was presented at each of three tempi: slow, medium, and fast. The medium tempo for each excerpt was equal to the tempo markings indicated in the score. The fast tempo was 20 bpm faster than the medium tempo, and the slow tempo was 20 bpm slower than the medium tempo.

**Apparatus**. Musical excerpts were entered into an SE/30 Macintosh computer using Professional Composer software, and saved as MIDI files. MIDI files were then used with Experimental software created by A. McKinnell. The order of presentations was randomly and independently determined for each listener. Excerpts were output using the piano 1 timbre of the Roland U-20, and played through Sennheisser headphones.

**Procedure**. Part 1: First, a standard duration was established by presenting a black rectangle on the computer screen for 10 seconds. Listeners were instructed that, for each presentation, they would hear a drum sound shortly after the music started. Their task was to press any key on the Roland U-20 keyboard when a period of time had elapsed, beyond the drum sound, that was equivalent to the standard duration. The musical presentation stopped as soon as a response was made. Listeners were instructed to judge subjective time, rather than attempt to make veridical judgements. Practice trials were provided to help acquaint listeners with the task.

Part 2: Ten second excerpts were created from the excerpts used in Part 1. Listeners were presented these 10 sec excerpts, and rated them on five subjective qualities, using scales from 1-7. The qualities were: closure, information, varied, pleasing, and expectancy. Ratings of 7 for each quality indicated that the excerpt had closure, was high in information, was highly varied, was very pleasing, and ended with an implication that an important musical event was imminent. Practice trials were again provided.

## Results and Discussion

For each presentation, the amount of time between the drum sound and each listener's key-press response was recorded. These values ranged from 6.39 seconds to 27.51 seconds, and averaged 13.03 seconds across all listeners and excerpts. A response 10 seconds after the drum sound would indicate veridical time perception. Thus, it is apparent that responses were nonveridical and that the musical excerpts generally had the effect of decreasing the perceived amount of time elapsed.

The 24 time values for the 16 listeners were then entered into an analysis of variance, with repeated measures on two factors -- Excerpt (eight levels) and Tempo (three levels). A significant main effect of Excerpt indicated that the perception of duration was dependent on the excerpt presented, $F(7, 105) = 4.59$, $p < .001$. There was no main effect of Tempo. However, a significant interaction between Excerpt and Tempo suggested that the judged duration of an excerpt was not always consistent across the three tempi, $F(14, 210) = 2.34$, $p < .01$.

## Subjective ratings of excerpts

Mean ratings for the five sets of subjective ratings were then used as predictors of mean duration judgements in a regression analysis. For this analysis, we used data only from those listeners whose duration judgements on any trial were within two standard deviations of the mean. This procedure ensured that mean duration judgements were not distorted by data that reflected lapses in concentration. Using this elimination procedure, mean duration judgements were calculated from 10 listeners.

Correlations between mean duration judgements and mean ratings for subjective qualities were generally low -- the highest r values were associated with information ($r = -.35$) and pleasing ($r = -.38$). The negative correlations indicate that the greater the pleasingness and perceived amount of information, the sooner listeners thought the standard duration had elapsed. Using multiple regression, a reasonably good model of duration judgements included three predictors: information ($b = -.74$, $p < .05$), varied ($b = .68$, $p < .05$), and pleasing ($b = -.58$, $p = .08$), with a multiple $R = .65$, $p < .02$.

The relationship between information and perceived duration is compatible with Ornstein's (1969) storage-size hypothesis, if one assumes that information and complexity are similar constructs (Berlyne, 1971). Ornstein argues that the perceived duration of an event depends on the complexity of that event. Events that are high in complexity require more storage space in memory. This increase in storage requirement is thought to lengthen perceived duration.

It is notable that the relationship between information and pleasingness is thought to be nonlinear (Berlyne, 1971). Quite possibly, some of the subjective predictors also are nonlinearly related to judgements of duration. Thus, predictions of judged duration may be improved by considering nonlinear regression models.

## Objective descriptions of excerpts

As a further analysis, 14 objective descriptions of each excerpt were obtained. The descriptions included: tempo, pitch dispersion (i.e., standard deviation), total number of notes, mean duration, mean pitch, number of key changes, and mean note density. Correlations between mean duration judgements and these objective descriptions were again generally low -- the highest r values were associated with pitch dispersion ($r = .49$), mean pitch ($r = -.35$) and mean note density ($r = .32$). Using multiple regression, a good model of duration judgements based on objective descriptions included two predictors: pitch dispersion ($b = .14$, $p < .01$) and mean note density ($b = .45$, $p < .05$). This model had a multiple $R = .63$, $p < .01$. This finding suggests that as pitch dispersion and note density increased, listeners waited longer before they responded that the standard duration had elapsed.

Finally, both mean subjective ratings and objective descriptions of excerpts were used in another multiple regression analysis. In this case, the most successful model included three predictors and had a multiple $R = .74$, $p < .001$. These predictors were: pitch dispersion ($b = .11$, $p < .01$), total number of notes ($b = .008$, $p < .05$) and perceived amount of information ($b = -.98$, $p < .005$). To simplify, the model suggests that a musical event will be perceived to be longer if the pitch dispersion is small, there are fewer notes, but the perceived amount of information is high. However, it is important to note that this model accounts for only 55% of the variance in duration judgements.

The perception of duration in music is not veridical, but is influenced by characteristics of the music. Factors such as the perceived amount of information and pitch dispersion appear to interact with the process of perceiving time. Although a complete understanding of these influences is not possible from the present analyses, the methodology used provides a useful alternative to that employed by Clarke and Krumhansl (1990). The process of unraveling the many possible influences on duration judgements presents a challenge for future research.

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# Perception of musical tonality as assessed by the probe-tone method.

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## Introduction

The purpose of the experimental research reported in this presentation was twofold: (1) to assess the reliability of the probe-tone technique as a method of recovering listeners' representations of the tonal hierarchy; and (2) to examine both sensory and cognitive accounts of the data by comparing the relative success of selected predictor variables.

**The tonal hierarchy.** Tonality is a function of the hierarchical organization of pitch relationships. Hierarchical pitch relationships exist at three interrelated levels, that of note, chord, and key, and involve increasing levels of abstraction. The concept of the tonal hierarchy describes the relationship among the single tones within a key. One single tone, the tonic, forms a reference point for all tones in the key. Each of the remaining tones is located in a hierarchical relationship with the tonic. It is hypothesized that listeners relate individual tones to these reference tones in an ongoing fashion as music is heard, a process that contributes to a sense of tonality.

**The probe-tone method.** The probe-tone method was first used by Krumhansl and Shepard (1979) as an effective means of quantifying listeners' responses to tonality. As generally applied (Krumhansl, 1990), subjects listen to a stimulus context, typically consisting of a short sequence of notes or chords, or a scale or melody context, and then rate a following tone, the "probe tone", for degree of completion or goodness-of-fit with the context. The probe tones consist of all possible tones from the 12-note chromatic scale. The set of ratings, one for each of the 12 probe tones, is called the probe-tone profile for the context. The profile is assumed to reflect the hierarchy of stability of tones in the context. Where key-defining contexts have been employed, the profile is thought to reflect the relationship between each probe tone and the tonal centre of the context.

## Method

The data for the probe-tone tests described below were collected as part of a larger study involving an extensive series of music and cognitive tests.

**Subjects.** One hundred subjects participated in this study, ranging in age from 18-40 years (mean = 27, SD = 6.2), with years of formal education ranging from 7-22 years (mean = 15.6, SD = 2.7). Sixty-one subjects had little or no music training, 22 subjects had moderate amounts of music training, and 17 subjects had high amounts of music training or were professional musicians.

**Apparatus.** A Yamaha TX81Z synthesizer controlled by a Zenith Z-248 computer running "DX-Score" software was used to create the single-note stimuli for the Probe-tone Melody test. A Yamaha TX802 synthesizer controlled by an Atari 1040ST computer using "Notator" music processing software was used to create the "Shepard-tone" stimuli for the Probe-tone Major and Minor Cadence tests. Shepard tones and chords do not have a well-defined pitch height. Each tone contains sine-wave components distributed over a multi-octave range under a half-cosine amplitude envelope which approaches hearing threshold at the high and low ends of the range (Shepard, 1964). All music sequences were recorded on audiocassettes and reproduced through the speakers of a portable tape player at comfortable listening levels, as determined by each subject.

**Probe-tone procedures.** For the Probe-tone Melody test, the melody used was the "March of King Laois" (from Johnston, 1985), an obscure 16th century Celtic tune, highly tonal in nature, characterized by simple elaborations of the tonic triad. The melody was played at a tempo of MM=100 (dotted half note) and lasted 12.2 seconds, and was followed, after a one-second pause, by a probe tone of one second duration. Each of 12 probe tones in the chromatic pitch set was presented twice, in random order, for a total of twenty-four presentations of the melody and probe tone. The Yamaha TX81Z synthesizer was set to factory present A15 (Wood Piano) while recording the melody and probe tones. Subjects were required to rate, on a scale of 1 - 10, how well the probe tone fit in with the melody that came before it. A rating of "1" indicated a poor fit and a "10" indicated a good fit.

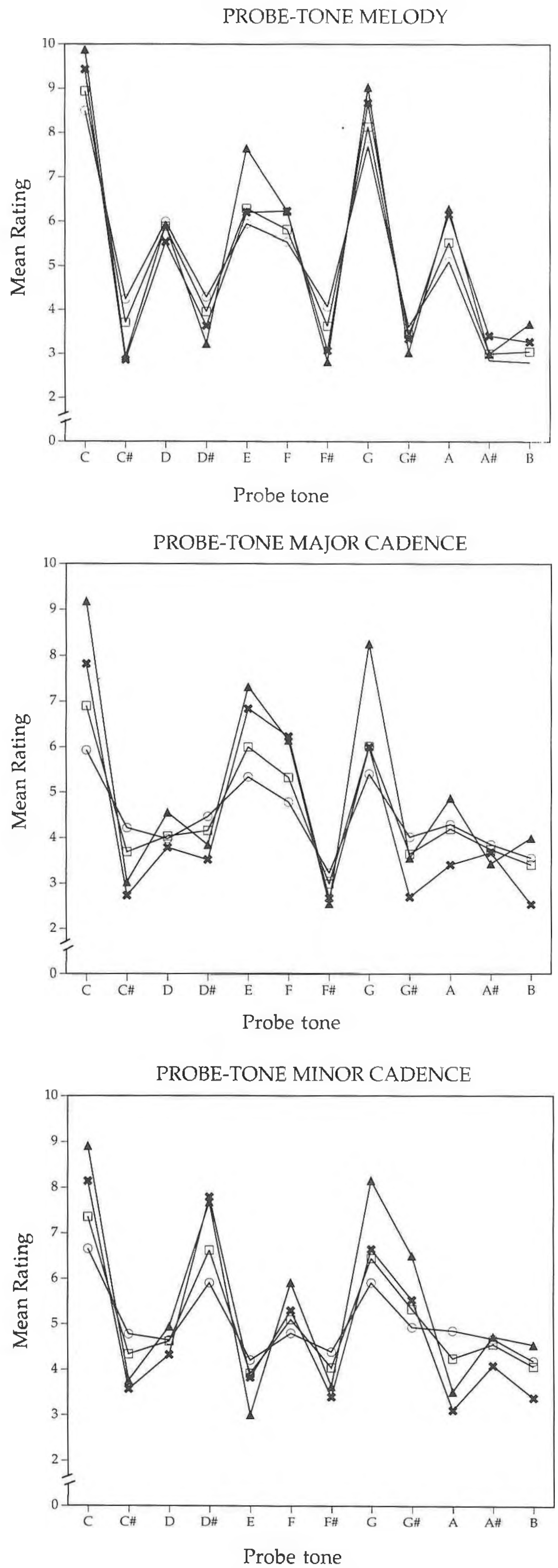
For the Probe-tone Major and Minor Cadence tests, twenty-four major (IV-V-I) and twenty-four minor (iv-V-i) perfect cadences, of three seconds duration, was each followed, after a one-second pause, by a probe tone of one second duration. The cadences were played at a tempo of MM=110 (quarter note). Each of 12 probe tones in the chromatic pitch set was presented twice, in random order, for a total of twenty-four presentations of each cadence and probe tone. Each chord was constructed as a Shepard chord and consisted of fifteen notes spaced over six octaves. The probe tones were similarly constructed as Shepard tones and consisted of sine-wave components spaced at octave intervals. The Yamaha TX802 synthesizer was set to produce sine tones for each chord tone and probe-tone note, with rise and decay times of 20 ms. each. Subjects were required to rate the probe tone, on a scale of 1-10, on how well it fit in with the preceding chord pattern. A rating of "1" indicated a poor fit and a "10" indicated a good fit.

## Results and Discussion

All obtained probe-tone profiles resembled the standardized profiles reported in the literature (Krumhansl, 1990). The order of the ratings was consistent with music theoretic descriptions of the tonal hierarchy. Subjects generally rated the tonic note (C) the highest, followed by the other notes of the tonic triad (E and G for the melody and the major-cadence contexts; Eb and G for the minor-cadence context), the remaining diatonic notes, and, finally, the nondiatonic notes (see Figure 1). These results suggest that each of the probe-tone contexts reliably assessed subjects' sense of tonality, and that all subjects evidenced a sense of tonality, regardless of level of music training.

It was noted that subjects with high music training produced profiles in which ratings were more clearly differentiated than subjects with little or no music training, and correlations of the profiles with music theoretic predictions were highest for subjects with high music training. Nevertheless, there was close correspondence between the profiles for the three training groups (average $r = .93$, all $p < .001$).

We next examined psychoacoustic and musical correlates of the perceived tonal hierarchies. A number of predictor variables were derived and the amount of variance in the




profiles accounted for by each predictor variable was computed.

The first set of predictors tested the acoustical property of consonance. Six sets of consonance values were obtained from a summary table in Krumhansl (1990, p. 57). These predictors assigned values to each probe tone in terms of the degree of acoustical consonance between the probe tone and the key centre of the musical context. Values were obtained, for example, from the roughness calculations of Helmholtz (1885/1954) and also from critical bandwidth modifications of the Helmholtz values.

The second set tested the predictive power of statistical properties of tonal music. The values assigned by these predictors were the relative frequencies with which each probe tone occurred in both major and minor keys. The values are obtained from statistical distributions of samples of tonal music (see Krumhansl, 1990, p. 67).

The third set of predictors were derived from Parncutt's (1989) model of pitch salience. This model, which applies specifically to the cadence contexts, assigns a value to each probe tone in terms of its spectral and virtual pitch weight in the stimulus context. Three versions of the model were tested.

Parncutt's model of pitch salience yielded the greatest success, both for the cadence contexts, and, surprisingly, for the melody context. One version (containing a sensory trace decay) consistently accounted for over 80% of the variance in probe-tone profiles for all three contexts. Possible reasons for the goodness-of-fit of this model will be discussed.

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**Figure 1.** Mean probe-tone ratings for Probe-tone Melody and for Probe-tone Major and Minor Cadences for all subjects and by levels of music training. (Open squares - all subjects; open circles - low music training; x's - moderate music training; triangles - high music training.)

# How grouping improves the categorisation of frequency in song birds and humans and why song birds do it better.


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There is evidence that song bird species produce, recognize, and discriminate song notes on the basis of frequency within a range. How song birds do this is unknown. One hypothesis is that song birds represent individual frequencies separately, somehow knowing which identify conspecifics. This hypothesis suggests that song birds memorise individual song frequencies as a list without underlying rules (see Herrnstein, 1990). Because of natural continuous variability in song frequency within and among individuals (Borror, 1961; Weisman et al. 1990) song birds might need to memorize a number of individual frequencies to discriminate conspecifics from heterospecifics. A second hypothesis is that song birds categorise song notes into frequency ranges, forming large open-ended categories (see Herrnstein, 1990), and use knowledge about these ranges in song. Here, song birds treat exemplars of a frequency range collectively as suggested by theories of category learning (Keller & Schoenfeld, 1950).

To decide between these hypotheses we trained zebra finches and, for comparison, humans in a (distributed S+) discrimination that required memorisation of individual frequencies and in a (compact S+) discrimination that could be acquired by classifying frequencies into ranges. If acquisition of the compact S+ discrimination is faster and more accurate then both species can use frequencies in a common range as a category. We examined transfer to novel frequencies to test whether the compact S+ groups form open-ended frequency range categories. The frequency range hypothesis predicts more control over responses to transfer tones following compact than distributed discrimination, because category learning causes subjects to treat tones within each frequency range collectively.

**Method**

We assigned 4 birds and 4 humans each to the compact and distributed S+ discrimination groups.

Discrimination Training

Subjects heard 27 tones, beginning at 2000 Hz separated by 120 Hz. In the compact S+ groups, 9 tones in the frequency range 3080-4040 Hz were positive: approach to the feeder produced food (zebra finches), breaking a photo-beam by hand produced visual and auditory feedback (humans); 9 tones each in the ranges 2000-2960 Hz and 4160-5120 Hz were negative: approach to the feeder produced lights out. In the distributed S+ groups, 9 tones spread across the 3 ranges (2000-5120 Hz) were positive and the remaining 18 tones were negative.

Transfer

Training continued for several days before the transfer test. The procedure alternated daily between discrimination and transfer sessions, which substituted 9 test tones for training tones. Over 3 transfer days, both groups heard 27 test tones, 9 per day, beginning at 2060 Hz spaced 120 Hz apart, and 18 training tones. The consequences for responding to training tones were unchanged, but responding to test tones ended the trial without food or feedback.

**Results and Discussion**

Acquisition to Day 15

In both species (Fig. 1), the compact group learned significantly ($p < .001$) more quickly and to a higher asymptote than the distributed group, whose discrimination ratios rose only slightly above chance. In comparison to zebra finches, humans learned the compact discrimination more slowly and less completely, and did not learn the distributed discrimination.

Transfer tests

Training tones. Percent responding in the compact groups increased significantly after all positive tones in zebra finches ($F(2,6) = 993.75$, $p < .001$), and humans ($F(2,4) = 18.94$, $p < .01$). In the distributed groups, the percent response to S-s just above and below S+s did not differ significantly in humans, ($F(2,6) = 2.68$, ns), but after 20 days of training (in addition to that in Fig. 1) zebra finches discriminated all S+s from all S-s ($F(2,6) = 22.48$, $p < .01$) (Fig. 2).

Transfer tones. The compact group had a significantly higher percent response to transfer tones in the middle (positive) range than in the lower and upper (negative) ranges in both zebra finches ($F(2,6) = 394.22$, $p < .001$) and humans ($F(2,4) = 28.74$, $p < .01$), showing that both species categorised transfer tones within each range. The distributed groups showed no evidence of transfer of discrimination: percent response to tones similar and different from training S+s did not differ , $F < 1$, (Fig. 2).

## General Discussion

Both zebra finches and humans can sort tones into frequency ranges. The compact groups learned quickly to categorise training tones in the middle range as S+s and tones in the lower and upper ranges as S-s. By contrast, zebra finches discriminated distributed S+s very slowly (about 40k trials) and humans discriminated them not at all. Discrimination of compact S+s transferred to other tones in the three frequency ranges, but discrimination of distributed S+s did not transfer to adjacent tones.

Because of natural variability in song (Borror, 1961; Weisman et al. 1990) the ability of song birds to identify conspecifics by individual frequencies would require an enormous amount of memory capacity. We found song birds have difficulty with this kind of memorisation, acquiring a distributed S+ discrimination very slowly. Given their facility in compact discrimination and their difficulties in distributed discrimination, it is likely that song birds identify the frequency of conspecific song using open-ended categorisaton rather than rote memorisation. This level of categorisation effectively reduces memory capacity. Humans, too, can use frequency range to categorise tones, but their performance lacks the crisp accuracy of zebra finches. Interestingly, humans, unlike zebra finches, can not memorise 9 distributed tones.

The frequency range hypothesis helps to explain why song birds produce the frequencies in song notes over narrow ranges and why the territorial responses of song birds decline abruptly when playback songs are transposed outside their normal frequency range (Falls, 1962; Emlen, 1972; Nelson, 1988).

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Fig. 1.

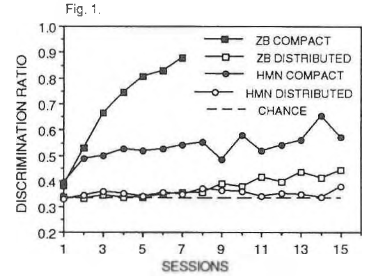
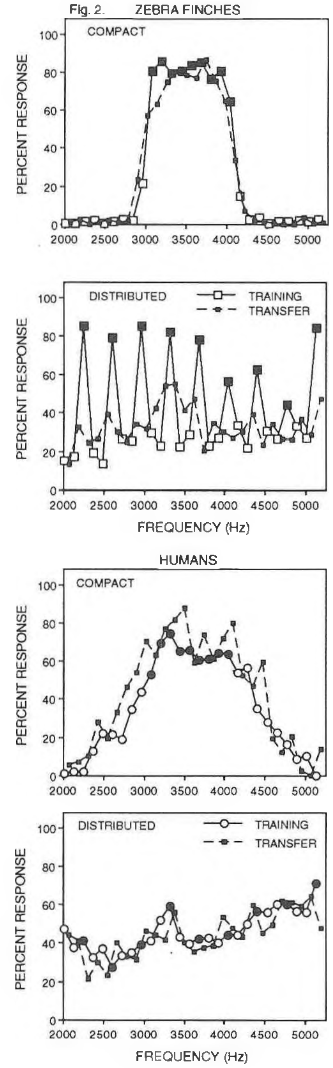



Fig. 2.



# The Perception of Rhythmic Similarity: A Test of a Modified Version of Johnson-Laird's Theory

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**Introduction**

Although many aspects of musical experience appear to be dominated by impressions of *resemblance*, music theory has provided little guidance concerning how to characterize or measure the degree of similarity between various musical passages. How is it that a listener is able to say that passage 'A' is more similar to passage 'B' than it is to passage 'C'?

Recently, Johnson-Laird (1991) has proposed a theory of rhythmic prototypes that has repercussion for the perception of rhythmic similarity. Specifically, Johnson-Laird claims that all rhythms generated from a given prototype are perceived as rhythmically similar. Further, Johnson-Laird suggests that certain prototypes are particular to specific genres of music.

Johnson-Laird assumes that (Western) rhythms are perceived within a metrical framework of duple, triple, or quadruple beats. Johnson-Laird distinguishes three basic ways in which a beat may be experienced. These basic beat experiences are referred to as "note" (a note onset coincides with the beat), "syncopation" (a note within the beat is followed by a note whose onset is after a metrical unit of greater importance than the metrical unit corresponding to the onset of the first note), and "other" (a beat that conforms to neither of the two previous types). A prototype of each measure is constructed by assigning each beat as one of three beat types. An example of a prototype for a triple meter measure with note onsets on each beat is "note-note-note". Johnson-Laird proposes that all one-measure rhythms derived from a single prototype should be judged as more similar to one another than to those based on a different prototype. Using the previous example rendered in 3/4 time, a measure of 3 quarter notes and a measure of 6 eighth notes are more similar to each other than a measure of one eighth note rest followed by 5 eighth notes (corresponding to the measure prototype "other-note-note").

Johnson-Laird's syncopated beat type relies on the events in the following beat because it is a theory of rhythm production. To create a syncopated note, there must be no note coinciding with the following beat of greater metrical unit. However, from a perceptual point of view, a listener is unable to distinguish the syncopated beat type from the "note" or "other" beat types until the next beat occurs. Using a theory of rhythmic production to model perception fails as it implies that listeners are retrospectively perceiving beat types. We suggest rather that listeners perceive a syncopation at the moment when a beat of greater metrical unit than the onset of the current note is traversed. We therefore assign the syncopated beat type to a beat which has been traversed by a note whose onset coincided with a beat of lesser metrical unit. Using two-beat prototypes, "sync-other" now becomes either "other-sync" or "note-sync".

In this paper, three tests of a perceptually modified version of Johnson-Laird's theory are reported. In the first instance, a perceptual experiment is described whose goal was to determine whether listeners perceive rhythms sharing the same prototype as more similar than rhythms of different prototypes. In the second instance, a sample of notated music was examined in order to determine the degree to which actual musical practice conforms to one of several predictions made by Johnson-Laird. In the third instance, a further perceptual experiment is described where listener's perceptions of real music is correlated with prototype analyses. The results provide broad empirical support for Johnson-Laird's theory of rhythmic similarity.

**Experiment I**

Four musician subjects were recruited from an undergraduate population. Stimuli were presented in a two-alternative forced-choice paradigm. For each trial, four different rhythms were presented in two pairs. One pair was composed of two rhythms that shared the same prototype while the other pair had two rhythms from different prototypes. Subjects were asked to judge which of the two pairs contained rhythms that sounded most similar to each other.

Each trial maintained a constant metric framework. Each pair of rhythms was preceded by two measures of metronome clicks in order to establish the meter and tempo. The first beat of each measure in the metric context was accented. A two measure rest separated the first and second pair of rhythms. After the second pair of rhythms, a four measure response period elapsed before the onset of the next trial.

Trials were organized in two blocks of 10 trials. All trials in each block employed stimuli whose measures had the same number of beats. Hence, each subject heard a triple-meter block and a quadruple-meter block.

The results of experiment I are summarized in Table 1.

| Subject | Agreement with Theory |
|---|---|
| 1 | 16/20 |
| 2 | 14/20 |
| 3 | 15/20 |
| 4 | 19/20 |

**TABLE 1.** Results of Experiment I

These results are significant at the $p < 0.001$ confidence level. Experiment I showed that rhythms conforming to the same rhythmic prototype are more likely to be judged similar.

**Experiment II**

Johnson-Laird suggests that musical genres can be distinguished by the use of stereotypic measure prototypes. As testing this hypothesis suggests an exhaustive notated music analysis, a different hypothesis is proposed. The redundancy in the use of measure prototypes is expected to be high for early music and to progressively decrease as musical styles adopt more and more complicated rhythms.

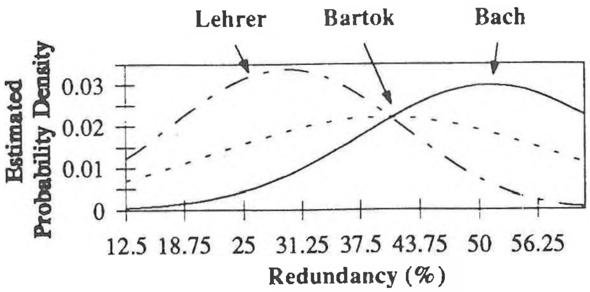


**Figure 1.** Results of Experiment II

To test this hypothesis, the redundancies in the use of one-measure rhythmic prototypes in 54 individual works by Bach, Bartók , and Lehrer were calculated using an information measure. Probability density functions were then estimated using Parzen windows (Duda & Hart, 1973) and are presented in figure 1. As a histogram, Parzen windows approximates the underlying probability function that is responsible for producing a set of data. However, unlike a histogram, Parzen windows requires no assumption about the size of the bins which may greatly affect the shape of a histogram.

There is a significant difference in the means of the distributions. Works by Lehrer have a quarter the redundancy in the use of rhythmic prototypes as compared to Bach. Bartók sits in the middle of the two, both in redundancy and in history.

#### Experiment III

Three subjects were recruited from an undergraduate population. Subjects were all musicians. An ecological test was administered to collect subjective measures of similarity within an audited musical work. Recordings of the 15 Inventions by J.S. Bach were made available to each subject. Subjects were instructed to listen to the inventions as often as wished and in any desired order. They were asked to rate each invention according to the perceived rhythmic complexity within each piece.

As a first estimate of perceived rhythmic similarity within a musical piece, the redundancy in the use of one-measure prototypes was calculated for each invention. The results of experiment III are summarized in Table 2. Kendall's Q measures the concordance between two variables. The coefficient ranges between 0 and 1, with 0 signifying no agreement and 1 signifying complete agreement.

| Subject | Kendall's Q | Significance |
|---|---|---|
| 1 | 0.5802 | 0.2985 |
| 2 | 0.4781 | 0.4964 |
| 3 | 0.5379 | 0.3741 |
| All | 0.5320 | 0.0554 |

**TABLE 2.** Results of Experiment III

Although the results are not significant taken individually, the concordance is skewed in the predicted direction. As a group, the subjects showed marginal significance.

#### Discussion

The first perceptual experiment provided significant empirical evidence supporting the modified Johnson-Laird theory of rhythmic prototypes as a useful predictor of human perception of rhythmic similarity. However only musicians were tested, amongst whom the best results were obtained from those most involved with musical activity. It may be that rhythmic prototype recognition is an acquired means of organizing perceived rhythms. Further tests with non-musicians could provide empirical support for this hypothesis.

The ecological test produced marginally significant results ($p = 0.0554$).

Johnson-Laird's theory of rhythmic prototypes applies only to a single stream of notes and so examining 2-part music raises some problems. For the two-part inventions, each part was analyzed separately and their redundancies averaged. The listener however was perceiving both lines simultaneously. This divided attention across two auditory streams and its effect on rhythmic perception is beyond the scope of this proposed theory of rhythmic similarity.

Johnson-Laird's original theory remains untested. As there was no experimental evidence to support the original theory, a modification was undertaken and the resulting theory directly tested. Generally speaking, Johnson-Laird's original theory makes less distinctions between beats than the modified theory. Therefore using the original prototype definition to rate the results of the perceptual experiments would not be expected to produce greater significance.

#### Conclusion

The modified version of Johnson-Laird's theory of rhythmic prototypes presented in this paper helps in characterizing the human perception of rhythmic similarity. The three tests carried out provided for broad empirical support of the proposed theory.

The first of the two perceptual tests produced significant results, whereas the second more ecologically valid showed a marginally significant positive correlation with the proposed theory. The theory examined in light of a sample of notated music supports the hypothesis that musical genres can be distinguished by the redundancy in the use of prototypes. The empirical support presented in this paper invites further study of rhythmic prototypes along the lines suggested by Johnson-Laird.

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# DETECTION OF BEARING FAILURE IN MACHINES

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## Introduction

One of the potential problems common to all types of bearings is that at failure, the resultant machine repair can be costly, both financially and in production loss.

The vibrational energy emitted from rolling contact bearings can be monitored in a number of ways:

1. Overall amplitude of vibration level, based on time domain data.
2. Frequency spectrum of the time signal, usually up to 25 kHz.
3. Examination of the shock waves generated through the bearing housing when the rolling elements move over a damaged area.
4. Statistical parameter measurements applied to the time signal.

It is the latter approach that will be highlighted in this presentation.

Machined or ground surfaces are not perfectly smooth but consists of a pattern of asperities. In fact, only about 0.1% of the nominal contact area actually touches under normal loading conditions. When these surfaces are moving, as is the case of rolling element bearings, the asperity tips are alternatively welding and breaking off. Even so, most bearing surfaces exhibit randomness in asperity distribution in the direction of the machine process.

It has been well established in the metrology literature that the distribution of asperity weights for an undamaged surface to be normal, that is Gaussian in nature. It is this feature that allows the statistical parameter approach to work.

## Background to the Statistical Parameter Method

The asperity distribution can be detected with an accelerometer, attached as near as possible to the bearing housing. This will, of course, also collect data relating to ball passing frequencies and other structural resonances. It is necessary therefore to separate the random data from deterministic data, by filtering. Fortunately, this usually works out quite well, and removing signal content below 2.5 kHz is usually satisfactory. The remaining data, from most bearings at normal running speed, should be mainly random. As the surface deteriorates due to damage, the "bell" shaped probability density function starts to lose its classic characteristics. Statistical moments can then be used to sense these changes.

In general, odd moments are related to information about the position of the peak of the PDF in relation to the mean value, while even moments indicate the characteristics of the spread of the distribution. It can be shown theoretically that the fourth moment, normalized by dividing with the square of the second moment, results in a coefficient called 'Kurtosis.' This has a value of 3.0, if the data exhibits a normal PDF. Values greater than 3.0 indicate damage developing, and less than 3.0 usually indicate deterministic signal contamination.

$$M_2 = \frac{1}{N}\sum_{i=1}^{N}\left[x_i^2 - \overline{x}^2\right] \tag{1}$$

$$M_4 = \frac{1}{N}\sum_{i=1}^{N} x_i^4 - \frac{4}{N}\,\overline{x}\sum_{i=1}^{N} x_i^3 + \frac{6}{N}\,\overline{x^2}\sum_{i=1}^{N} x_i^2 - 3\overline{x^4} \tag{2}$$

and

$$Kurtosis = \frac{M_4}{(M_2)^2} \tag{3}$$

One significant advantage that statistical methods have over the spectral approach is that the results from the former are essentially insensitive to load and speed changes, Fig. 1. Table 1 shows the moments as calculated from theory for different signal types. Notice the difference between random and Gaussian random.

## Typical Results

Raw data can be collected in a number of ways. Under laboratory conditions, accelerometers positioned on the casing surrounding bearings under investigation can input analog signals directly into an A/D board in a microcomputer. Conditioning, filtering, and analysis can then be carried out using a variety of software packages. In-house software can be written in 'C' or Pascal for example, or such general commercial packages such as MATLAB, can be used very effectively. How the data processing is handled depends on the funds available, and the flexibility required for the mathematical processing.

In field work, for example on top of a 54 storey building in an elevator shaft, recording data from the accelerometers directly into a good quality tape recorder is often a much easier route to take. The processing can then be carried out in the relative comfort of the computer at home base.

Table 1

| | Moment | | Normalized Moment using $(M_2)^n$ | | | | |
|---|---|---|---|---|---|---|---|
| Data type | $M_1$ | $M_2$ | $N_3$ | $N_4$ | $N_5$ | $N_6$ | $N_7$ |
| Sine | 0.00 | 0.50 | 0.00 | 1.48 | 0.00 | 2.44 | 0.00 |
| Square | 0.00 | 1.00 | 0.00 | 0.98 | 0.00 | 0.98 | 0.00 |
| Triangle | 0.00 | 0.34 | 0.00 | 1.75 | 0.00 | 3.71 | 0.00 |
| Random | 0.48 | 0.07 | 0.13 | 1.94 | 82.90 | 4.74 | 2.81 |
| Random Gauss | 0.01 | 0.94 | 0.20 | 3.01 | 2.44 | 14.30 | 20.70 |

Fig. 2, shows the time domain results for similar bearings under different damage conditions, as summarized in Table 2.

| Table 2 | | |
|---|---|---|
| Bearing Number | Condition | Kurtosis Value |
| 1 | good | 2.9 |
| 2 | Hair line mark outer raceway | 3.8 |
| 3 | Severe scratches outer raceway | 4.4 |

Plotting Kurtosis values against acceleration magnitudes is often termed a damage map, so for example in Fig. 3, a good bearing is first run lubricated, and then cleaned out and run dry. The numbers indicate Kurtosis measurements taken in the following frequency bands,

1. 2.5 to 5.0 kHz
2. 5.0 to 10.0 kHz
3. 10.0 to 20.0 kHz
4. 20.0 to 40.0 kHz
5. 40.0 to 80.0 kHz

Laboratory tests have shown that the damage map can be used to identify the type of damage process in progress. This summary is shown in Fig. 4.

## Conclusions

The wide range of methods of detecting bearing damage that are available, have their individual strengths and weaknesses and are therefore dependant on the application. An understanding of the methodology involved in digital data processing is important in order to ensure valid interpretation of the results.

Statistical methods of various types are now being investigated, Kurtosis being the most tested approach so far and seems to offer a lot of potential for the future. Currently investigations are being carried out using other distributions such as the Beta and Weibull functions, applied to gearing as well as bearings. However, confidence still has to be built up with extensive field testing of these techniques.

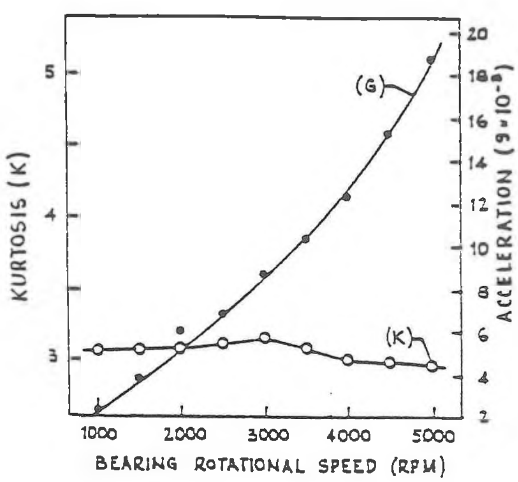



Fig 1 . Kurtosis and "g" level as Functions of Bearing Rotational Speed.

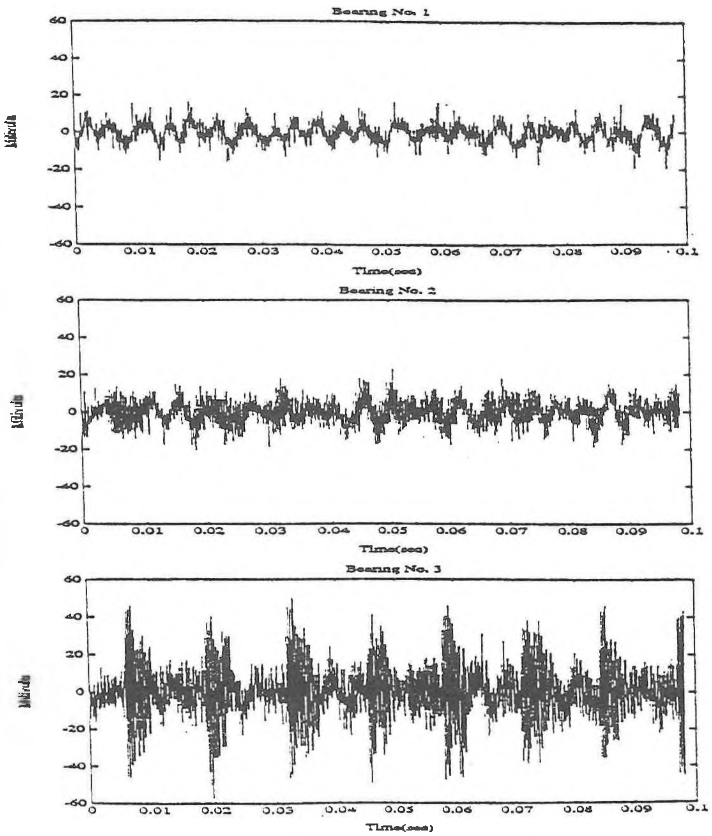



Fig 2. Vibrational Data for Different Stages of Damage.

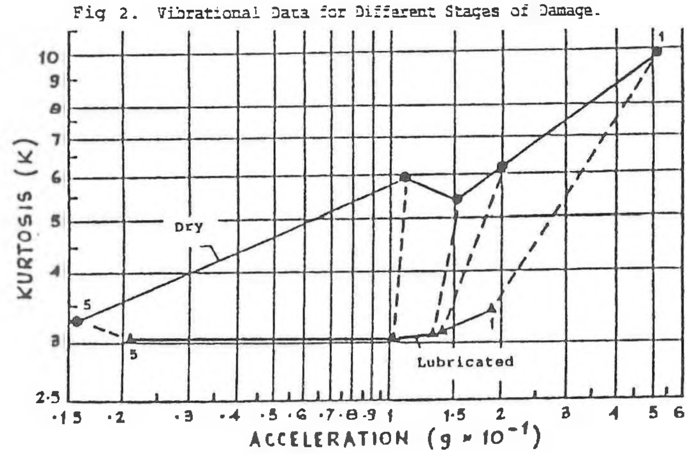
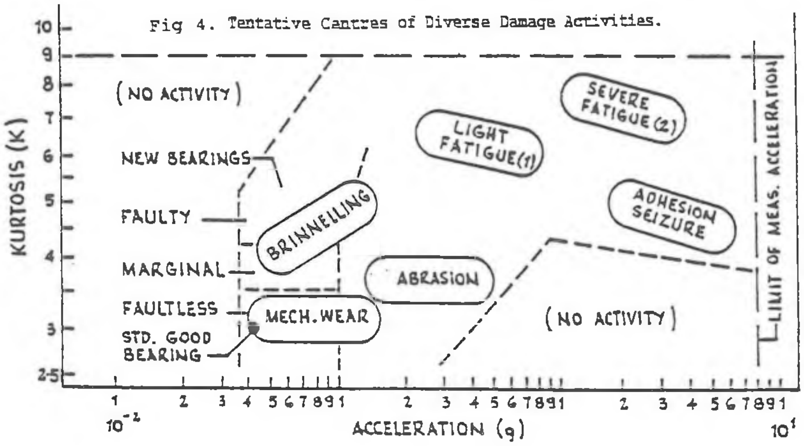



Fig 3. Difference between a Dry and Lubricated Bearing.

Fig 4. Tentative Centres of Diverse Damage Activities.



# The Development of a Cost Effective Engine Dynamic Signal Monitoring and Diagnostic System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## ABSTRACT

This paper describes the development of a cost effective engine dynamic signal monitoring and diagnostic system for use in a high volume engine manufacturing plant as well as in a research environment for development of new engine components.

Modal analysis was performed to determine the optimum locations of vibration transducers. The techniques of decomposition of raw signals obtained from an engine are described.

The engine dynamic signal monitoring and diagnostic system has been successfully implemented in on-line assembly of engine. In all cases, the system is capable of detecting and isolating manufacturing and assembly defects.


## INTRODUCTION

Today, one of the key goals of an engine manufacturer is to achieve higher quality and productivity at competitive cost. This coupled with higher standard of customer satisfaction, requires not only innovative concepts in engine design, manufacturing and assembly processes, but also the integration of on-line engine monitoring and diagnostics system which is capable of detecting manufacturing and assembly defects.

This paper describes the development of a viable and cost effective engine dynamic signal monitoring and diagnostic system for detecting and locating manufacturing and assembly defects by means of time domain noise and vibration signature analysis.

## TIME DOMAIN AVERAGING

The time domain signal processing techniques are extremely useful for engine diagnosis because of their abilities to correlate the amplitude of the signals at corresponding angular positions of the crankshaft.

Time domain averaging (TDA) is a powerful technique for extracting periodic components from a complex dynamic engine signal. It is accomplished by simply averaging several time traces, data point by data point, producing an average time trace. It is useful for detection of faults which occur consistently at a certain locations in a cycle.

## VARIANCE ANALYSIS

In the last section time domain averaging was described, which allows the extraction of periodic components which are always present in the signal. The other components which are not always present but occur at specific location in a cycle are termed as semi-periodic can be extracted by variance analysis of the signals.

The mathematical descriptions of the TDA, including its properties as a filter in eliminating non-cycle linked signals, and variance are provided in reference (1).

## MEASUREMENT SYSTEMS

Two systems, time locked and position locked, are used to accomplish time domain averaging and variance analysis. The time locked system is consisted of a digital storage oscilloscope interfaced and controlled by computer that is triggered by TTL (Transistor-Transistor Logic) signals which are generated by an inductive voltage at #1 spark plug firing.

In the position locked system, exact measure of angular position can be achieved by sampling the data at a rate dictated by a 512 pulse per revolution encoder, installed in-line with the drive train. This encoder generates external sampling clock. Thus, a sampling resolution of up to 0.6 degrees of crank rotation can be achieved. The custom designed engine signal monitoring and diagnostic system was used coupled with specially build encoder system. This system uses the PIP (Profile Ignition Pickup) and CID (Cylinder Identification) signals from the engine electronic control for triggering the start of the event and hence enable to reference the zero crank angle to piston # 1 on the firing stroke.

Figures 1 and 2 show the waterfall plots of vibration variance of a defective engine at varying speed with time lock and position lock data acquisition system, respectively. With position lock system the angular location of the peaks indicating defect are consistenly at 390 degree. While the time lock system the locations of the peaks were scattered.

## EXPERIMENTAL RESULTS

The experimental results are divided into two sections: (a) Dynamometer test and (b) cold test.

## DYNAMOMETER TEST

Warranty field return engine with noise complaint was analyzed to determine the root cause. Vibration measurements were made with accelerometers mounted on the cylinder block wall. Figure 2 illustrates the waterfall plot of vibration variance with respect to crank angle. The plot indicates that the distinct amplitudes are occuring at about 390 degrees crank angle which coincide with the peak pressure of cylinder #5. Upon engine teardown and measurements, it was revealed that the concentricity of the piston ringland with respect to skirt was excessive in the thrust direction. When piston number 5 was replaced with concentric ringland with respect to skirt piston, the vibration was drastically reduced as clearly illustrated in Figure 3.

## COLD TEST

With the successes of using the custom designed engine dynamic signal monitoring and diagnostics system for evaluating piston design as well as diagnosing warranty field return engines, the next step is to apply the same system in cold test for on-line detection of manufacturing and assembly defects.

Modal analysis was performed in a partially assembled engine (as shown in Figure 4) and the results were reviewed for all modes of vibration up to 3200 Hertz. The optimum locations of accelerometers were selected for two reasons: (a) exhibiting the maximum response and (b) accessible for accelerometers carried by hydraulic actuated arm to reach.

Figure 5 illustrates the waterfall plot of vibration variance which is generated by partially assembled engine with connecting rod bearing missing. Distinct amplitudes appear coresponding to the side force reversals in the piston.

The experiments were repeated and the results indicate the feasibility of using the system for consistent on-line detection of missing rod bearing, missing cap bearing and loose connecting rod nuts.

## CONCLUSIONS

The signal processing techniques used in this paper are useful in relating the vibration signal to the rotational angle of the crankshaft of the engine and hence to its components such as the piston regardless of fluctuation of engine speed.

As the result of experimentation in the cold test stand, a custom designed engine dynamic signal monitoring and diagnostics system is implemented for the rapid and precise detection of manufacturing and assembly defects. This has resulted in increased productivity and quality and reduced manufacturing cost.

The dynamometer test results indicate that the system has been successfully used for the root cause determination of warranty field return engines as well as for evaluating engine component design changes.

## REFERENCE


1. Braun, S. and Seth, B., " Analysis of Repetitive Mechanism Signatures", J. of Sound and Vibration, 1980.


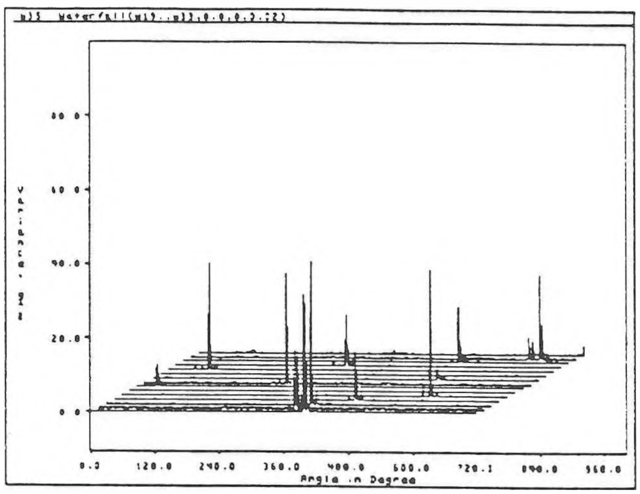

Figure 1: Vibration variance of defective engine at varying speed using time locked measurement system.

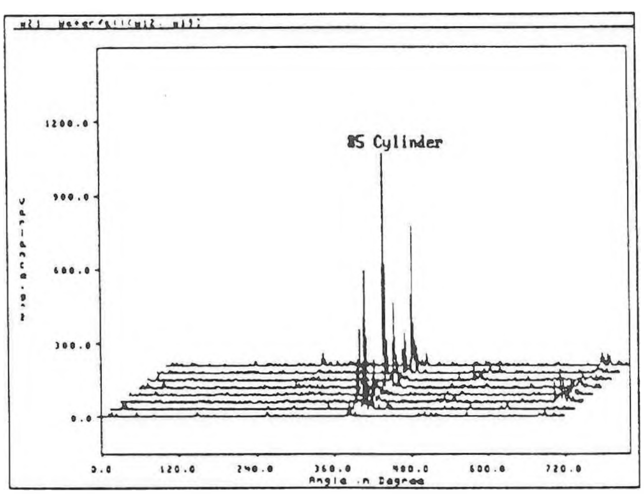


Figure 2: Waterfall plot of vibration variance with respect to crank angle for warranty field return engine - position locked measurement system.

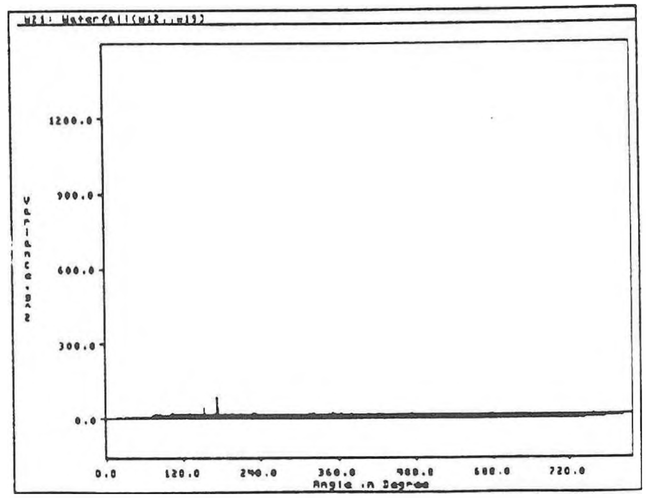


Figure 3: Waterfall plot of vibration variance with respect to crank angle for warranty field return engine after replacing with concentricity of piston ringland with respect to piston skirt.

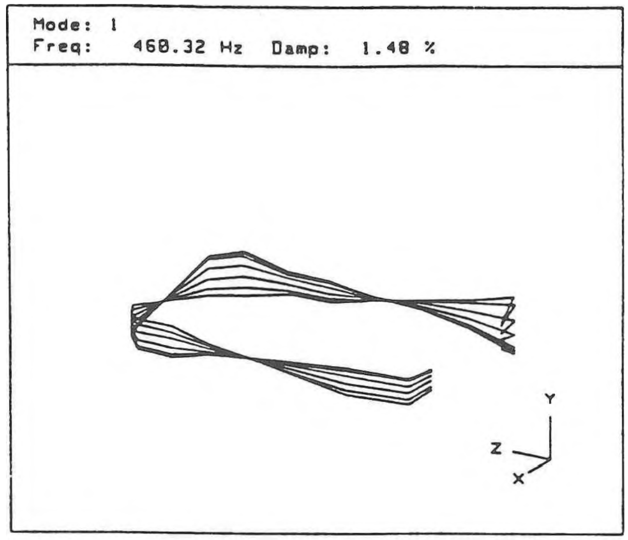


Figure 4: The first mode shape of the oil pan rail.

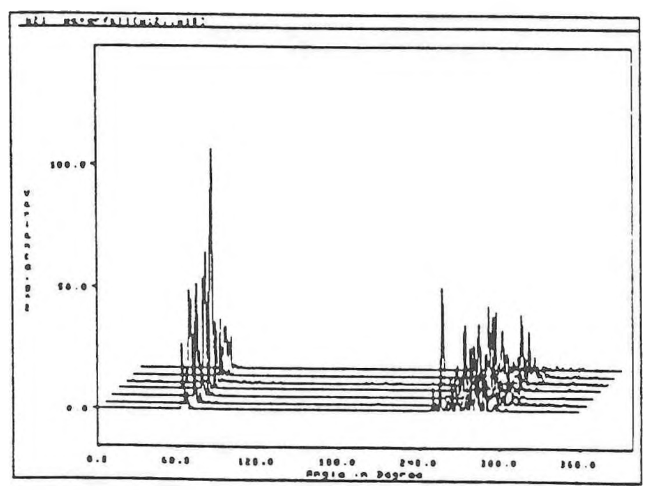

Figure 5: Waterfall plot of vibration variance of partially assembled engine with missing connecting rod bearing.

# AN OPTICAL METHOD FOR CHATTER AND FORM ERROR DETECTION IN GRINDING

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**Introduction**

Vibration is an inherent part of the machining processes. Any movement between the tool and work-piece during machining, however, may produce rough and unsightly marks on the parts. If excessive vibration is involved, it will cause dimensional errors on the finished part that results in the rejection of the product. In addition, vibration may reduce tool life, not to mention tool breakage or damage to the machine if the vibration is allowed to grow uncontrolled. Thus, elimination or reduction of machining vibration is a desirable but elusive goal in manufacturing. This requires an early detection of vibration so that a corrective action can be taken in time to remedy the problem.

To detect vibration in a production environment, noise and vibration sensors are used for monitoring the cutting process, however, it is difficult to relate the output of these sensors to the surface conditions and the dimensional accuracy of the part. Often, one needs not only to monitor the process but also the end product as well. In the grinding process for example, the easiest way for process monitoring is by visual inspection of the finished parts. This is normally carried out by experienced operators who are frequently biased and qualitative. For accurate measurement, one needs to resort to the conventional stylus instrument for checking the suspected part. This instrument, however, because of its contact nature, only operates in a clean room and cannot be adapted for high speed on-line inspection.

There is a need for a sensor to measure the surface quality of the part in the manufacturing, especially in grinding which is one of the most common finishing processes in production industries. Because of the non-contact and fast speed advantage, various optical methods were developed for the measurement of surface roughness in a production environment[1]. However they are limited to the measurement of short surface wavelengths[2]. Other sensors were also developed to extend the measurement range[3], nevertheless, their use was limited because of the complexity of the design.

The main objective of this work is to develop a simple optical method to detect chatter marks and form errors on the ground surfaces. These components are mainly periodic; however, they differ in direction, magnitude and frequency. Chatter marks, in general, have larger amplitude and longer surface wavelengths than the machining marks. Form errors, on the other hand have the greatest wavelengths and amplitudes. On cylindrical surfaces such as journal bearings which are the main focus of this work, these errors are termed as "lobbing".

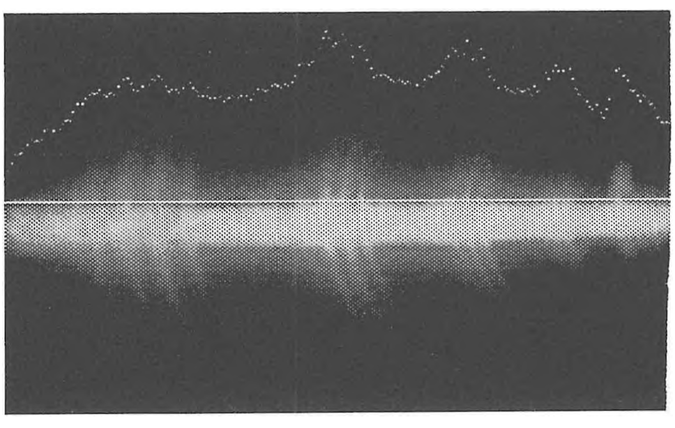

Figure 1

**Principle**

The proposed method is based on the measurement of the specular reflectance of the light from the ground surface. This reflectance is a function of the local roughness and the slope of the surface of interest. Vibration and chatter marks are the localized rough patches on the surface. As such, they tend to scatter the light in all directions and thereby reduce the reflected light intensity in the specular direction. Machining error or lobbing is the long wavelength component whose slopes modulate the intensity of the reflected beam. Figure 1 shows the light scattering pattern of a ground cam shaft journal with different chatter marks as captured by a CCD camera. These marks are identified as parallel curved lines of different spacings. As the machining marks, chatter marks and lobbing are of different wavelengths, it is possible to identify them in the pattern and subsequently separate them via signal analysis.

**Experimental Set-up and Procedure**

An optical surface topography sensing device was developed based on the above principle. A schematic diagram of the experimental set-up is shown in Figure 2. This includes a 5 mW diode laser (670 nm wavelength) which was used to illuminate the surface (at a 45 degree incident angle) through a focusing lens. A silicon photo-diode was placed in the specular direction of the reflected light for the measurement of its intensity. An aperture was used to reduce the active area of this photo-diode detector to 0.03 $cm^2$ for better resolution. A filter was also incorporated in the light path to match the reflected light intensity to the dynamic range of the sensor. The output of the sensor was digitized by an A/D converter and processed by a PC AT.

A number of journal bearings from different camshaft samples was used in the test. Each shaft was held in a fixture which can be rotated at a constant speed of 2 rpm (or 6 mm/s surface speed). As the shaft was rotated, a trace from the photo-diode output was obtained, see Figure 3.

The stylus trace of the sample was also obtained in the same manner using a Talysurf. The shaft in this case was rotated while the pick-up (stylus sensor) was stationary. This would produce a surface roughness profile of the journal. A form measurement was also obtained for the samples by using a Talyrond which would give a trace of the long wavelength components. The stylus traces were then compared to the optical ones to determine the performance of the optical method.

**Results and Discussion**

The quality of ground sample surfaces varied from good to unacceptable because of excessive chatter or lobbing. The surface roughness of the samples was in the range of 0.2 to 0.4 $\mu m$.

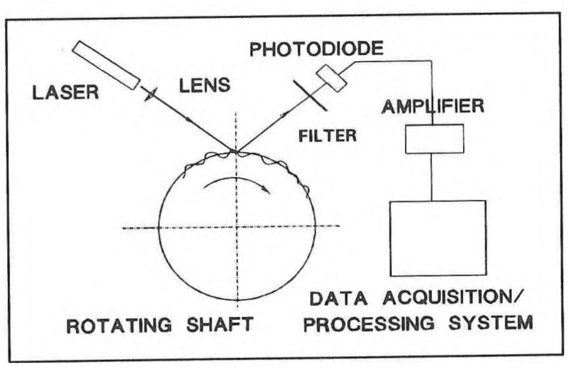


Figure 2

The performance of the optical method was determined by correlating the frequency and amplitude of the optical traces to those of the stylus. To correlate the frequency data, an FFT was performed on both signals and the locations of the major peaks were determined. A typical spatial frequency spectrum of the optical trace is shown in Figure 4. In this figure, one can identify the form error, chatter and vibration marks by their spatial frequencies. These frequencies were confirmed by the results from the Talysurf and Talyrond measurements. A polar plot of the Talyrond trace shown in Figure 5 indicates the presence of predominant chatter marks super-imposed on minor lobbing on the journal surface.

An amplitude (peak to peak) correlation between the optical trace and that of the stylus was also made. This was performed for the surface waviness of a given spatial frequency. Figure 6 shows a correlation curve for the waviness at a spatial frequency of 0.34 c/mm. It can be observed that the optical output increases as the profile amplitude increases, however, the relationship is not linear. This behaviour is likely governed by the relationship between the surface reflectance and roughness. A similar trend was also observed in the correlation at other spatial frequencies.

For a given grinding operation, it is possible to obtain a similar amplitude calibration curve for the part. This calibration curve can be used to determine the rejection threshold in the parts inspection process. For the present set-up, the proposed system can perform a measurement while the part is rotating at a speed of at least 400 inch /min.

**Conclusions**

An optical technique for detecting chatter marks and vibration undulation in surface grinding was developed based on the measurement of reflected light intensity from the surface. This method provides a useful tool for routine comparative inspection in production. This method also allows us to determine accurately the wavelength and amplitude of the surface waviness in the machine parts. From this information, one can infer the condition of the tool or machines so that corrective action can be taken at a proper time. With the implementation of an analog circuitry, this method can provide a high speed measurement where the inspecting part is rotating at a high speed. This will enhance the on-line measurement capability of the method. Furthermore, as the method is simple to design and operate, it offers a fairly attractive means for monitoring the grinding process in production or manufacturing industries.

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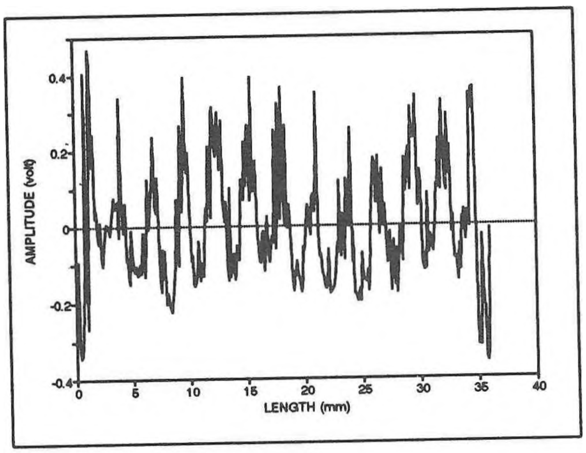


**Figure 3**

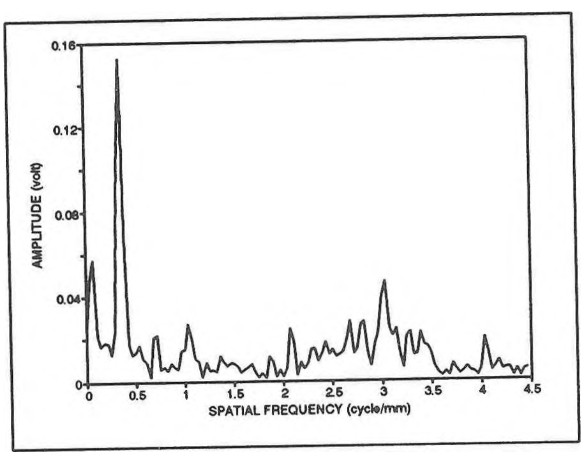


**Figure 4**

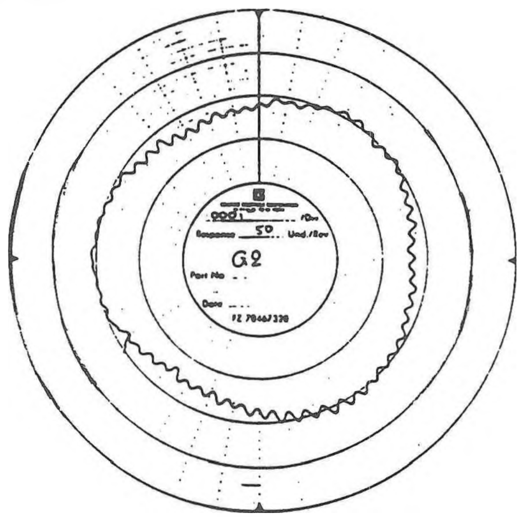


**Figure 5**

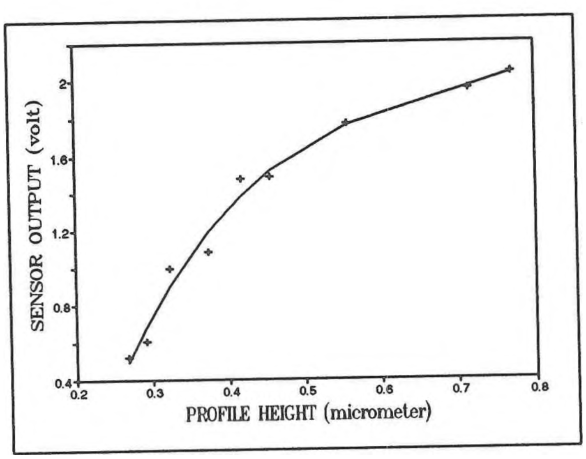


**Figure 6**

# Automotive Accessory Drive System Modelling


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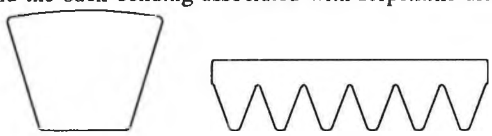
## INTRODUCTION

Belt driven accessories are ubiquitous in the automotive industry. The past practice was to include an additional belt for each accesory added as would be the case for the power steering pump and air conditioning compressor. This creates problems during replacement of a broken belt; particularly if it is located at the back of the belt stack since all other belts must be removed to affect its replacement. However, the automotive industry has began replacing many of the conventional multiple belt accessory drive systems with a new drive arrangement capable of powering several (if not all) of the accessories with a single belt.

The profile of this single belt, sometimes called a Poly-V belt, differs markedly from the traditional V-belt. As illustrated in Figure 1, it is much wider, thinner and uses a different profile on its primary contact surface. The drive systems that use this style of belt have been called "serpentine drives". This refers to convoluted path the belt must follow to accommodate the accessory locations and to provide sufficient contact with the accessory pulleys to transmit the driving force (especially when the flat side of the belt is used). The V-belts used in the conventional automotive accessory drives generally do not have sufficient flexibility to withstand the back bending associated with serpentine drives.



**V-Belt** **Poly-V Belt (6PL)**

Figure 1 Comparative belt profiles having the same minimum radius of curvature.

The thinner profile of these ribbed belts means a more compliant belt with a longer flex life and less centrifugal tension loss due to belt wear and seating. Serpentine systems with the ribbed belts also provide greater drive ratio flexibility since smaller diameter pulleys can be used without sacrificing the belt's flex fatigue life.

## PROBLEM IDENTIFICATION

Like their predecessors, the automotive serpentine accessory drive systems are still troubled with a number of noise and vibration problems. These are perceived as quality defects by the customer, and increased pressure has been placed on the suppliers of the accessory drive technology to reduce them. These problems frequently take the form of large amplitude belt span vibrations known as belt whip or belt flutter which can produce objectionable noise. Reducing the associated vibrations would not only decrease the generated noise levels but would also reduce belt and bearing wear.

It is generally accepted that the span vibrations can be directly attributed to two different classes of excitation. These are: (1) direct or internal - irregularities in the sheave, belt or shaft and (2) parametric or external - variations in the belt span tension. Of these two forms of excitation, the majority of the vibrations encountered in belt drive systems can be attributed to torque or tension variations, although impulses from the system irregularities can be significant. The geometric parameters of the system must be selected so that the natural frequency of vibration of the belt spans are not integral multiples of the excitation frequency caused by system irregularities. This last point is important since large amplitude vibrations of the belt span may be caused by this resonance effect.

The majority of the investigations concentrating on this research area have been associated with the determination of the natural frequencies of the span vibrations and the instability which results from belt tension fluctuations. Little emphasis has been placed on the role played by the angular motion of the accessory pulleys due to complex crankshaft oscillations and accessory loading characteristics. The magnitude and frequency of this angular motion is directly responsible for the belt tension fluctuations and hence the instability of the span vibrations. Thus, in response to those who would suggest that non-linear modelling is the appropriate direction to follow, it should be recognized that it in necessary to first determine whether that degree of modelling is required.

## THEORETICAL MODEL

A lumped parameter multiple–degree–of–freedom model was developed from which the angular modal parameters (resonance frequencies and mode shapes) of an automotive serpentine accessory drive system could be estimated.

### Belt Configuration Code

The general equations governing the angular motion of a serpentine belt drive system depend on the belt configuration (the path the belt follows around the accessories). A code was established to identify the belt configuration and is illustrated in Figure 2. Illustrated are the eight possible combinations of three consecutive pulleys which must be considered when analyzing the motion of a particular pulley or accessory. With this code "Y" represents a pulley which is reverse-wrapped (driven by the belt's flat surface) and "N" indicates a pulley which is not reverse-wrapped (driven by the belt's ribbed surface).

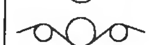
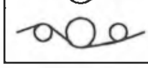
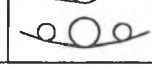
| ILLUSTRATION | CONFIGURATION | ILLUSTRATION | CONFIGURATION |
|---|---|---|---|
|  | N–N–N |  | Y–N–N |
|  | N–N–Y |  | Y–N–Y |
|  | N–Y–N |  | Y–Y–N |
|  | N–Y–Y |  | Y–Y–Y |

Figure 2 The possible belt configurations for three consecutive pulleys.

### Belt Span Stiffness

It has been observed that, over the range of belt tensions typically used in automotive applications, the belt can be assumed to obey Hooke's Law (i.e., constant stiffness). The stiffness value for each span is found from the belt span geometry and modulus of elasticity.

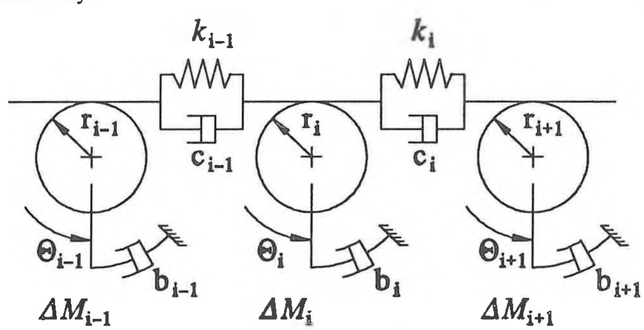



Figure 3 Lumped parameter model of connected pulleys and belts including elasticity and damping effects.

### Equations of Motion

The analytical model of the accessory drive system represents the accessories and pulleys as inertias constrained to move in only one direction. These are coupled by belt strand connections which are assumed to be linear and massless springs with linear viscous dampers as illustrated in Figure 3. By applying Newton's second law to the analytical model the internal and external torques are equated and a mathematical model is obtained in the form of a system of second order ordinary differential equations. These equations govern the rotational motion and are assumed to represent deviations from the steady motion. The internal torques consist of the inertia, damping and elasticity terms while the external torques are the excitation forces and include the moments due to fluctuating accessory loads and engine torques.

A system of statically coupled ordinary differential equations describing the rotational motion of each pulley can be obtained by applying the analysis technique to each of the pulleys in the system. The resulting equations can be expressed in the standard matrix form

$$[J]\ddot{\theta} + [C]\dot{\theta} + [K]\theta = [F] \qquad (1)$$

where $[J]$ is the inertia matrix, $[C]$ is the damping matrix, $[K]$ is the static or stiffness matrix and $[F]$ is the forcing function vector. The inertia matrix consists of the pulley and accessory inertia values along the diagonal while the damping and stiffness matrix elements ($C_{i,j}$ and $K_{i,j}$) are defined by the system geometry.

Unlike the typical spring–mass–damper damping and stiffness matrices, the off–diagonal elements may be positive and the elements in the upper right ($i$=1, $j$=$N$) and lower left ($i$=$N$, $j$=1) positions are not zero. Once the stiffness and damping factor for each belt span and the component damping factors have been determined the system of equations can be solved for the resonance frequencies, mode shapes and forced response.

### Resonance Frequencies and Mode Shapes

The resonance frequencies and mode shapes of the accessory drive system are determined from the mathematical model; however, due to the presence of complex eigenvalues and eigenvectors, the process is slightly more complicated than with undamped systems. The homogeneous form of the general matrix equation, a set of $N$ second order ordinary differential equations, was reduced to the set of $2N$ first order equations as illustrated in Equation (2).

$$\begin{bmatrix} 0 & J \\ J & C \end{bmatrix} \begin{Bmatrix} \ddot{\theta} \\ \dot{\theta} \end{Bmatrix} + \begin{bmatrix} -J & 0 \\ 0 & K \end{bmatrix} \begin{Bmatrix} \dot{\theta} \\ \theta \end{Bmatrix} = \begin{Bmatrix} 0 \\ 0 \end{Bmatrix} \qquad (2)$$

The roots are complex with the real component being the exponential decay factor and the imaginary component being the damped resonance frequency in radians per second. For stable systems the eigenvalues are real and negative or complex with negative real parts; the complex eigenvalues occur as conjugate pairs.

## EXPERIMENTAL WORK

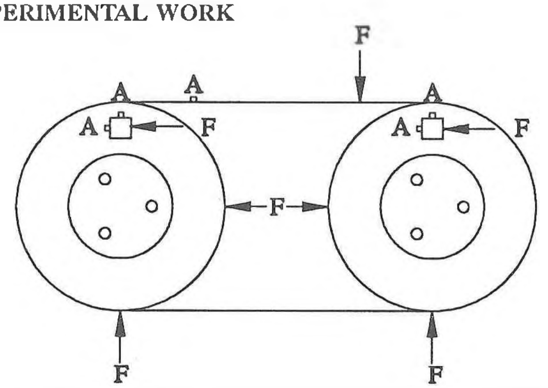



Figure 4 Experimental apparatus showing the excitation ("F") and vibration ("A") measurement locations.

A number of experiments were conducted on two- and three-pulley systems to investigate the angular vibration characteristics of simplified accessory drives. The effects of component inertia, belt tension and span length and belt tension on the belt span transverse vibration resonance frequencies. This investigation included the use of an impact hammer and accelerometer to extract modal information. Measurement locations are illustrated in Figure 4.

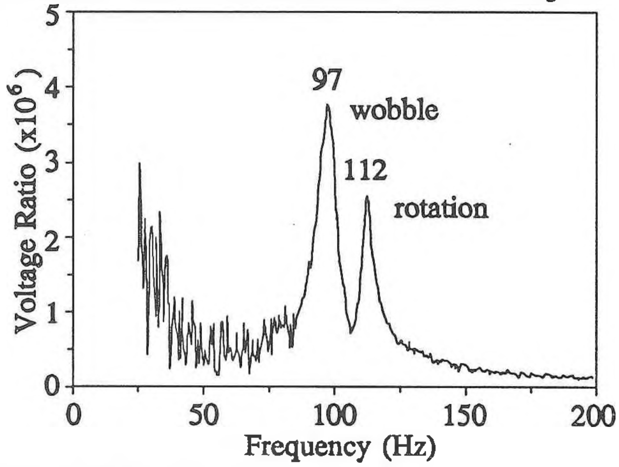



Figure 5 Sample system frequency response.

A sample of the experimental results are presented in Figure 5. Notable is the presence of both the rotational natural frequency as well as a "wobble" frequency. This phenomenon has been identified as a bending mode natural frequency of the shaft on which the pulley is mounted.
The "wobble" frequency has been observed to increase with belt tension, possibly due to a stiffening of the pulley in the bearing mount. It has also been observed to decrease as the pulley inertia is increased (mass of the pulley).

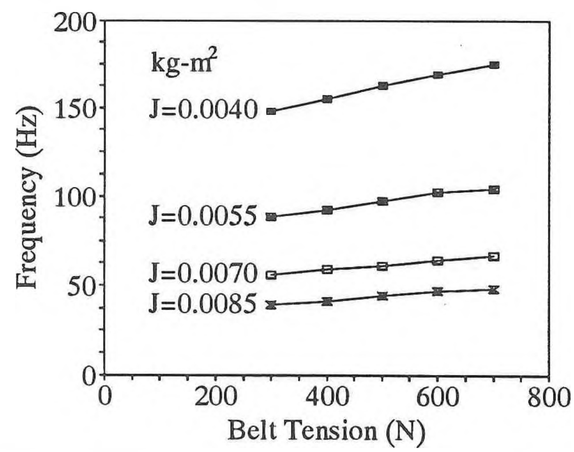



Figure 6 Experimental "wobble" frequencies for the system illustrated in Figure 4.

## CONCLUSIONS

Thirteen different systems were investigated, and the model has been shown to provide good agreement with the measured resonance frequencies, both angular and transverse. The systems tested also exhibited a "wobble" frequency in the direction of the resultant belt span tensile force which was associated with the pulley shaft/bearing compliance. Present also was a "wobble" perpendicular to the resultant belt tensile force. This frequency was affected by the same parameters but was lower due to the decreased compliance in the direction of the resultant belt tensile force. For the two-pulley system with a wrap angle of 180 degrees on the pulleys the difference between the two "wobble" frequencies was quite significant. This difference was much less significant for the three-pulley systems because the angle of wrap was only 120 degrees.

# Reduction of Hand Transmitted Vibration in Rock Drills

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## INTRODUCTION

It has long been known that operators of vibration producing hand tools, such as rock drills, chain saws and chipping hammers, often suffer from tingling, numbness and blanching of their fingers (1). This complex of "vibration-induced white finger" and the various associated symptoms occurring in other systems, in addition to the arterial systems, are known as the Hand-Arm Vibration (HAV) Syndrome, to distinguish them from disorders caused by whole-body vibration (2). The mining industry has recognized its obligation to minimize operator exposure to sources of vibration that could produce HAV syndrome. In particular, efforts have focused on the reduction of hand-transmitted vibration associated with the use of jack-leg rock drills. This type of drill, see Figure 1, is used extensively in the mining industry.

In previous work developed by the authors (3, 4), a research program, was undertaken to determine the effectiveness of using elastomer covered (cushioned) handles in place of the standard steel design. This paper presents an investigation of the long term effectiveness of the elastomer covered handle (cushioned handle) to minimize hand-transmitted vibration under different conditions underground and its applicability to mine production. In this study, five identical cushioned handles were quantitatively and qualitatively tested in different operating mines.

## THE TEST HANDLE

The cushioned handle employs a material known as HD Damped Elastomer, with a durometer of 56. The handle was approximately 4.1 cm in diameter. The cushioned handle was designed to provide "motorcycle grip" control of the jack leg air pressure, that is, the pressure was controlled by twisting the handle. This is in contrast to the steel handle which is fixed, and control of the jack leg pressure is achieved by adjusting a thumbwheel located in one end of the handle.

## FIELD TESTING PROGRAM

The actual test data was obtained from a number of underground operations. In each mine, the standard steel handle used was monitored using the vibration measuring techniques developed by the authors (3, 4) and then replaced by the cushioned handle. The cushioned handle was monitored and the operator was interviewed and evaluated. A questionnaire was also submitted to the operator to develop a data base on acceptance of the drill handle. The handles were used underground under normal production conditions for approximately six months, during which a log book containing all pertinent drill data was filled out daily by the operator. During this time, progress of the field application was closely followed. At the end of this period of application, the cushioned handles were re-evaluated by monitoring the reduction in vibration level experienced by the operators. The operators were interviewed and re-evaluated, and the handles were evaluated in terms of durability, performance and effectiveness (4).

The basic field test approach was to drill a series of holes under simulated production conditions, with the standard steel handle and the cushioned handle. In this manner, the measured vibration provides a direct comparison of the isolation performance of the cushioned handle relative to the steel handle. In all cases, the operators were experienced drillers. They did not use gloves during drilling sequences. The fixture used, fabricated from aluminum, had three accelerometers mounted in three orthogonal directions, as shown in Figure 2.

The upper frequency limit of the measuring system was fixed by the tape recorder at approximately 8000 Hz for direct recording and 900 Hz for fm recording. All data was tape recorded for later laboratory analysis. For each test condition approximately 45 seconds of acceleration data was recorded.

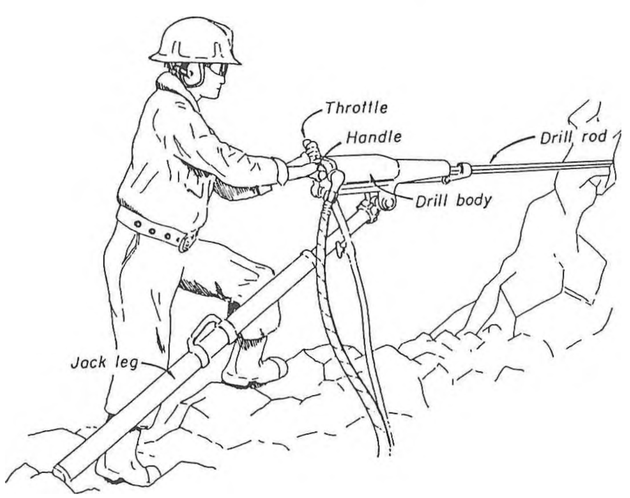


Figure 1 Jack-Leg Rock Drill

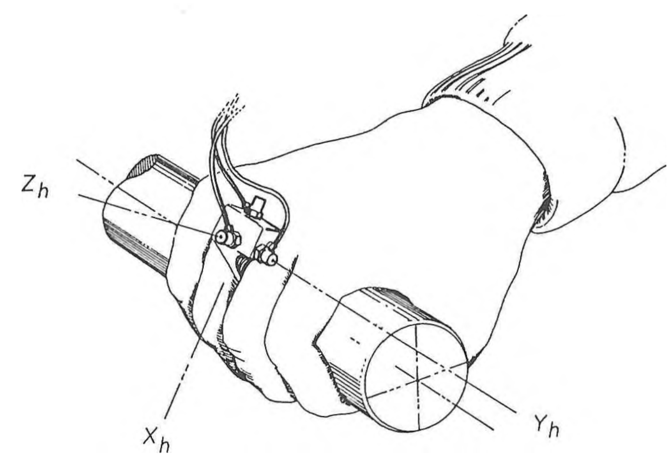


Figure 2 Mounting Fixture And Coordinate Measuring System

## VIBRATION DATA ANALYSIS

The vibration data presented in this paper were obtained using the vibration measurement protocol previously developed by the authors (3, 4) and based upon procedures detailed in ISO 5349-1986 (5). For the purposes of evaluating the long-term field performance of the cushioned handle, it was felt that determination of overall vibration levels and narrowband frequency analysis were sufficient. For this series of tests the analog data were obtained for two different frequency bandwidths. The first was from DC to 900 Hz and the second was from 50 Hz to 8000 Hz. These will be referred to as the *low frequency* range and *high frequency* range results, respectively. In this manner, it is possible to assess both the low and high frequency performance of the cushioned handle. Table 1 summarizes the averaged relative reduction in overall vibration level when the steel handle was replaced by the cushioned handle. This represents the data from the five mine sites, a total of 85 individual drilling sequences, for both initial installation of the handles and after six month's service. Narrowband spectra were obtained for frequency ranges of 0 to 2000 Hz and 0 to 8000 Hz. All spectra were 400-line resolution and were averaged using five ensembles. Figure 3 shows a typical spectrum for the z-direction. The results in the x and y directions were very similar.

## DISCUSSION OF VIBRATION RESULTS

In terms of overall vibration levels, it is apparent that the cushioned handle provides significant reduction in overall hand/handle interface vibration (Table 1). The reduction in overall level is approximately 10 dB in each coordinate direction for the high frequency range. This represents a reduction of the transmitted vibration by a factor of 3. For the low frequency range the reduction is on the order of 5 dB (although the y-direction shows a surprisingly high 10 dB). This confirms that much of the effectiveness of the cushioned handle is based on reduction of higher frequencies. The results for the cushioned handle measured 6 months after initial introduction show no significant change in performance. If anything, there seems to be a slight improvement in performance, although this is not considered to be statistically significant.

A review of the narrowband frequency analyses indicates that the reduction in overall vibration level noted for the cushioned handle occurs as a result of attenuation of higher frequency components of the vibration. This is clearly shown in Figure 3. Generally, significant reduction in acceleration components is noted above 500 Hz. The frequency content of the vibration signals was not noticeably modified for the measurements made 6 months after initial field introduction.

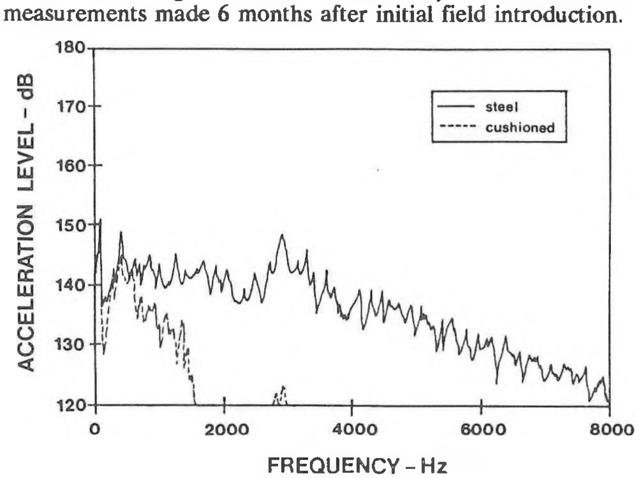


Figure 3 Typical Narrowband Spectra for the Z-Direction

## QUALITATIVE EVALUATION OF HANDLES

A qualitative assessment of the cushioned handles by the drill operators indicated an increase in comfort over the standard steel handle. The assessment program, however, did not indicate preference for the cushioned handle as a number of required design modifications have been identified. The concept of the motorcycle grip to control the jack leg air pressure, was well accepted and given some preference to the standard thumbwheel control. In all mines, the handle was found to be slippery and that, to ensure safe operation, it should be covered with an anti-skid material. An end cap was also found necessary to minimize handle twist. Other minor design modifications were also identified for incorporation in future designs. The above results were based on interviews performed on site and on responses to a questionnaire administered to the drillers at handle installation, and again at the end of the field testing program during re-testing of the handles.

## CONCLUSIONS

Measured hand-transmitted vibration levels were used to evaluate the performance of an elastomer based handle, relative to the standard, solid steel handle. It was found that this cushioned handle reduced overall vibration levels (over the frequency range 0 to approximately 8000 Hz) by a factor of 3. These reductions were consistent for all three coordinate directions. Relative to the steel handle, the cushioned handle was found to provide vibration reductions for frequencies above approximately 500 Hz. Isolation performance improved with increasing frequency, up to approximately 8000 Hz. A qualitative assessment of the cushioned handle by the drill operators indicated some improvement in comfort over the steel handle.

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| DIRECTION | INITIAL (dB) | FINAL (6 months) (dB) |
|---|---|---|
| | LOW FREQUENCY RANGE | |
| X | 5 | 4 |
| Y | 10 | 12 |
| Z | 4 | 5 |
| | HIGH FREQUENCY RANGE | |
| X | 9 | 10 |
| Y | 10 | 14 |
| Z | 9 | 11 |

Table 1 Summary of Relative Reduction in Vibration Level

# HEARING AID RESEARCH AT ONTARIO's HEARING HEALTH CARE RESEARCH UNIT (HHCRU)

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## OVERVIEW

The Hearing Health Care Research Unit (HHCRU) was established in July, 1989 by the Ontario Ministry of Health as part of its new Health System-Linked Research Units Program. This approach to health research links a clinical partner with an academic sponsoring agency in an attempt to increase the relevance and effectiveness of health research.

HHCRU is a partnership between the Otologic Function Unit (OFU), a shared audiology/vestibular service between Mount Sinai Hospital and The Toronto Hospital, and the Aural Rehabilitation Group in the Department of Communicative Disorders at the University of Western Ontario (UWO). Krista Riko, the Director of the OFU, represents the Clinical Partner on the Unit's Steering Committee. The Unit Director is Dr. DG Jamieson of UWO.

The focus of HHCRU is hearing-related assessment and rehabilitative services, with a view to improving the efficiency and effectiveness of these services. The ultimate aim is to minimize the handicap that accompanies hearing loss.

## THE UNIT

Within the linked research unit, the partners focus on the clinical questions that need to be addressed to improve hearing health care service delivery. The grant from the Ontario Ministry of Health covers two-thirds of the basic costs to support the Unit's infrastructure, with the remainder covered by UWO and the OFU. The costs for the individual research projects undertaken by Unit researchers must be provided by specific grants from federal and provincial granting agencies, by contributions from foundations, and through contracts with companies and government departments. Members of the Research Unit also participate in the Ontario Rehabilitation Technology Consortium (ORTC). This Consortium brings researchers, consumers, clinicians, engineers, and industrial partners together to develop, evaluate and make available new assistive devices and technologies.

## CURRENT PROJECTS

The research undertaken by the Unit is organized into several projects which are both identified as important by the Clinical Partner and as feasible and appropriate by one or more of the Unit's researchers. Research is organized into three themes: (1) Development and Evaluation of Diagnostic Technologies and Services; (2) Development and Evaluation of Therapeutic Technologies and Services; and (3) Cost-effectiveness of OFU Operations. Projects relating to amplification systems have been, or are being, undertaken with the first two of these areas. Some examples are provided below.

## SELECTED RESULTS RELATED TO HEARING AIDS

### *The hearing aid test system (HATS)*

Clearly, the accurate measurement of the electroacoustic hearing aid performance is a necessary precursor to all behavioral hearing aid evaluations -- and thus to the improvement of hearing aid design. Unfortunately, standardized hearing aid testing procedures do not provide information of the form and type required either for clinical hearing aid evaluation or for behavioral research to advance the state of hearing aid fitting and application (cf., Jamieson, 1992). Standardized tests were developed to address issues of manufacturing quality control, not to address clinically-relevant questions. Further, standardized tests are unable to characterize the real-life electroacoustic performance of modern, signal-processing hearing aids. In fact, many modern hearing aids **adapt** their electroacoustic characteristics to the acoustic environment -- so that their response in real life situations is often considerably different from that observed in a test situation.

To address these issues, we are developing a suite of "consumer-based" hearing aid testing procedures. Schneider and Jamieson (1993) describe the development and application of the first MLS-based testing procedure for hearing aids. Jamieson and Schneider (1993) report additional aspects of the test suite, including its use to study hearing aid performance in various types of acoustic environments (including performance with real-life background noise bias signals) and the presentation of test results to address specific clinical questions.

### *Development and Evaluation of Advanced Hearing Aid Signal Processing Procedures*

The most common complaint of hearing aid wearers is their difficulty hearing in a background of noise. Hearing aid companies devote considerable effort and money to the design and marketing of procedures which might reduce the impact of such background noise. Our group continues to be involved in research to develop and evaluate new signal processing schemes which might have advantages for hearing impaired listeners (e.g., Jamieson & Brennan, 1992; Jamieson & Cornelisse, 1992).

### *Hearing aid selection methods*

A continuing challenge for audiologists is the prediction of which hearing aid characteristics are likely to produce the maximum benefit for a given individual. The Desired Sensation Level (DSL) method is a systematic approach to electroacoustic selection and fitting, which seeks to ensure that amplified speech will be audible, comfortable and undistorted across the broadest relevant frequency range possible. The theoretical and empirical basis of this approach has been developed over more than a decade of research (cf., Seewald, 1992; Seewald, Zelisko, Ramji & Jamieson, 1992). The current implementation of this system, version 3.1, implements the entire procedure in an easy-to-use program for IBM/AT-compatible computers. Systems are now being used in audiology clinics throughout North America and Europe.

***Development of an advanced signal delivery system to assist in hearing aid selection***

In conventional hearing instrument fitting average adult values are applied in lieu of measurements on each individual. client. Clearly, for any individual hearing instrument user, this approach can result in inaccurate fitting. ***This is a particular risk for children.*** The signal delivery project seeks to develop new systems and fitting protocols to make it easy to personalize fittings for individuals of all ages. A clinical protocol and prototype system for custom hearing aid fitting in adults has been developed and is being incorporated into the hearing aid analysis and fitting system produced by Etymonic Design Inc. In addition, a procedure and prototype system for predicting the performance of hearing instruments in young children has been developed and tested.

## DISSEMINATION AND IMPACT BEYOND THE OFU

HHCRU communicates its findings through published research reports, conference presentations, workshops and special initiatives. The results of some of our projects are now being distributed internationally.

One of our research "products" is the set of systematic procedures for hearing aid selection contained in the DSL approach (Seewald, 1992). We have also developed procedures to assess speech intelligibility performance (e.g., Cheesman, 1992; Gagné, Tugby & Michaud, 1991) and are implementing these for routine use in Audiology clinics.

We have also developed the ***Canadian Speech Research Environment (CSRE)*** software for speech recording, editing, analysis, synthesis, and testing using IBM/PC-compatible computers (Jamieson, Ramji, Kheirallah & Nearey, 1992). Our own uses of ***CSRE*** include the development of speech intelligibility test procedures (e.g., Cheesman, 1992; Cheesman, et al., 1992) and evaluating amplification systems (e.g., Jamieson and Brennan, 1992). ***CSRE*** is now used in laboratories and clinics in 12 countries to support basic and applied research, and clinical evaluation.

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# ADVANCED CLINICAL AUDIOMETRY: MEASURING RESIDUAL AUDITORY CAPACITY USING A STANDARD AUDIOMETER UNDER COMPUTER CONTROL.

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This paper describes a clinical audiometer-based computer controlled Psychoacoustic Test System (PATS). Experimental psychoacoustic test procedures offer the possibility to improve the measurement of residual auditory capacity for diagnostic and rehabilitative purposes. Experimental procedures of interest include measures of frequency resolution and loudness growth, as well as more advanced measures of speech intelligibility and of hearing threshold. Unfortunately, within Clinical Audiological settings, equipment and time constraints have frustrated attempts to introduce such procedures. One reason is that these measurements often require equipment which is not available in a clinical setting, and indeed is generally considered to be unsuitable for such settings. Moreover, these experimental procedures typically require more time than is available in a clinical setting. To address these concerns, we have implemented an approach using a general-purpose, widely-available, audiometer, controlled by a PC. With special-purpose software, this configuration permits us to perform psychoacoustic test procedures which could not otherwise be performed in a clinical setting.

At present the following tests are implemented on the Psychoacoustic Test System: 1) an Adaptive Speech Reception Threshold (SRT), 2) the modified Distinctive Features Differences Test (DFD[m]), 3) a high-resolution, Swept Frequency Audiogram (SFA), 4) measures of Dynamic Range, including, Threshold, Loudness Discomfort Level (LDL), and Growth of Loudness, and 5) a Psychophysical Tuning Curve (PTC) procedure. These procedures have undergone evaluation in our laboratory and have recently been installed in a clinical setting for further evaluation. We anticipate that combinations of these measures should provide additional audiological information which is of relevance to the fitting of personal amplification devices.

**1. Psychoacoustic Test System (PATS)**

The psychoacoustic test system requires a 386 PC with a D/A board, anti-alias filters, and a GSI-16 audiometer. All of the stimuli are stored digitally and are played-back by the PC's D/A board. After passing through the anti-alias filter the test signal goes to the Tape input of the GSI Audiometer. As well, all of the test stimuli are calibrated in dB Hearing Threshold Level (ANSI S3.6-1969) for presentation via TDH earphones or ER3 insert earphones.

The computer program controls all aspects of each test procedure, including presentation of stimuli via remote control of the GSI audiometer, data collection and scoring. In order to simplify use of the PATS program, a graphical interface with drop down menus and dialogue boxes was instituted (see Figure 1). Each client is assigned a numeric code and the clinician can enter personal data (i.e., name and address), as well as audiometric threshold data. The program automatically generates the data file name for each test procedure based upon the client code. The PATS program includes a routine to initialize and calibrate the GSI audiometer. As well, the clinician can specify the test ear and level (dB HTL) of both the test stimuli and noise. The PATS program has a facility to view all test results for a particular client.

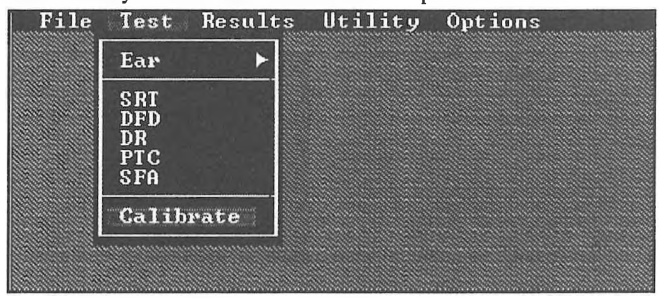


**Figure 1**. Main PATS screen showing test menu items.

**2. Adaptive Speech Reception Threshold (SRT) Test**

The adaptive SRT test procedure is a modified version of the automated SRT and was described by Cheesman (1992). This adaptive procedure provides an efficient, accurate, and reliable estimate of a listener's SRT in quiet or noise. During each trial, one of six spondees is presented. The client indicates which word they heard by selecting one of the six corresponding response alternatives displayed on a video monitor. The adaptive procedure follows the Levitt (1971) rule to converge on the 70% point of the underlying psychoacoustic function. Clinical trials to date have focused on improving the efficiency by reducing the number of trials required for convergence while maintaining accuracy and reliability.

**3. Modified Distinctive Features Differences Test (DFD[m])**

The DFD[m] is a test of speech intelligibility. The test consists of 21 nonsense syllable stimuli. All consonants are presented in the same context (A_IL). The target is the middle consonant of the VCVC word. The test stimuli were produced by four talkers (2 male and 2 female). During each trial one of the 84 stimuli is presented. The client indicates which word they heard by selecting one of the 21 corresponding response alternatives displayed on a video monitor (see Figure 2). The DFD[m] test is scored for percentage of identification errors and the errors are analyzed for type of confusion made (i.e., percentage of voicing, manner, and place confusions). The DFD[m] test of speech intelligibility provides more detailed analysis of speech perception than traditional speech tests, which typically only provide percentage correct. Clinical research to date has focused on the sensitivity of the DFD[m] to differences between hearing aid electroacoustic circuits and acclimatization to amplification for individual subjects, as well as, normative data (Cheesman, Lawrence and Appleyard, 1992).

| | | |
|---|---|---|
| aBil | aCHil | aDil |
| aFil | aGil | aHil |
| aJil | aKil | aLil |
| aMil | ◆ aNil | aPil |
| aRil | aSHil | aSil |
| aTHil | aTil | aVil |
| aWil | aYil | aZil |

**Figure 2.** DFD[m] response alternatives.

**4. Dynamic Range**

The auditory dynamic area is measured using a categorical rating scale for loudness and is based upon a report by Allen, Hall and Jeng (1990). First, the pure tone threshold and loudness discomfort level are measured. Then, a pure tone stimulus is presented at 30 levels between the threshold and LDL in a random sequence. After each presentation of a pure tone pulse at a sound pressure level the client is required to indicate the category which corresponds to the perceived loudness of the sound. The categories range from; nothing, very soft, soft but OK, comfortable, loud but OK, very loud, to uncomfortably loud. Previous clinical trials have focused on the reliability of the threshold and LDL procedures (Gagné, Seewald, Zelisko, Hudson, 1991; Gagné et. al., 1991). Clinical trials will determine the reliability of the loudness scaling procedure, and relate the loudness measures to satisfaction with various types of amplification (i.e., linear vs compression).

**5. Swept Frequency Audiogram (SFA)**

The SFA uses the Bekésy tracking procedure to measure auditory threshold across frequency with greater detail than can be obtained when threshold is only measured at the 8 standard audiometric frequencies. The SFA procedure measures sensitivity to pure tone pulses (presented 2 per second). Frequency is swept logarithmically at a rate of two minutes per octave. Intensity is adjusted at a rate of 2 dB per second. The procedure is based upon a report by West and Evans (1990). Threshold can be measured across 6 octave bands centred at; 250, 500, 1000, 2000, 4000, 8000 Hz. Normative data have been collected on a group of normal hearing subjects.

**6. Psychophysical Tuning Curve (PTC)**

The PTC procedure is based upon reports by Patterson (1976) and Glasberg and Moore (1990). An adaptive procedure (70% rule) is used to measure thresholds for pure tones (500, 1000, 3000 Hz) in notched guassian noise. The lower and upper limit of the notch width varies from 0 to 0.5 (0,0; .2,.2; .3,.3; .5,.5; .3,.5; .5,.3 lower and upper respectively). Normative data are being collected on a group of normal hearing listeners.

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# Auditory Distortion Measures for Coded Speech Quality Evaluation†

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## 1 Introduction

Distortion measure plays an important role in the quality evaluation of coded speech synthesized by a medium or low bit-rate coder. The quantification of distortion involves mapping the signal onto an appropriate domain and formulating a suitable comparison in that domain. In our work, both original speech and its coded version are transformed from the time-domain to a perceptual-domain (PD) using an auditory (cochlear) model. This PD representation provides information pertaining to the probability-of-firing in the neural channels at different clock times. This article proposes two distinct approaches to process these information and measure the degree of distortion in coded speech. The remainder of the article is organized as follows. Section 2 describes the Lyon's cochlear model. Sections 3 and 4 introduce the idea of cochlear discrimination information and hidden Markovian measures; and also study their use in coded speech quality evaluation. Section 5 proposes their use in some applications of speech coder analysis.

## 2 Lyon's Cochlear Model

Time-domain speech is transformed onto a PD where the time-place components become the fundamental bases of analysis. This conversion is performed here using Lyon's cochlear model [1] as shown in Fig. 1. The main features of the model are outlined below (for details, please refer to [2, 3]).

A first-order high-pass filter is designed to simulate the outer and the middle ear effects. The physical structure of the inner ear (cochlea) is modeled by discrete-place ear-filter stages. The basilar membrane (BM) in cochlea is stiff and thin at the basal end (where the sound enters), but compliant and massive at the apical end. Accordingly, each place along the BM resonates most strongly with a pressure wave of a characteristic frequency associated with it. By combining notch filters and resonators, sixty-four ear-filter stages are designed where the band-pass filters have an almost constant Q-factor implying a fixed ratio of the center frequency to the bandwidth for all of them.

Observations of the BM motion indicate that the inner hair cells act as half-wave rectifiers (HWRs) whereas the outer hair cells provide a gain control effect (i.e., amplification or compression). The most important adaptation mechanism in sensory systems is the lateral inhibition by which the sensory neurons reduce their own gain as well as the gain of the others nearby. To emulate this effect, Lyon proposed coupled automatic gain control (AGC) stages. The auditory neurons

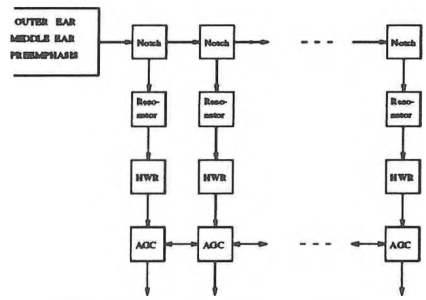


**Fig. 1** Lyon's cochlear model

attached to the hair cells 'fire' (i.e., generate all-or-none electrical spikes) depending on the strength of the gain-controlled signals. These neural firing events are communicated from the auditory system to the brain through neural fibers (termed hereafter as the 'neural channels'). In essence, the normalized (w.r.t. the maximum possible output value) cochlear model output provides the probability-of-firing information (the PD representation) in the sixty-four typical neural channels at each clock time.

## 3 Cochlear Discrimination Information (CDI)

With each of the neural channels, is associated a neural converter which generates impulses based on the probability-of-firing information. These neural converters may equivalently be conceived as a discrete information source with an alphabet of size two, i.e., firing and non-firing. Due to the lack of our knowledge about the exact neural conversion process, the firing/non-firing probabilistic information derived from an original and a coded signal are compared to quantify the degree of distortion. Discrimination information which has emerged as a powerful tool [4] for measuring the 'closeness' of two probability distribution functions is applied here for defining the CDI measure.

This measure evaluates the amount of new information (the increase in neural source entropy) associated with the coded signal when the neural source entropy associated with the original speech signal is known or vice versa [5]. Different variations of this cross-entropic CDI measure, based on the Rényi-Shannon and Havrda-Charvat entropies, are investigated in [3, 5]. These measures are used for speech coder evaluation; it is found that the lower the amount of additional information, better is the signal quality of the coded speech w.r.t. the original one. The effects of different entropies, gain changes, sample delays etc. are also studied in [3]. Finally, a rate-distortion analysis is performed using the Blahut algorithm. State-of-the-art speech coders with rates ranging from 4.8 kbps to 32 kbps are studied from the viewpoint of their performances (as assessed by the CDI measure) with respect to the rate-distortion limits [6]. This analysis has indicated that there is ample scope for improving coder architectures and associated coding algorithms for a specific bit-rate transmission.

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## 4 Cochlear Hidden Markovian (CHM) Measure

In this article, we propose another measure methodology, namely the CHM measure. Here, we attempt to capture the basics of high-level processing in the brain with simple hidden Markovian models (HMMs). We characterize the firing events by HMMs where the order of occurrence of observations and correlations among adjacent observations are modeled suitably. A two-state (one each for firing and non-firing events) fully-connected HMM is associated with each of the neural channels for a fixed-duration, short time segment (in our work, 480 samples). We consider the PD observation process for the entire original speech as a concatenation of many such small segments.

Now, let us consider the observations for any one of the neural channels for a specific time segment. An HMM for any such observation set is defined [7] by describing the complete parameter set of the model given as $\lambda = (\pi, A, B)$, where $\pi$ is the state probability vector, $A$ is the state transition probability matrix and $B$ is a set of two continuous mixture probability density functions (pdfs), each with a few mixtures. Each component of these mixture pdfs is assumed to be a beta density function with values between 0 and 1.

For computing coder distortions, at first, all the HMMs are 'train'ed (i.e., various parameters of the HMMs are estimated) with the pertinent observation vectors corresponding to the original speech segment. There is actually no optimal way of estimating the model parameters from any finite-length observation sequence. The Baum-Welch reestimation algorithm is used to derive the HMM parameters iteratively starting from an initial estimate. The model $\lambda_n^{(o)}$ corresponding to $O_n^{(o)}$, the $n$-th channel observation sequence for original speech, is chosen by maximizing $P(O_n^{(o)}|\lambda_n^{(o)})$. This algorithm is quite powerful as it ensures a monotonic increase in the likelihood with the successive iterations of the algorithm.

Now, let the $n$-th channel PD observations of the $v$-th coded speech be represented by $O_n^{(v)}$. Next, we compute $P(O_n^{(v)}|\lambda_n^{o})$ for all the sixty-four channels (i.e., 'match'ed against the derived HMMs). For simplicity, we assume that the information conveyed through all the neural channels are independent and hence the likelihood probability scores are multiplied to provide a distortion measure (to be precise, a similarity measure). Experimental results have shown that the coded speech signals could be ranked (same as the subjective ordering) by this measure with fair accuracy. The effects of the iteration numbers involved in the reestimation algorithm, the initial estimates of the HMM parameters, the number of mixtures in the pdf etc. are addressed in [8].

## 5 Applications in Coder Analysis

For a low bit-rate speech coder, a proper bit allocation among the coder components (e.g., pitch or formant filter) is vital to achieve a good perceptual quality in the coded speech. In [6], we have described an analysis procedure for determining the pitch frequency by exploring the output space of the cochlear model. The CDI measure form is applied in comparing the outputs for each of the sixty-four neural channels with its delayed version (delayed by $\tau$ samples, $\tau$ up to 160 samples). Subsequently, a one-dimensional cross-entropogram is derived which shows the first significant dip at a $\tau$ value corresponding to the perceptual pitch period.

Next, we consider a code-excited linear predictive (CELP) speech coder which uses three-way split vector quantization for a 16-th order linear predictive coder filter parameters and allows fractional pitch lag values in the pitch predictor. In the present-day analysis-by-synthesis CELP coders, the filter parameters and the codebook entries are selected by minimizing a noise-weighted mean-square error criterion. As a second application, we have investigated the relative superiority of different noise-weighting schemes (e.g., simple noise weighting, codebook shaping filter, enhanced noise weighting).

## 6 Concluding Remarks

A further refinement of the cochlear model and an extensive formal testing with a wide range of linear and nonlinear coder distortions would definitely help improving the measure. Nonetheless, we emphasize that the present framework of comparing the firing/non-firing probabilities could still be maintained. Although we have not attempted to use our measure formulation in a closed-loop fashion in any speech coder, it may very well be possible to use it for 'populating' a codebook in the training phase and/or for 'selecting' an appropriate codebook entry in the transmission phase.


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# TRAINING OF THE ENGLISH /r/ AND /l/ SPEECH CONTRASTS IN KOREAN LISTENERS

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Native speakers of Korean have particular difficulty with the distinction between English /r/ and /l/ contrasts (e.g., Borden, Gerber & Milsark, 1983). While English has two separate labels for the sounds we call /r/ and /l/, the Korean language groups both such sounds into a single category. The Korean /r/ tends only to occur intervocalically and it occurs in the form of a rhotic flap which is articulated with the tip of the tongue making a quick flipping contact against the alveolar ridge (Pyun, 1987). The Korean /l/ sound occurs only postvocalically, and only in a "light" form (Borden, et al., 1983). Korean orthography does not distinguish these sounds (i.e., these sounds do not constitute a phonemic contrast; Borden, et al., 1983). The Korean listener may therefore be unable to hear the difference between the English /r/ and /l/ sounds.

In the present study we sought (1) to measure the abilities of young, adult, native Korean speakers to discriminate between the English /r/ and /l/ speech contrasts; and (2) to evaluate the possibilities that these listeners could be taught this perceptual skill using procedures similar to those used by Logan, Lively and Pisoni (1991).

## METHOD

### *Subjects*

Five native speakers of Korean (3 male and 2 female) between the ages of 21 and 31, participated in this experiment. All were native to Korea, had resided in Canada between 5 weeks and 1.5 years (M = 0.8 years), and were enroled in an English-as-a-second-language (ESL) program.

### *Procedure*

Our stimuli were a superset of the words used by Logan, et al. (1991). There were nine minimal pairs (e.g., rock-lock) within each of five phonetic environments: initial singleton (IS), initial consonant cluster (IC), medial (M), final consonant cluster (FC) and final singleton (FS). Nine native English speakers (4 males,5 females) produced each of the eighteen words, contrasting /r/ and /l/ in the phonetic environments mentioned. Speakers six to nine had incomplete stimulus sets, so that there were approximately 162 individual words in each phonetic position, with 775 stimulus words used in total.

All aspects of stimulus sequencing and presentation, response recording, and experimental control were carried out using the experiment generator and controller utility ***ECoS*** contained in the CSRE 4.0 software package (Jamieson, Ramji, Kheirallah & Nearey, 1992). Stimuli were presented to listeners over Etymotic Research ER-2 insert phones. All phases of testing and training took place in an IAC, double-walled, acoustically-shielded room.

### *Design*

Two matched-groups were formed when assigning subjects to a control or training group. All subjects were tested individually.

***Pretest and Postest***. The pretest preceded any training and the posttest followed it; otherwise both tests were identical. Each of the five phonetic environments was tested separately. Following each stimulus presentation, subjects indicated whether an /l/ or /r/ sound was heard, by pressing the "L" or "R" key on a computer keyboard. No feedback was provided. Each test required < 75 minutes.

***Training***. Subjects were trained with the two phonetic environments we predicted would be the most difficult: the initial singleton and initial consonant cluster positions. Subjects in the training group received 15 minute training sessions for 15 days. Subjects in the control group only received the pre- and posttests. All subjects continued to attend their ESL program throughout the experimental procedures.

Training used a two-alternative forced-choice identification procedure, with feedback. Only words produced by five of the nine talkers (4 male,1 female) were used during training.

## RESULTS AND DISCUSSION

Our analysis addressed three questions: (1) what effect did phonetic environment have on intelligibility? (2) to what extent were different talkers differentially intelligible? and (3) to what extent did training improve intelligibility performance? These questions are discussed in turn.

### *Phonetic Environment*

Overall, the final consonant cluster (FC) position was the most difficult ($M$=66%) and the final singleton (FS) position was the least difficult ($M$=83%). The initial singleton (IS; $M$=73%), initial consonant cluster (IC; $M$=71%) and medial (M; $M$=73%) positions were of equal difficulty (cf., Figure 1).

Sounds in final position tend to be longer in duration which may assist perception (Logan, et al, 1991; Sheldon and Strange, 1982). We were therefore surprised that Korean listeners tended to find the final consonant clusters to be the most difficult to identify. This may reflect differences in native-language transfer, but future research is required on this topic. The high accuracy of Korean listeners with sounds in the initial singleton position is similar to performance found with Japanese listeners.

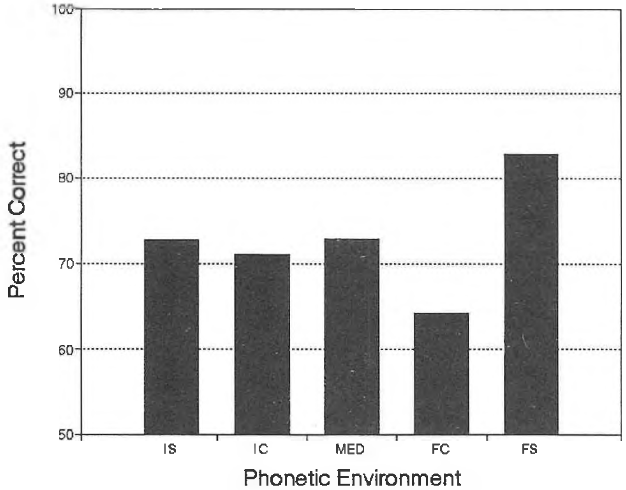


*Figure 1.* Performance in each of the five phonetic environments. Data are based on pretest performance, and are collapsed across all five listeners.

### *Speaker Differences*

Talkers were differentially understandable, with talkers 6 and 4 being the most intelligible (cf., Figure 2). Studying only Talkers 1-5, Logan, et al., (1991) had found Talker 4 to be the most intelligible; to our knowledge, Talkers 6-9 have not been studied previously.

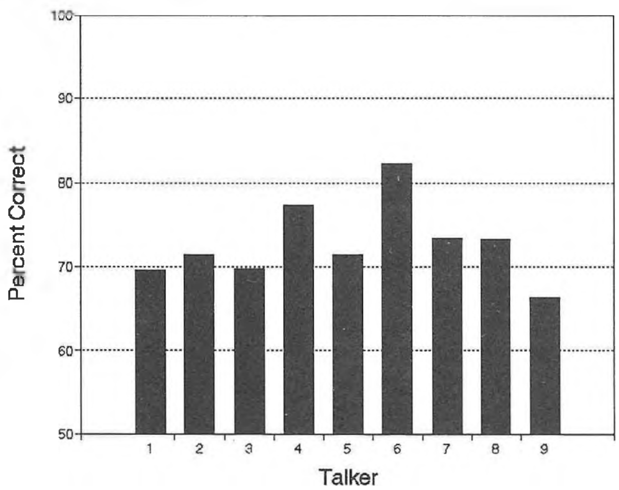


*Figure 2.* Mean intelligibility of tokens spoken by each of the nine Talkers. Data are based on pretest performance, and are collapsed across all five listeners.

### *Effects of Training*

Mean accuracy increased for each of the subjects who received training (from 61% at pretest to 83% at posttest, for one trained listener and from 86% to 94% for the other; cf., Figure 3). Smaller, but statistically insignificant improvements were also found in identification performance of the control group subjects. AH showed the greatest increases in accuracy for the initial consonant cluster, medial, and final consonant cluster positions. CP improved most with the final consonant clusters and showed equivalent increases in accuracy for all other phonetic environments.

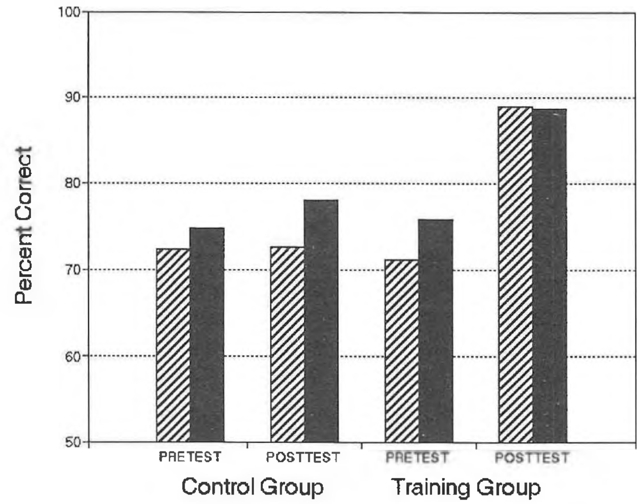


*Figure 3.* Mean intelligibility performance during pretesting and posttesting for listeners in the trained and control groups. Cross-hatched bars indicate performance with those tokens spoken by the talkers used in training trials; solid bars indicate performance with those tokens spoken by talkers used only in the pretest and posttest.

## DISCUSSION

Although increases in performance may be obtained through regular attendance in ESL programs, the present identification training procedure significantly increased subjects' abilities to identify English /r/ and /l/ tokens. Training generalized to tokens spoken by unfamiliar talkers, and to phonetic environments not used in training.

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# USING THE CANADIAN SPEECH RESEARCH ENVIRONMENT (CSRE) TO TEACH SPEECH PERCEPTION TO UNDERGRADUATES

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This paper describes four computer-based learning (CBL) sessions that were designed for and used in the speech acoustics and perception sections of a course in Speech Science offered by the Department of Communicative Disorders at the University of Western Ontario. Students enrolled in the course have undergraduate degrees in diverse disciplines, including the physical, social, and biological sciences, arts and humanities. Consequently, students vary greatly in their previous experiences with audio and computer hardware, computer software packages, and in their knowledge of speech perception and physical and speech acoustics.

All of the laboratory experiences described below were developed with the Canadian Speech Research Environment (CSRE; Jamieson, Ramji, Kheirallah and Nearey, 1992). CSRE is a comprehensive, microcomputer-based system designed to support speech research using IBM/AT-compatible microcomputers. CSRE provides a powerful, low-cost facility to support speech research using mass-produced and widely-available hardware. The software has been used in several graduate and undergraduate courses since 1989. Cheesman and Alsop (1990) reported on initial experiences with CBL modules used with this system. The most recently released version, CSRE4.0[1], is currently being used for CBL applications. The software includes speech sampling, editing and replay functions; a selection of frequency analysis procedures and colour two- and three-dimensional displays; formant and other parameter extraction and tracking; parametric speech (KLATT80) and non-speech synthesizers, and a comprehensive experiment generator and controller that support behavioural testing using several common testing protocols.

For each of the laboratory sessions, students are given a printed handout containing basic instructions and guidelines for the laboratory session. Students prepare a written report, following a set of directed questions included on the handout.

**1. Analysis of misarticulated speech sounds**

In this session, students digitally record and spectrographically analyze their own productions of normal and misarticulated speech sounds that they have encountered in their clinical phonetics course. The lab assignment is structured as a step-by-step sequence that results in a tutorial introduction to the use of the CSRE sampling, editing and analysis functions. Figure 1 illustrates a spectrographic comparison, using auto-correlation, of a normal and misarticulated "play".

**2. Vowel perception laboratory**

This three-part laboratory experience focuses on the relationship between formant frequencies and vowel perception and allows students to test their own perception. Acoustical analysis of test items allows students to confirm the observations they have made in the perceptual testing.

I. Effect of varying formants 1 and 2. A two-dimensional grid

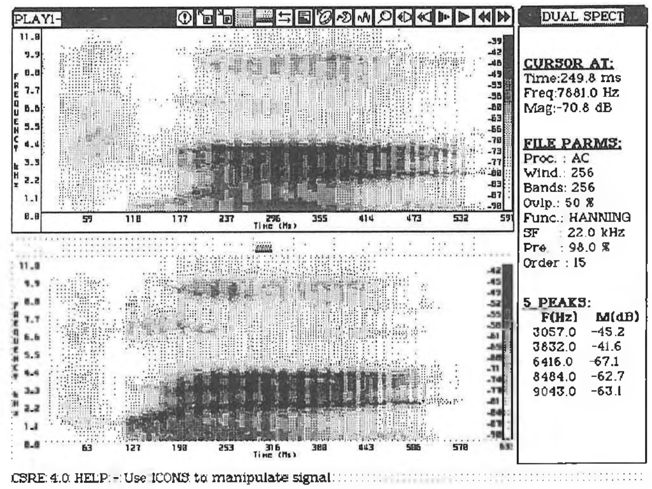


**Figure 1. Dual spectrogram of normal and misarticulated /plei/**

of F1 and F2 space is presented. Each point on the grid corresponds to a synthetic vowel which the student can listener to repeatedly. Students transcribe each of the sounds and then examine the pattern of transcription with the formant space. In the example in Figure 2, the sound corresponds to an /ɛ/-like vowel with a first formant centred at 550 Hz and a second formant at 1650 Hz.

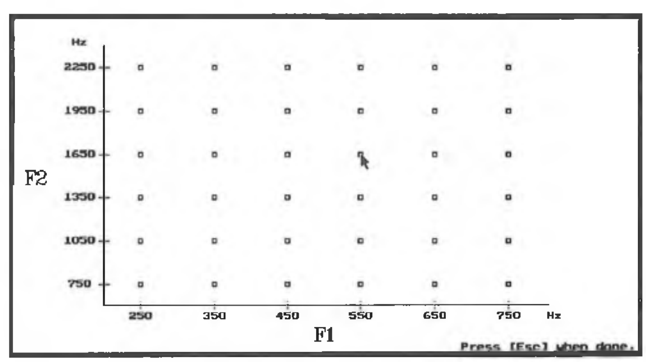


**Figure 2. Example of monitor display during vowel perception module.**

II. Non-English vowel sounds. A series of foreign-language (German) vowel sounds, spoken in various phonetic contexts, are presented. Students select one of four vowel categories to assign to each vowel and are given correct answer feedback. The task is repeated three times, so that students can examine their initial performance and any learning effects. The exercise underscores the dominant role of the native language categories in the perception of even relatively steady-state foreign contrasts.

III. Acoustical analysis of vowel sounds. The German vowels used in the task described above are analyzed with respect to formant frequencies. Performance on the F1/F2 vowel space and German vowel identification tasks are re-analyzed using this information.

**3. Stop/semi-vowel perception laboratory**

I. Acoustic and perceptual analysis of consonants. Students analyze, display and listen to the medial consonant portion of four digitized nonsense words taken from the modified distinctive features difference test (DFD[m] - Cheesman, Lawrence and Appleyard, 1992). Using the partial playback feature of CSRE, students can listen to and attenuate selected segments of the words in order to study the perceptual consequences of removing selected acoustic information.

II. Creation of stop consonant percept. Using the editing features of CSRE, students insert a silent interval after the fricative portion of the nonsense word /ʌsIl/ to observe the perceptual effect. This silence is typically perceived as the oral closure associated with stop consonants and the dominant percept is /ʌspIl/. Figure 3 shows the waveform editing process used to isolate the end of the frication noise.

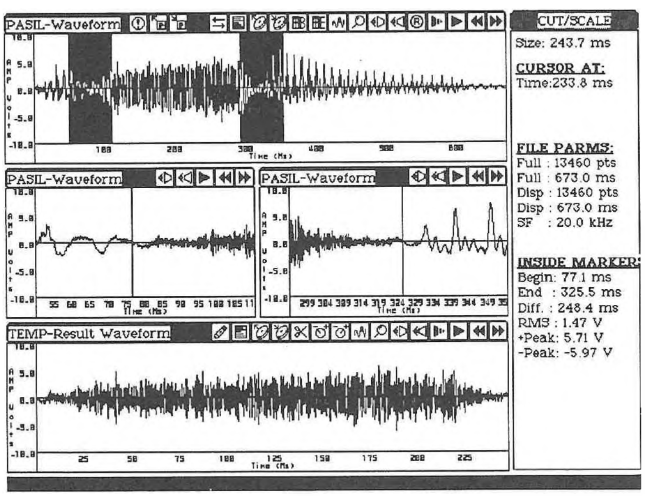


**Figure 3. Waveform display of the nonsense word /ʌsIl/ during editing.**

**4. Fricative perception and context effects**

I. Context-conditioned voiceless fricatives. Synthetic stimuli consisting of nine bandpass noises followed by /i/ and /u/ vowels are identified by students as /s/ or /ʃ/ and the effect of the vowel on the identification of the fricative continuum is examined. Students also compare their own data with that of other class members to observe individual differences in category boundaries and context effects.

II. Comparison of synthetic speech acoustics to natural tokens. Spectrographic analyses of natural and synthetic /s/ and /ʃ/ sounds are made and the results are compared. Figure 4 displays such a comparison. Students are encouraged to examine alternative spectral representations of the same signal. For example, they may compare the representation obtained with traditional linear predictive analysis to the cone-kernel spectrogram.

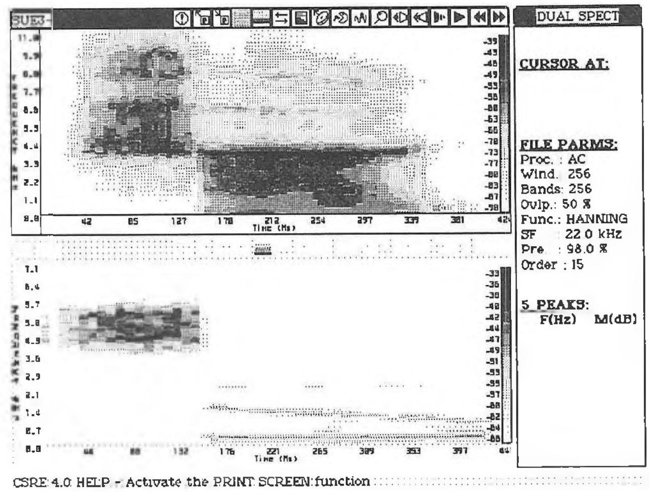


**Figure 4. Spectrograms of naturally-produced and synthetic tokens of /su/.**


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[1]CSRE4.0 software is available from the Hearing Health Care Research Unit at the University of Western Ontario.

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# Clinical Application of Computer-Driven Methods for the Assessment and Treatment of Speech Perception Disorders

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The relationship between speech perception and speech production errors in children with impaired phonological skills has been a controversial issue in the practice of speech-language pathology for many decades. During the past 20 years speech-language pathologists have largely ignored the perceptual abilities of their phonologically impaired clients as a consequence of vigorous critiques of clinically available methods for assessing and treating speech perception disorders (6,10). As Locke observed in 1980 (6), "much of our thinking on this issue is weakened by the fact that efficient perception measures have not been in general use. Because perceptual questions, historically, have not properly been asked in the laboratory, it is not obvious that perceptual questions currently should be answered in the clinic (p. 432)." Fortunately, since that time, perceptual questions are being properly asked in the laboratory by researchers who are borrowing methods from the study of the perceptual abilities of adults learning a second language. These methods include the use of : synthetic or digitally altered speech stimuli for both the assessment and treatment of perceptual difficulties; microcomputers to present large numbers of stimuli to subjects for identification, with motivating feedback when appropriate; and, acoustic analysis of speech samples in order to carefully relate speech perception to speech production abilities in the same subject. This paper describes a series of studies designed to apply these methods to the assessment and treatment of speech perception errors in phonologically impaired children who misarticulate fricatives. The contributions of this line of research to phonological theory and clinical practice will be summarized.

Rvachew & Jamieson (8) constructed 2 synthetic speech continnua, one contrasting the sounds /s/ and /ʃ/, the other contrasting /s/ and /θ/, in the prevocalic position of a single syllable real word (i.e., seat-sheet, or sick-thick). Normal speaking adults, normal speaking 5 year olds, and phonologically impaired 5 year old children identified multiple tokens of the /s/ - /ʃ / stimuli, while normal speaking adults, normal speaking 7 year olds, and phonologically impaired 7 year old children identified the /s/ - /θ/ stimuli. In both cases, 16 repetitions of each of the 7 stimuli in the continuum were presented for identification by pointing to the appropriate picture . The normal speaking children were not as reliable as adults in their identifications, but clearly perceived the stimuli in each continuum as belonging to 2 distinct sound categories. Some of the phonologically impaired children identified these stimuli in a manner similar to their normal speaking peers, but others were completely unable to categorize the stimuli appropriately. Some of these children placed all of the stimuli in the /s/ category while others responded in essentially random fashion to all of the stimuli in the continnum.

As a logical follow-up to this study, the effect of speech perception training on children's speech production errors was investigated (5). The "fading technique" (3) was used in 5 single subject studies to train phonologically impaired children to identify synthetic syllables containing a prevocalic fricative (/ʃ/ misarticulators listened to /sa/ and /ʃa/ syllables and /s/ misarticulators listened to /sa/ and /θa/ syllables). After 2 hours of sound identification training, small but significant improvements in production performance were observed for 3 children who demonstrated poor identification performance prior to training, and improved identification performance following training. Improved production performance as a consequence of sound identification training was not observed for a subject who failed to learn the identification task, nor for a subject who demonstrated good identification of the stimuli prior to training.

Another study was undertaken to enhance the clinical utility of the fading technique by incorporating a computer game format which was expected to increase the child's enjoyment and compliance during sound identification training. In this study, natural speech stimuli were used so that the target sound (/ʃ/) could be contrasted with a range of error sounds. Twenty-seven phonologically impaired children who misarticulated /ʃ/ in a variety of ways were assigned to 1 of 3 groups: Group 1 listened to correct versions of the word "shoe" contrasted with a variety of incorrect productions recorded from other phonologically impaired children (i.e., /tu/, /tʃu/, /su/, etc.); Group 2 listened to the words "shoe" and "moo", both produced by an adult talker; and Group 3 listened to the words "cat" and "Pete", recorded from an adult talker. Children in Groups 1 and 2 were required to point to a picture of a "shoe" when they heard a correct version of this word, and to an "X" when an incorrectly produced version of this word was presented. The response alternatives for Group 3 children were pictures of a cat and an "X". The auditory training stimuli were presented over headphones, and visual reinforcement was presented via the monitor, using the Experiment Control System of the Canadian Speech Research Environment (4). The children enjoyed the computer game format and readily agreed to complete 60 consecutive training trials during each 15 minute session of sound identification training. Each child in all 3 groups also received traditional sound production training, targeting the /ʃ/ sound, concurrently with sound identification training, for 6 consecutive weekly sessions.

Following training, the children's sound production abilities were assessed by asking them to name five objects representing words containing the /ʃ/ sound in prevocalic position, and to produce a single isolated /ʃ/ sound. The isolated /ʃ/ productions were submitted to acoustic analysis using CSRE (4). The results indicated that children who were taught to identify correct and incorrect versions of the word "shoe" progressed further in production training, produced more correct /ʃ/ sounds during the spontaneous naming task, and produced better quality /ʃ/ sounds with more appropriate acoustic characteristics, in comparison to control children who learned to identify the words "cat" and "Pete".

These findings are consistent with the results of other studies which have employed synthetic speech to demonstrate that phonologically impaired children have concomitant speech perception difficulties (e.g., 1). The use of acoustic analysis to describe children's phonological systems and to document progress in therapy is also occurring with increasing frequency (e.g., 2,7). However, the studies described above constitute the only published support to date for the efficacy of speech perception training in the treatment of sound production errors, and continued research on this topic is clearly required.

The primary theoretical implication of this line of research is that some phonologically impaired children possess a system of underlying phonemic contrasts which differs from that of their adult models (c.f., 9). With respect to a given sound contrast, 3 patterns of perceptual and productive error may occur: both members of the sound contrast may be absent from the child's system; both members of the sound contrast may belong to a single phonemic category in the child's system; or, both members of the contrast may exist in the child's system but are differentiated in terms of nonstandard acoustic cues. All 3 patterns involve a correspondence between the child's perceptual performance with the sound contrast and the child's articulation of the sounds.

When treating such a child, the clinical task is to help the child resolve the mismatch between his or her underlying system and the adult system. This can best be achieved when an accurate description of the child's perceptual and productive abilities is obtained. The studies cited above have shown that perceptual abilities cannot be adequately assessed with natural speech stimuli, especially when presented live-voice; rather, effective perceptual testing must involve careful control of stimulus characteristics, which can be obtained by synthesizing speech or digitally altering natural speech recordings. In addition, it has been shown that children's speech production skills are best described using a combination of phonetic and acoustic analyses.

To paraphrase Locke (6), perceptual questions have been properly asked, and answered, in the laboratory; we have now reached the point where perceptual questions can and should be answered in the clinic. The clincal application of the research methods discussed above has been made possible by the availability of relatively inexpensive and user friendly systems for speech synthesis and analysis, such as CSRE (4). Currently, the primary impediment to clinical application is a shortage of clinicians with the necessary technical knowledge, and a lack of funding for research in clinical settings which would serve to expand and enhance the clinical utility of these technologies.

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# APPLICATION OF SPEECH RECOGNITION/CLARIFICATION TECHNOLOGY TO DYSARTHRIC SPEECH

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The aim of this study is to develop and evaluate a communications system which uses speech recognition as a method of speech input and clarification to produce intelligible synthesized speech as the output.

Specifically, this study will address the differences between the intelligibility ratings of unclarified (natural) dysarthric speech and speech clarified with a speech recognition and voice output system. To do this, differences in the intelligibility of words versus phrases across both the clarified and unclarified speech conditions will be investigated. Further, differences in the intelligibility ratings by naive versus experienced listeners will be investigated.

## BACKGROUND

The ability to produce speech that is understandable to normal listeners is denoted as the intelligibility of speech. Dysarthric speakers produce speech characterized by distorted vowels and imprecise consonants, and so, often have low speech intelligibility, (SI). This inhibits all forms of verbal communication. Even though some dysarthric speakers are difficult or impossible to understand, they are still capable of producing a variety of sounds and word attempts with some level of consistency[1,2]. Coleman and Meyers[2] found dysarthric speakers to have relatively consistent speech, even though it was less intelligible than normal speakers. If dysarthric speakers are consistent, and capable of differentiating sounds, then speech recognition technology coupled with voice synthesizers provides a potential solution to the problem of low speech intelligibility.

Speech recognition systems fall basically into two categories, speaker dependent and speaker independent. These two categories are further subdivided into two classes, namely continuous and discrete. Typically, the most expensive systems are the speaker independent continuous recognition systems which require no training period and exert no restrictions on the user. Less expensive and more widely available systems, are the speaker dependent discrete systems. It is this type of system that is employed in this study.

Studies have demonstrated that speech recognition systems recognize words of impaired (dysarthric) speakers better than human listeners. Stevens and Berstein[3] found that the speech recognition system recognized single words for five deaf speakers more accurately than did human listeners. Human listeners recognized 5% to 74% of the single word utterances while a speech recognition system recognized 75% to 99%. Carlson and Berstein[4] tested 50 disabled speakers comprising of hearing impaired and cerebral palsy individuals (dysarthric speakers). Word recognition was compared between naive human listeners and the speech recognition system and was found to be better for the speech system. This type of technology appears to have great potential for dysarthric speakers.

Intelligibility of synthesized speech has also been studied. Green, Logan and Pisoni[5] evaluated the intelligibility of eight text-to-speech systems. Naive listeners transcribed words produced by the speech synthesizers resulting in an average intelligibility for synthesized speech systems of 85.7% with DECtalk achieving the best performance of 96.7%. Mitchell and Atkins[6] found intelligibility rates of 63% and 66% for the Echo II Plus and EvalPac respectively. The intelligibility of words and sentences of natural and synthesized speech was compared by Miranda and Beukelman[7]. They found the intelligibility of three DECtalk voices to be statistically equivalent to natural speech. In general, sentences were found to be more intelligible than single words. It appears that with specific synthesized speech devices, replacing unintelligible speech with intelligible synthesized speech is a viable alternative.

## METHOD

### *Subjects*

Upto ten individuals with dysarthric speech will be recruited for inclusion in the study. The subjects will be over ten years of age and have a differential diagnosis of Dysarthria confirmed by qualified speech language pathologists. For inclusion in this study, the subjects' speech must be rated at greater than 20% intelligibility on the Computerized Assessment of Speech Intelligibility (CAIDS, Yorkston,K., Beukelman,D., Traynor,C., 1984) by naive listeners.

### *Procedure*

The subject's SI will be determined by the transcription method of rating the CAIDS test. The subjects undergo two 1 hour training sessions to use the speech recognition system and then train the system to recognize their voice. The subjects will then proceed to the testing session where they will imitate a standardized list of phonetically balanced words and phrases with and without the speech recognition/clarification system. The sessions will be videotaped and the naive and experienced listeners will review the videotapes at a later date. Intelligibility ratings of the subjects based on the listeners experiences will be computed by comparing what was transcribed by the listeners to what was said by the subject.

*Equipment*

For the purposes of this study, the DragonWriter 1000 speech recognition development system is being used as the speech recognition system. Software application programs have been developed in conjunction with the speech recognizer to train the subject on how to use the system as well as enabling the user to train the system to his or her voice. Development is underway on a software application that will present phonetically balanced words and phrases to the subject so that he/she may repeat them for the speech recognition system when prompted by the system. The DragonWriter 1000 speech recognition board takes the vocalizations of the user and turns them into text. These text strings are then output to the voice synthesizer, which in this case is DECtalk. DECtalk is a text to speech device which includes a choice from 9 predefined voice patterns including male, female and children's voices and one user definable voice. For this study, voices will be chosen that reflect the age and character of the individual.

## SUMMARY

By training the speech recognition system to interpret the utterances of the dysarthric speaker, and converting these instructions into synthesized speech, it is hoped that the whole process of communication for a dysarthric speaker will improve. Currently, there are problems with the communication rate of contemporary devices. There is a need to find ways to enable users of such a system to communicate in a more timely and effective manner. By converting speech via speech recognition technology, into highly intelligible synthesized speech, it is hoped that some of these issues can be addressed.

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# Three Dimensional Transient Sound Intensity Measurements for Comprehensive Room-Acoustic Evaluation

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## INTRODUCTION

Contemporary objective room-acoustics indicators are based on the capture and subsequent analysis of room impulse response but subjective criteria are also influenced by the spatial distribution of sound energy. The spatial distribution of sound energy is usually not considered due to lack of an efficient, accurate and easy to perform measurement procedure. This study introduces a 3 dimensional sound intensity measurement for obtaining spatial information of the sound field in an enclosure. The measurement technique is described and its merits and potential are identified. Specular and diffuse sound field quantification is also introduced.

## DIRECTIONAL DISTRIBUTION OF SOUND : STATE OF THE ART

Sound Directionality in rooms has been studied and assessed by a number of workers [1,2,3,4], however the measurement procedures do not allow ready or consistent evaluation. To resolve these difficulties a new measurement system is proposed. *CBS-RAIMS* is a new measuring system developed to fully evaluate sound quality in an enclosure by measuring a number of potential useful room-acoustic indicators and to provide directional information in a manner which allows ready interpretation for both sound quality assessment and diagnostic purposes. The description of the measurement system hardware and software components and data processing have been presented in Refs. [5,6]. In this study the measurement method and merits are elaborated.

## THREE DIMENSIONAL TRANSIENT SOUND INTENSITY MEASUREMENT

The method utilizes sound intensity measurement from three microphone pairs arranged in cartesian coordinates or one pair in three successive orientations to establish 3-D intensity vectors. The sound field can then be visualized on an energy directional basis versus arrival time. The filtered sets of impulse responses X-X, Y-Y and Z-Z in each selective octave or third octave band allow three orthogonal intensity vectors components to be digitally processed in the time domain using a finite difference approximation approach given by the equation [7]:

$$I_n(t) = \left(\frac{1}{2\rho_o d}\right)\left[p_1(t)+p_2(t)\right]\int_{-\infty}^{t}\left[p_1(\tau)-p_2(\tau)\right]d\tau \quad (1)$$

where,

$p_1$ = sound pressure of channel 1, Pa.
$p_2$ = sound pressure of channel 2, Pa.
$\rho_o$ = air density, kg/m$^3$
$d$ = spacing between the microphone pair, m

When processing equation (1) the full transient record length for each set is used to avoid erroneous results from segmentation and time windowing procedures. The resulting instantaneous intensity vectors are then used to obtain specular sound reflection directions; however if one is only interested in sound energy direction the envelope intensity technique can be used; resulting in a smooth sound intensity components.

To yield a visually detailed 3-D image of incoming sound intensity vectors at the listener location on a time base, the 3-D intensity vectors are then calculated and a conversion from rectangular to spherical coordinates is also made. The directional information is identified in five principle directions with respect to the listener; these are front, back, right, left and up; the contribution of each can now be separately examined. The full directivity patterns can be displayed with time of arrival or viewed from different angles with respect to the listener as shown in Figure 1. The signal can be further processed to reveal left and right, up and down reflections in isolation. In practice the graphical output of vectors is color coded for ease of interpretation.

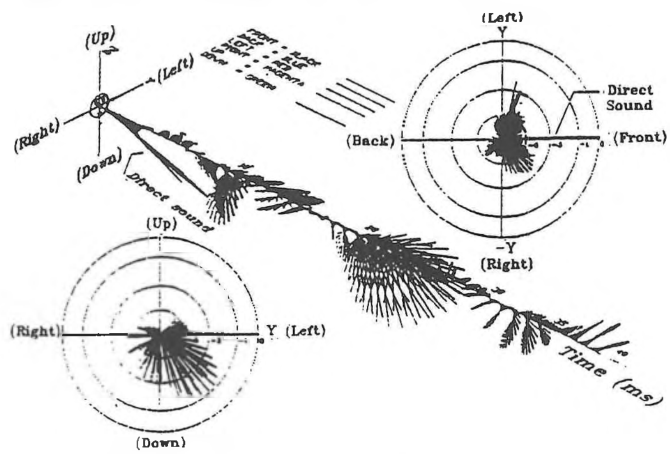


*Figure 1.* Example Directional Information (at 500 Hz).

It must be accepted that any instantaneous intensity vector is in fact an instantaneous resultant, thus directional components can be hidden for objective diagnostic purposes; this fact can be a problem, however by employing an instantaneous pressure intensity index, a measure of correct directional sensing can be established.

When using the intensity technique, the accuracy of direction sensing is influenced by the microphone pair channel phase mismatch; this causes a distortion of the probe directional characteristics. Further, directional characteristics vary with frequency and are usually problematic in the low frequency range. To validate the method and investigate its accuracy, known reflective surfaces and sound source positions in anechoic and reverberant environments have been examined. Measurements are found useful particularly for identifying specular reflections in the early reflections period.

## SOUND DIFFUSENESS QUANTIFICATION

Measurements in existing halls show that the field is unlikely to be diffuse. To what extent the sound should be diffuse and how that could be judged or quantified is now of concern. Knowledge of directional distribution would identify the degree of sound diffuseness exhibited at a listener location.

The diffusion of a sound field can be defined and viewed from different perspectives. These can be examined utilizing the now available directional information obtained by *CBS-RAIMS*.

### Sound Diffuseness : Visual Examination

Initially, with the current measurement system the sound directional distribution in a given solid angle of interest can be isolated and visually examined from different views for the uniformity of sound reflections. If the sound is fully diffuse, the envelope of the incoming intensity vectors tends to be smooth with no significant irregularities or sudden dips otherwise irregularities of their level and coherence of directional incidence dominates.

### Sound Diffuseness : Net Sound Energy Flow

An ideal diffuse sound field is 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.

To quantify the sound field diffusion with respect to acoustic net energy flow, a "Directional Diffusion", *DD* is proposed :

$$DD = \frac{1}{\Delta t}\int_{t}^{t+\Delta t} \vec{I}(t)\,dt \Big/ \int_{0}^{T} |\vec{I}(t)|\,dt \qquad (2)$$

where, the numerator is the mean energy flow, $w/m^2$ in a given direction, that is the magnitude of intensity which would result if all intensity vector components on a given directional axis, received within the time period t to $\Delta t$, were then divided by the time period over which the assessment is considered. The denominator involving $|I(t)|$ is a measure of the total energy passing through the measurement point over the total impulse response period $T$. The resulting sign of *DD* indicates the direction of the net energy flow. If $DD_x$, $DD_y$ and $DD_z$ are the cartesian component Directional Diffusion calculated from equation (2), the Spatial Diffusion (*SD*) will be given by:

$$SD = \sqrt{DD_x^2 + DD_y^2 + DD_z^2} \qquad (3)$$

*DD* and or *SD* if close to zero indicates that overall intensity vectors are evenly distributed directionally and in magnitude about the measuring point, suggesting that the sound field is diffuse. High *DD* or *SD* values imply a highly unidirectional sound field. The window $\Delta t$ may be fixed for variable t, or $\Delta t$ may be variable. The instantaneous sound intensity can be windowed by a successively sliding rectangular window of some interval to obtain successive values of *DD* and *SD* from the end of the direct sound; this would indicate the change of the mean energy flow with time. An example of $DD_x$ is shown in *Figure 2* for measurement in a reverberation chamber at 500 Hz.

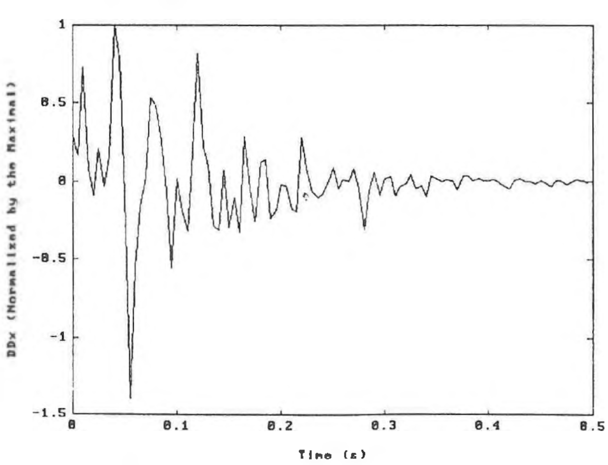


*Figure 2.* $DD_x$ at 500 Hz in Reverberant Environment.

### Sound Diffuseness : Balanced Spatial Sound Energy

A general indication of sound wave homogeneity may be inferred from examining received spatial sound energy ratios at frequencies of interest, excluding the direct sound, in six principle directions with respect to the listener (i.e. front , back, right, left, up and down). *Figure 3* shows a comparison of sound energy ratios (at lower frequency octave bands) received in six directions for measurements in reverberant field with the direct sound energy. Other measures of diffusivity will also be discussed.

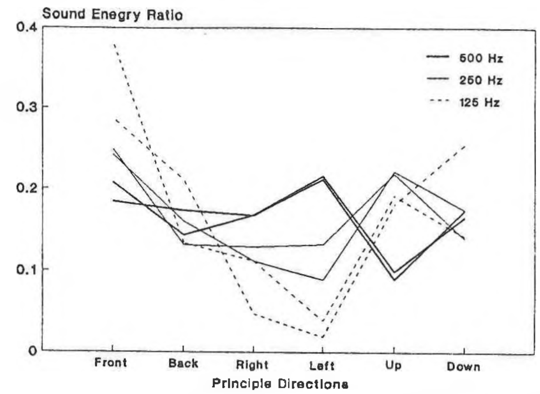


*Figure 3.* Comparison of Spatial Sound Energy Ratios at Low Frequency Octave Bands.

## CONCLUSIONS

A 3-D transient sound intensity measurement method for obtaining directional information in enclosures is reported and shown to be effective. Contributions of particular boundary surfaces to early sound reflections can then be identified, isolated and examined for diagnostic purposes. Sound Diffuseness quantifiers are proposed; example results are given. Further work is necessary to directionally or diffusively qualify the newer room-acoustic indicators. Measurements are currently under way to investigate relationships between objective measures and spatial energy distribution.

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# Determination of Flanking Transmission and Field Sound Transmission Loss in Wood-Framed Constructions Using Intensity Methods

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## Introduction

The method of acoustic intensity is used to determine the presence and magnitude of flanking transmission in a common double wood stud construction. This work was conducted as part of a joint research project with Canada Mortgage and Housing Corporation. Two construction specimens are considered. The first, without a construction fault, represents the ideal case in which there should be no flanking, (base condition -- See Figure 1). The second specimen has a potentially common construction fault. The plywood floor decking of the upper rooms is continued across the party line, (see Figure 2). The results of the intensity measurements are presented for the various surfaces. Difficulties encountered when using the intensity technique in the presence of flanking transmission are also discussed.

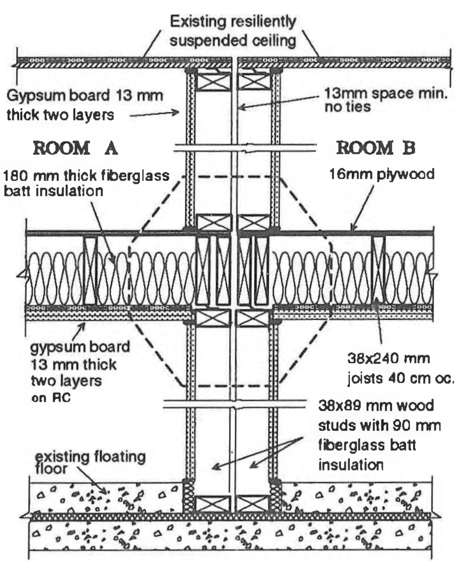


*Figure 1: Section of base condition specimen at the party wall.*

## Measurement Technique

It is generally assumed that conventional measurement procedures involving either a P-P or P-V intensity probe will provide an accurate measure of an individual surface's radiated sound power. In fact, significant difficulties can be encountered when measuring the intensity of a surface that is physically connected at right angles to a much more energetically radiating surface. Consider measuring the

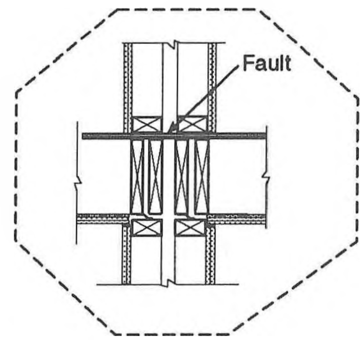


*Figure 2: Section showing the construction fault.*

transmission loss (TL) of the party wall shown in Figure 1 using the intensity technique when the floor is the dominant radiator, (i.e., under extreme flanking caused by the construction fault, Figure 2). Figure 3 shows the party wall transmission loss as computed from the measured intensity with and without the floor masked. The masking consisted of 1/2 inch thick gypsum board over 5/8 inch thick plywood separated from the measurement surface by 2 inch thick fiberglass batt insulation. A resilient air-tight joint between masking and measurement surfaces proved to be critical. The joint was made by using closed cell neoprene pipe lagging placed over the edge of the masking panels butting the measurement surface. The measurement surface was 4.54 m wide and 2.40 m high. Ninety-five points were used to sample the surface; using 10 columns over the width and 11 rows over the height. The probe was located at 6 cm from the measurement surface. The integration time was at least 60 seconds for each measurement point and the receiving room had at least 25 $m^2$ of 50 mm thick rigid fiberglass absorbing material. The results indicate that the P-P intensity probe is incapable of determining the normal radiated intensity of the measurement surface when there is an adjacent non masked radiating surface coupled at right angles. This has a significant impact on the usefulness of the method under extreme flanking conditions. Under these conditions, masking should be considered. In all subsequent intensity test data presented here, flanking surfaces were masked.

## Party Wall Intensity

Figure 4 shows the measured TL between rooms A and B. From the figure, it can be seen that the fault affected the TL of the party wall as derived from the transmitted acoustic intensity. In terms of a single number rating the sound

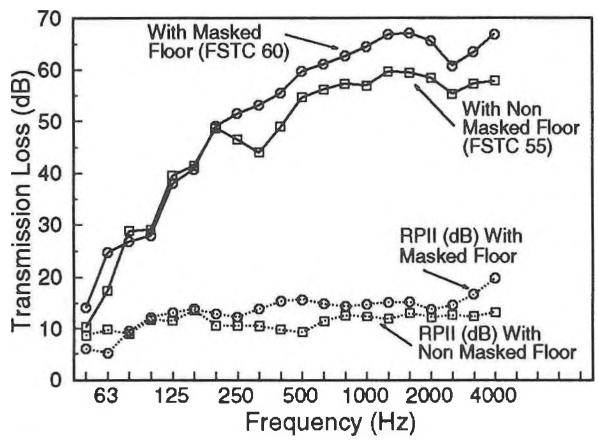


*Figure 3: Party wall transmission loss obtained from the acoustic intensity with and without the floor masked.*
RPII=$[L_p - |L_I|]_{\mathrm{calibration}} - [L_p - L_I]_{\mathrm{measurement}}$.

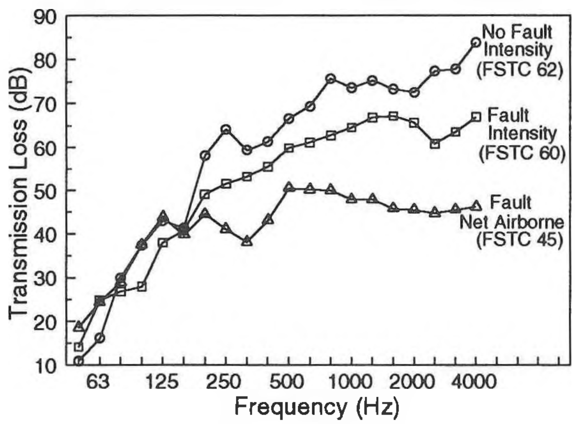


*Figure 4: Party wall transmission loss obtained from the acoustic intensity with and without the fault. For the case of the fault, the net airborne sound insulation (which includes the flanking paths) is included for comparison.*

insulation dropped from FSTC 62 to FSTC 60. Comparing the net airborne sound insulation with the fault (FSTC 45) to the TL of the party wall with the fault (intensity method, FSTC 60), it is evident that for frequencies greater than 200 Hz, the party wall provides much greater sound insulation. Thus, there is at least one very significant flanking path between rooms A and B, and the party wall is probably connected to the flanking path, but it is not the predominant radiator.

**Floor**

Figure 5 shows the measured radiated sound power for the floor and the party wall of room B when room A is the source. It is evident that the floor is the predominant radiator of acoustic energy for frequencies greater than 200 Hz. Figure 6 shows the average radiated intensity of

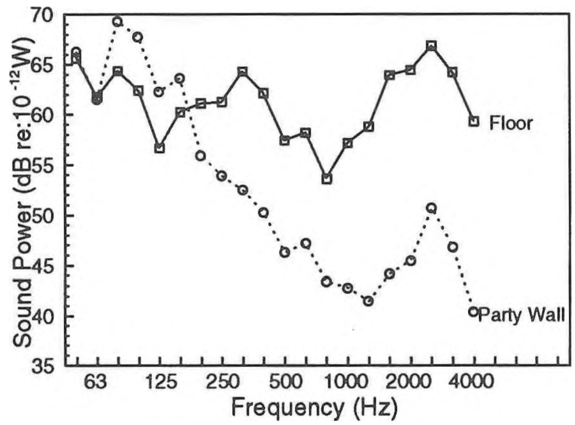


*Figure 5: Measured total radiated sound power from the party wall and the floor.*

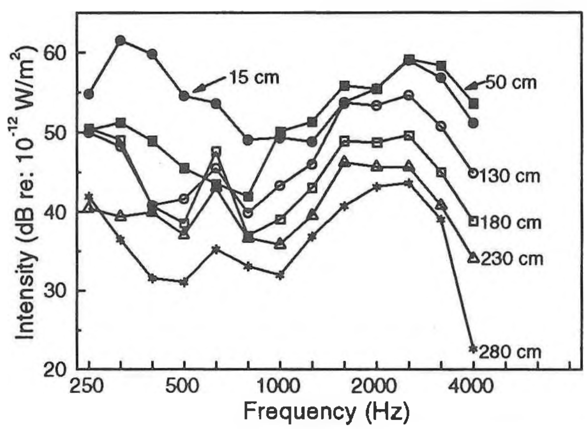


*Figure 6: Measured floor intensity averaged along lines at the indicated distance from the party wall.*

the floor along rows at the indicated distances from the party wall. Eight points were used in each row. There is a strong gradient in the radiated energy, indicating that the floor is more energetic near the party wall. This is especially true for frequencies greater than 800 Hz. The gradient in the radiated energy of the floor suggests that the floor is connected to the flanking path at or near the floor/party wall intersection.

**Conclusions**

1. Under conditions of extreme flanking, intensity methods fail to correctly isolate the normal component of the measurement surface. This can cause significant errors in the measured sound power and hence the transmission loss. For this reason, the use of masking is suggested when measuring next to a significant radiator.

2. Intensity methods allow for sound power measurement of surface sub-areas. This can be very useful to identify flanking paths.

3. Flanking transmission can significantly degrade and in severe cases control the net sound insulation.

# Effect of Perforated Sheets on Rectangular Silencer Performance

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## 1.0 INTRODUCTION

HVAC duct silencers use porous materials such as glass wool, rock wool etc. to absorb the sound. Mathematical models for predicting the insertion loss of the silencers are a critical part of an optimum design and models to evaluate the insertion loss of absorptive rectangular silencers were developed by different researchers [1, 2]. The porous absorbers of the silencers are usually covered with a perforated sheet metal to protect the material from the main flow. Cummings [3] considered the effect of the perforated sheets and showed that the effect is usually negligible. The conventional prediction methods [4] therefore ignore the effects.

The effect of the perforated sheets is considered in this paper. The effect is included in the numerical model as a jump condition. The present results are limited to rectangular silencers only. Preliminary results of a parametric study is presented.

## 2.0 PERFORATED SHEET MODEL

The typical rectangular silencer consists of baffles with porous absorbers covered with perforated sheet metal and the main flow is confined to the open airway between two baffles. A cubic Galerkin finite element method [1] is applied to solve the equations governing the sound propagation in the rectangular duct. This method was chosen because it is easily extendable to aircraft engine nacelles with shearing airflow. The porous material is modelled as bulk reacting in the finite element method. The predictions include the effect of all propagating modes. The loss due to the absorbing material is used to approximate the insertion loss of the silencer. Further details concerning the model are described in Reference 1.

The finite element method applies two conditions, continuity of pressure and continuity of velocity across the absorbing material - open airway boundary. The perforated sheet metal provides a discontinuity across the material - open air way boundary. The sheet is treated as a limp sheet of finite surface density [4]. The pressure continuity is recovered for zero surface density. The continuity conditions are:

$$\begin{aligned} v_- &= v_+ \\ p_- - p_+ &= Z_p . v \end{aligned} \tag{1}$$

where $v_-$ and $v_+$ are the velocity across the boundary, $p_-$ and $p_+$ are pressure across the boundary, and $z_p$ is the acoustic impedance of the perforated sheets.

The impedance of the perforated sheets can be determined from the resonator design formulations of Ingard [5]. Maa [6] established a low frequency expression for the perforated impedance. Cummings [3] provides a general expression for the perforate impedance and is used in this paper. The perforated impedance is given by,

$$z_p = \frac{ik_0 z_0}{Q} \left[ \delta \left(1 + \frac{z_a k_a}{z_0 k_0}\right) + d \right] \tag{2}$$

where $k_0$ and $k_a$ are the open air way and porous material wave numbers, $z_0$ and $z_a$ are the open air way and porous material characteristic impedances, d is the thickness of the perforate, Q is perforate open area, and $\delta$ is the perforate mass end correction.

The end correction $\delta$ has been approximated for square and hexagonal array of holes by Cummings [3] using the theoretical curves of Ingard [5]. For square array of holes $\delta$ is given by,

$$\begin{aligned} \delta &= 0.850\, r_0 \left(1 - \frac{2.34 r_0}{l}\right), \quad 0 < \frac{r_0}{l} < 0.25 \\ \delta &= 0.668\, r_0 \left(1 - \frac{1.90 r_0}{l}\right), \quad 0.25 < \frac{r_0}{l} < 0.5 \end{aligned} \tag{3}$$

where $r_0$ is the radius of perforate holes and l is the distance between the holes. The coefficients of 2.34 and 1.90 in Eq. (3) become 2.52 and 2.0 respectively for an hexagonal array of holes.

The boundary conditions given in Eq. (1) are appropriately entered in the finite element code and the insertion loss at any given frequency is evaluated from the attenuation rate of all the propagating modes.

## 3.0 RESULTS AND DISCUSSION

The finite element code was used to evaluate the attenuation rate for the plane propagating wave for a simple rectangular duct tested by Cummings [3]. The duct had two baffles of 5 cms thick with the 10 cm wide open air way. The porous material had flow resistance of 20,000 MKS Rayls per metre. The perforated sheet had a porosity of 14% and was 1.27 mm thick. The plane wave attenuation of Cummings [3] matched with the results of the finite element scheme. The above comparison was used as a test on the results of the finite element scheme.

Test data for a set of conventional rectangular silencers were available. The insertion loss results for three octave band of frequencies are presented in Table 1. The perforated sheets of the silencers were constructed from standard 22 gage sheet metal with the following properties: 0.853 mm thick, 23% open area, 2.38 mm diameter holes and the holes were staggered at 4.76 mm spacing. The results clearly show that the sheets used in conventional rectangular silencers had minimal influence on the insertion loss results even though the perforate open area was less than 40% for Cummings had predicted that sheets with open area less than 40% may drastically reduce the high frequency performance.

Perforate open area was varied for the standard 22 gage sheet metal perforates and the results are shown in Tables 2 and 3. The flow resistance of the porous absorber was 20,000 MKS Rayls for the results of Table 2 and the flow resistance of the absorber for the results of Table 3 was 8000 MKS Rayls. The thickness of the perforate sheet was 0.853 mm. It is seen that for the thin perforate sheet used for the parametric study, the effect with perforate open area is negligible both for the dense (20000 MKS Rayls) as well as for the loose (8000 MKS Rayls) porous absorber.

The perforate open area was set at 10% for the 2.76 mm diameter hole sheet and the thickness of the sheet was varied from 1 mm to 9 mm. The results are shown in Tables 4 and 5 for two different silencer overall width. The results for 0.305 m wide silencers are presented in Table 4 and the results for 0.610 m wide silencer are given in Table 5. The results show that the effects predicted by Cummings [3] are realised with thicker sheets with low perforate open areas. The performance at 1000 Hz and 4000 Hz octave bands reduce drastically with increasing thickness of the perforate sheets. The results seem to hold for the two sizes of the silencer sets. The drastic degradation in the performance of the silencers at high frequencies is true only with thick perforate sheets.

Additional parametric study is under way with hole diameter and results will be reported subsequently.

## 4.0 CONCLUSIONS

The effect of perforate sheets used in conventional rectangular silencers was studied and the results were presented in this paper. The sheets used in HVAC system silencers (0.853 mm thick sheets with 23% open area) were shown to have minimal impact on the acoustic performance of the silencers. The thin sheets, even with small open areas, had minimal influence on the overall performance. The study showed that perforate sheets degrade the high frequency performance of the silencers only if the sheets are reasonably thick.

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**Table 1. Insertion Loss Results in dB**

| | | Insertion Loss, dB Octave Band | | |
|---|---|---|---|---|
| Silencer Size | Condition | 250 Hz | 1000 Hz | 4000 Hz |
| 305 mm | No Perf | 11 | 29 | 16 |
| | Perf | 11 | 29 | 16 |
| 457 mm | No Perf | 13 | 29 | 7 |
| | Perf | 13 | 29 | 7 |
| 610 mm | No Perf | 28 | 50 | 17 |
| | Perf | 28 | 50 | 17 |

NOTE: 22 gage perf. sheets (23% open area, 2.38 mm dia.)

**Table 2. Silencer Insertion Loss with Perf. Open Area.**

| | Insertion Loss, dB in Octave Band | | |
|---|---|---|---|
| Perforate Open Area | 125 Hz | 1000 Hz | 4000 Hz |
| 0.1 | 10 | 22 | 5 |
| 0.2 | 10 | 22 | 6 |
| 0.3 | 10 | 22 | 6 |
| 0.5 | 10 | 22 | 6 |

NOTE: Silencer is 610 mm wide, 203 mm thick baffles, 20000 MKS Rayls material, 0.853 mm thick perfs

**Table 3. Silencer Insertion Loss with Perf. Open Area.**

| | Insertion Loss, dB in Octave Band | | |
|---|---|---|---|
| Perforate Open Area | 125 Hz | 1000 Hz | 4000 Hz |
| 0.1 | 14 | 22 | 6 |
| 0.2 | 14 | 22 | 6 |
| 0.3 | 14 | 22 | 6 |
| 0.5 | 14 | 22 | 6 |

NOTE: Silencer is 610 mm wide, 203 mm thick baffles, 8000 MKS Rayls material, 0.853 mm thick perfs

**Table 4. Silencer Insertion Loss with Perf. Thickness.**

| | Insertion Loss, dB in Octave Band | | |
|---|---|---|---|
| Perforate Thickness | 125 Hz | 1000 Hz | 4000 Hz |
| 1 mm | 10 | 22 | 5 |
| 3 mm | 10 | 20 | 8 |
| 5 mm | 10 | 17 | 6 |
| 7 mm | 10 | 15 | 4 |
| 9 mm | 10 | 13 | 2 |

NOTE: Silencer is 610 mm wide, 203 mm thick baffles, 20000 MKS Rayls material, 10% open perfs

**Table 5. Silencer Insertion Loss with Perf. Thickness.**

| | Insertion Loss, dB in Octave Band | | |
|---|---|---|---|
| Perforate Thickness | 125 Hz | 1000 Hz | 4000 Hz |
| 1 mm | 15 | 38 | 26 |
| 3 mm | 15 | 37 | 22 |
| 5 mm | 15 | 32 | 13 |
| 7 mm | 15 | 27 | 7 |
| 9 mm | 15 | 21 | 4 |

NOTE: Silencer is 305 mm wide, 102 mm thick baffles, 20000 MKS Rayls material, 10% open perfs

# POWER MEASUREMENT FOR AN ACOUSTICALLY PULSED JET FLOW

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Previous work[1] showed that acoustic modulation of the dilution-air jets of a small tubular combustor improved performance in that selective and progressive control of the exit plane temperature distribution was possible. Further work on acoustically pulsed free-jet mixing[2,3] showed that the entrainment coefficient of the jet, and therefore mixing, could be considerably increased, by up to 6 times. The effects correlated well with the amplitude of the jet pulsation velocity $U_e$. This new work therefore attempts to relate the jet pulsation velocity amplitude to the acoustic power modulating the jet flow $\dot{W}_{af}$ thereby leading to a more practical and direct correlation between mixing and acoustic power. This article is an abbreviated version of the Inter-Noise 93 paper, Leuven, Belgium, by the same authors.

**Experimental.** Figure 1 shows the apparatus whereby the modulating acoustic power and the corresponding exit air jet velocity pulsation amplitude $U_e$ were measured for a flanged open ended copper tube and for a range of exit nozzle sizes, over a range of acoustic driver powers and exit jet velocities, at optimum excitation frequencies determined by frequency response testing. The acoustic power was measured using two Kistler 601L pressure transducers, transversely mounted in boss positions 2 and 3 in the air feed tube, air flow tube or driver tube as required. The measuring diaphragms were mounted flush with the tube bore, and unoccupied bosses were sealed by steel plugs. The exit air velocity pulsation amplitude was measured by a hot-film anemometer mounted on the centre-line in the exit plane. The air velocity in the flow tube $U_f$ was varied from 0 to 8 m/s giving an exit jet velocity up to 78.6 m/s whilst the driver input electrical power W was varied up to a safe maximum of 32W.

From the pressure measurements, corrected for amplitude and phase differences, the acoustic power $\dot{W}_a$ was calculated from Munjal's Equn. (3.18)[4] using the computer and dual channel analyser.

Acoustic excitation, at a particular frequency, pulses the jet velocity $U_j$ at its orifice exit plane causing the jet flow to develop a train of toroidal vortices. The entraining action of the travelling toroidal vortices is the primary mechanism of the acoustically augmented mixing process.

Figures 2 and 3 show typical measured acoustic power results. The influence of air flow velocity is shown to be of secondary importance. The three acoustical powers are approximately proportional to driver electrical power, but only for the 6.35 mm and 12.70 mm dia. nozzle systems is the sum of flow and feed tube acoustic powers approximately equal to that of the driver tube. For the open ended flow tube (maximum acoustic powers), the sum of flow and feed tube acoustic powers is less than that of the driver tube acoustic power. Also, the choked valve feed tube acoustic power is dissipated. This suggests that the acoustic power sum is less than the driver tube acoustic power by extra dissipation at the "T" junction. Furthermore the acoustic powers progressively decreased as the flow tube exit diameter was reduced by the nozzles.

Figure 4 presents typical jet pulsation velocity amplitude measurements versus the flow tube acoustic power $\dot{W}_{af}$ using the correlation of Ref. 2, where D is the jet orifice diameter and $\rho$ is the jet flow density at the orifice exit plane. There is a distinct jet Reynolds number ($Re_j$) effect. Figure 5 summarises all the data and shows a geometrical effect. These figures show the jet excitation or pulsation strength $U_e/U_j$ to be proportional to $(\dot{W}_{af}/\rho D^2 U_j^3)^{1/2}$, the acoustic power number. The correlation is well behaved giving confidence in the acoustic power measurement technique, and ought to be independent of the acoustic driver used. Because of the nozzle contraction ratio (lack of geometrical similarity) the acoustic intensity at the orifice ($\alpha U_e^2$) is increased relative to that of the open flow tube. Thus all the data can be brought into approximate coincidence, except for Reynolds number effects, by scaling the acoustic power number by the ratio "flow tube bore diameter/jet orifice diameter". Because of the correlation, the acoustic power through $U_e/U_j$ strongly correlates with pulsed jet mixing factors[2,3] eliminating the need for difficult pulsation velocity measurements.

**Conclusions.** The two pressure transducer technique for the measurement of acoustic power has been successfully developed for a tube air flow. The jet flow modulating acoustic power measured was approximately proportional to the driver electrical power, and was a maximum 8% of the driver power for the open ended tube and progressively decreased as the exit diameter was reduced by the nozzles. Significant acoustic power dissipation at the "T" junction was indicated. The % jet pulsation velocity amplitude was proportional to (flow modulating acoustic power)$^{1/2}$, inversely proportional to exit orifice diameter and inversely proportional to (jet velocity)$^{3/2}$. The well behaved correlation was approximately unique, except for Reynolds number effects, when the nozzle contraction ratio was accounted for. Because of the correlation acoustic power relates well with pulsed jet mixing factors eliminating the need for difficult pulsation velocity measurements.

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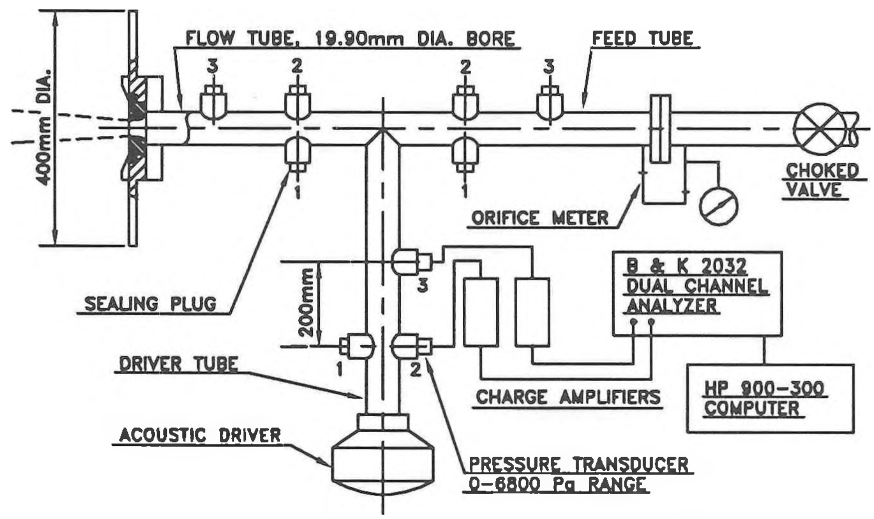


Fig. 1 Apparatus for Acoustic Power and Exit Pulsation Velocity Measurement.

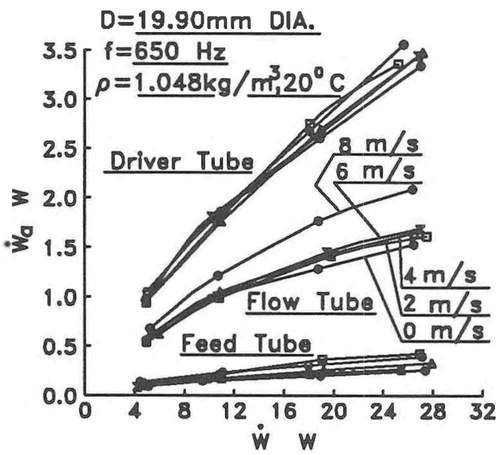


Fig. 2 Acoustic Power Versus Driver Input Electrical Power, Open Ended Tube.

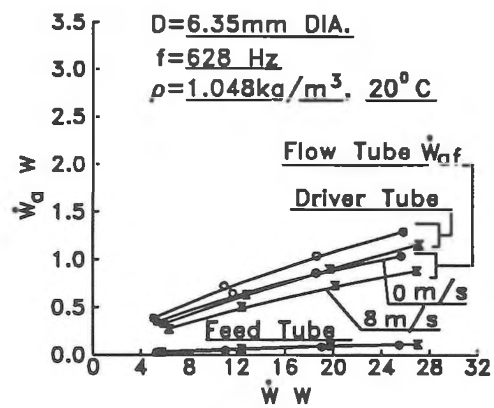


Fig. 3 Acoustic Power Versus Driver Input Electrical Power, 6.35mm Dia. Nozzle.

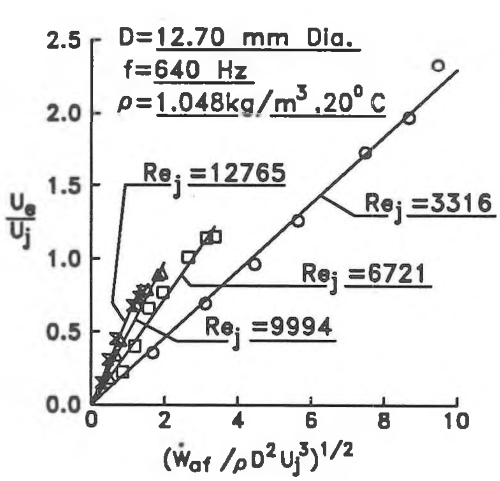


Fig. 4 Pulsation Strength Versus Acoustic Power Number, 12.70mm Dia. Nozzle.

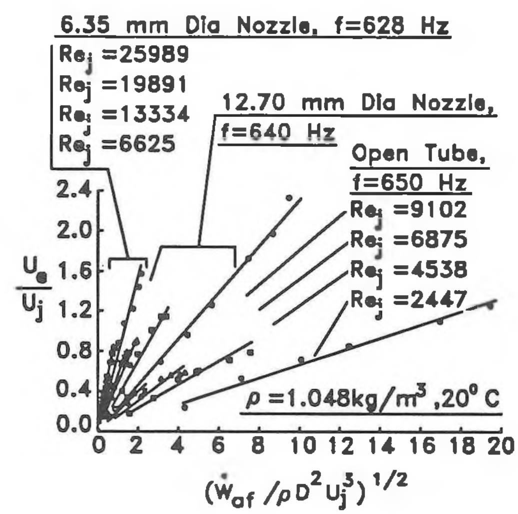


Fig. 5 Pulsation Strength Versus Acoustic Power Number for all Data.

# City of Toronto Noise By-law Review and Amendment

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## Introduction

By-law 44-75, Respecting Noises (the Noise By-law) was passed by Toronto City Council on February 20, 1975 for the purpose of controlling unnecessary or unreasonable noise. It was designed to eliminate noise at its source essentially by controlling noisy activities.

As a result of almost 20 years of By-law enforcement, it was recognized that the By-law needed to be updated to conform with current community standards, changes in land uses and enforcement realities. Valcoustics Canada Ltd. and Jade Acoustics Inc. were retained to assist in the review in order to deal with new sources, clarify wording and identify technical "loopholes" where found and also provide information on the experience and practice of other similar municipal jurisdictions.

During the consultation process, it became apparent that more would be gained by looking at the issues dealt with during the course of enforcing the By-law in addition to the wording of specific clauses.

## Loud Music

Of an average of 1300 complaints registered per year at the Noise Control Branch, approximately 35% are from loudspeakers, music etc. usually late at night from entertainment establishments. The sporadic or unpredictable nature of these sources of noise usually require that the Metro Police enforce the By-law. This poses a problem because noise is not considered a priority and therefore the response time is not ideal. In order to improve enforcement procedures a 24 hour answering service is proposed so that callers can be advised of the best course of action depending on their complaint. Other measures being considered are for improvements with licensing in the case of entertainment establishments which would require the installation and maintenance of acoustical insulation so that music is contained within the premises. This could be a condition of a business and/or liquor license.

## Source Equipment

This type of equipment includes air conditioners. heat pumps, ventilation fans, garage door openers etc. Approximately one quarter of all complaints are due to this type of equipment, primarily during warmer weather. Usually equipment requires servicing or minor alterations in order to bring about compliance.

The proposed revision to the By-law is to allow an increase of up 5 dBA over the ambient from the previous 2 dBA, to reflect the actual experience in the field. The feasibility of requiring permits to install air conditioning devices was reviewed and it was determined that an additional level of bureaucracy would be created which may discourage applicants, thereby defeating the purpose; aside from the added administrative burden and cost to the City. Educating installers/suppliers in proper installation techniques would better serve this function.

## Construction Noise

Construction is an inherently noisy activity and is primarily controlled by restrictions on times and in some cases equipment types. Generally, complaints are due to noise prior to 8:00 a.m. or after 9:00 p.m. on weekdays or at any time on weekends, especially Sunday. In other cases renovation projects involving heavy jackhammering, causes most concerns for people who are at home during the day. Whenever possible, contractors are requested to use the least disruptive equipment available, especially in densely populated or other sensitive areas such as hospitals and hotels.

For some construction activities there is flexibility in the enforcement of the By-law. For example, some after-hours construction can take place provided nearby residents are informed in advance. In other situations such as with concrete trowelling it is important to note that due to unforseen circumstances, such as inclement weather, a concrete pour may take place and the power trowelling may carry on beyond the hours provided for in the By-law. In these situations electric trowels are required and where blower heaters are needed during night time hours acoustical barriers are required around the site. For the most part, the construction industry has made extra efforts in meeting the City's guidelines.

## Powered Property Service Equipment

Equipment such as lawnmowers, leaf blowers, and chain saws are regulated according to hours of use. Gas powered leafblowers have become a particular concern primarily because of the unique tonal quality of the noise emission from these machines. Maximum permissable sound levels have been stipulated as of January, 1992 -70 dBA at a distance of 15 metres. The industry has been encouraged to follow U.S. guidelines established some years ago in California. With the proper use of this equipment as well as improvements in sound output levels, fewer complaints are expected.

## Security Alarms

According to the Metropolitan Toronto Police, who are responsible for investigating alarms, approximately 95% of all security alarms are false. The difficulty in enforcement is in locating the owner of a building during sleeping hours. In order to deal with this problem from a more pro-active approach the Police instituted an Alarm Response Policy which, among other things, requires a permit for a security alarm. This has resulted in a substantial decrease in false alarms, presumably due to improved systems and installation. The Police have made a submission to the Provincial government to make requirements for licensing a legislated requirement. In addition a public education program is being developed to advise people of the City's Noise By-law requirements jointly with those of the Police.

**Summary and Conclusions**

Overall, substantial changes to the By-law have not been made as a result of the review. Sections have been re-grouped to make wording clearer and easier to interpret and enforce, particularly for Police Officers not trained in noise assessment.

A Noise By-law, no matter how comprehensive, will not eliminate noise in the City. It does provide a benchmark which describes what the City expects of its citizens. Often informing someone who is creating a disturbance is sufficient to obtain compliance. Where voluntary compliance is not forthcoming the By-law also serves as an effective tool for enforcement. In every possible instance noise disputes are mediated so that voluntary compliance is obtained at much less cost to the City.

# City of Toronto
# Equipment Noise Suppression Programme

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In 1973 Toronto City Council considered a Noise Control Study and directed that a programme of noise control be adopted to begin a guiding principle of which noise would be controlled at its source.

As a result, the City of Toronto Noise By-law was passed in February 1975. A key part of the noise control programme was the adoption of a policy that the City would purchase acoustically designed equipment for use in its Operations and Sanitation Sections. The objective of this policy was to establish guideline noise levels to encourage private contractors to use the quietest available equipment on road construction, utility upgrading and maintenance work within the City boundaries. Further steps taken towards achieving a quieter environment included a social perception survey, and ongoing testing of construction machinery to establish practical noise emission levels for various kinds of construction equipment such as air compressors, jack hammers, backhoes etc. The information obtained resulted in an amendment to the Noise By-law in 1987, which included guideline noise emission levels for construction equipment. These levels were considered stringent by the construction industry, however any cogent opposition was was not presented, largely because such equipment in use by the City already met the required guideline levels.

Although no noise level criteria are available for refuse compaction equipment per se, studies of noise levels produced by City refuse compactors resulted in the vehicles being retrofitted with quieter hydraulic power systems. This modification resulted in lower engine operating speeds which significantly reduced a very intrusive component of noise produced during the compaction cycle (Noise levels which had previously been in the range of 85 to 90 dBA have been reduced to a consistent level between 76 and 80 dBA under load conditions). Notwithstanding these improvements, recent noise level monitoring of City refuse compactors has confirmed that further work should be done at the manufacturing stage to deal with excessive banging noises from the hydraulic system, which produces peak levels in excess of 90 dBA during the compaction cycle. Other equipment used in the construction industry, such as portable electrical generators, portable arc-welders, water pumps, backhoe tractors, and pile drivers are now being manufactured in response to nation-wide noise ordinances which are well within City of Toronto By-law requirements. Regretfully the same cannot be said of caisson drills. Complaints related to noise produced by these machines have confirmed that even recently manufactured units are excessively noisy under load conditions, producing noise levels measured at 15 metres in excess of 90 dBA. These noise levels are produced largely by the hydraulic pumps and engines.

During the past decade a technology known as hydro-demolishing has been developed for repairing salt damaged concrete in parking garages. This process, as well as greatly reducing the duration of the repair work, also transmits marginal noise through the structure of a building, with the added desirable benefit of minimal dust and vibration. In situations where there is mixed residential and commercial usage of a property conventional means of repair work such as multiple concrete breakers has sometimes resulted in the need for staggered hours to accommodate the needs of both groups i.e. late evening work to 11:00 p.m. Initially the power units for the high pressure pumps was problematic in the outside environment, producing noise levels when under load conditions well in excess of 90 dBA at 15 metres.

Ongoing noise complaints and a cooperative effort with the users of this technology has resulted in a significant reduction in noise level emissions from the power units. (The levels have been reduced to the range of 80 to 85 dBA measured at 15 metres). The response by residents in the vicinity of work sites has been encouraging.

Schedule A of the recently amended Noise By-law outlines maximum permissable sound levels for typical equipment used by the City and private industry.

**SCHEDULE "A"**

**Sound Regulations Respecting Equipment**

| Type of Equipment | Measurement Distance (metres) | Permissable Max. Sound Level (dBA) |
|---|---|---|
| AIR COMPRESSORS | | |
| - 70 L/second | 7 | 73 |
| LEAF BLOWERS | 15 | 70 |
| PNEUMATIC PAVEMENT BREAKERS | | |
| • 36 kg, manufactured prior to Jan. 1/81 | 7 | 92 |
| • 27 kg, manufactured prior to Jan. 1/81 | 7 | 90 |
| • all manufactured since Jan. 1/81 | 7 | 86 |
| REFUSE COMPACTING EQUIPMENT | 7 | 80 |
| HEAVY CONSTRUCTION EQUIPMENT (dozers, backhoes, mobile cranes, pile augers, trencher) | | |
| • Greater than 75 kilowatts: | | |
| • manufactured 1/1/79-31/12/80 | 15 | 88 |
| • manufactured 1/1/81 and after | 15 | 85 |
| • Less than 76 kilowatts: | | |
| • manufactured 1/1/79-31/12/80 | 15 | 85 |
| • manufactured 1/1/81 & after | 15 | 83 |

In addition to the ongoing enforcement of the Noise By-law for machines used in the City of Toronto by both public and private sector construction, for the most part, a cooperative effort with the industry has facilitated the achievement of, and compliance with, sound regulations respecting equipment. This has resulted in appreciable reduction of noise from these machines, to an extent where complaints have been significantly reduced.

# Handling Noise Complaints Through Conflict Resolution in Toronto

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## Introduction

The Conflict Resolution Service, a program of St. Stephen's Community House, was initiated in Toronto in 1985. The Conflict Resolution Service ("CRS") is a community based program providing dispute resolution services to help resolve neighbourhood, landlord/tenant, intra-organizational and inter-personal disputes. The CRS approach to mediation is based on conciliation and empowerment of the individuals involved in conflict situation, emphasizing improved communication which leads to shared problem solving and more amicable relationships.

## Volunteer Involvement

The CRS service area is Metropolitan Toronto, except for areas served by other community mediation services. Much of the actual work of mediation and case development is carried out by trained volunteers, who represent many of the racial, ethnic and cultural groups living in Metro Toronto. Using volunteers in this capacity is part of the CRS philosophy of "neighbours helping neighbours", and of spreading the skills on conflict resolution into the community as widely as possible. Confidence in the service is enhanced by the fact that the volunteers' backgrounds are matched to those of the disputants as much as possible, in terms of race, of ethnicity, age, gender, of other relevant factors, e.g. status as landlord or tenant.

## The Conflict Resolution Process

The process of conflict resolution at CRS has three stages. The initial intake interview is done over the phone by a staff person at CRS, who then assigns two trained volunteers to visit each person involved in the dispute in their homes. During these visits the volunteers explain the service to them, hear their point of view on the dispute, and try and persuade them both to use mediation to resolve their dispute. The third stage is the mediation itself, which is carried out by a panel of three volunteer mediators, usually at the CRS office.

The mediation is a structured meeting of about three hours in length. To begin the process both people are asked to explain their point of view on the dispute to the mediators, while the other person listens. They are then encouraged to face each other and discuss the issues. The role the mediators play is neutral and supportive of both sides of the dispute. When the disputants have begun to communicate and feel more comfortable with each other, the mediators guide them in discussing possible solutions which meet as much as possible the needs of each person. A signed agreement is written by the mediators, which the disputants take home with them. The CRS has an 80% rate of success in reaching agreements, once both parties have agreed to come to mediation.

## Benefits of Using Conflict Resolution

The benefits of this process are several: it is quick and, because volunteers carry out so much of the work, it is cost effective. In addition, unlike the court, it leaves decisions in the hands of the people with the dispute, thus ensuring that the solution will be in accord with people's individual needs and values. Because the process promotes communication and replaces the polarization of the adversary system with cooperative win/win negotiation, it builds rather than harms interpersonal relationships. In particular it strengthens communities, as 60% of the cases are between neighbours.

## Resolving Noise Complaints

Approximately 70% of the conflicts involving neighbours are about noise. Most often the noise that is disturbing the first party is heard either through the floor or ceiling of an apartment, or through party walls in a detached house. Noise complaints are generally among the most difficult for the police and/or the courts to resolve. Even in cases where a noise complaint goes to court the underlying problems are not dealt with and are often made worse by the adversarial nature of the court process and the fact that only one party can "win" while the other loses. The underlying problems, which involve the ongoing relationship of the neighbours and their difficulty in communicating with each other to resolve the problem on their own, have a much better chance of real resolution through mediation than through the court system. Mediation allows the neighbours to experience positive communication with each other, and usually results in new understanding and clearing up of wrong assumptions and misunderstanding. Cultural and language differences are often contributing factors in neighbour disputes of all kinds, which is addressed at the CRS through using multicultural teams of mediators and language support when necessary.

## Referral Sources and Statistics

The CRS is referred conflicts by the police, legal services, the offices of elected officials, the Noise Control Branch at City Hall, and other referral sources. Because of the difficulty of convincing second parties to use mediation to resolve their disputes (since conflict resolution is at present an entirely voluntary process), only about 20% of the cases referred to the CRS actually reach mediation. Of the cases involving noise that are referred to the CRS, approximately 25% are resolved either as a result of the case development, or through mediation. At present the CRS is involved in strengthening its relationship with the City's Department of Public Works and the Environment. The City and Metropolitan Toronto Police Force have been increasing their demands for this service especially in matters that are far more than just noise-related and where the enforcement of the Noise By-law does not solve the problem. By introducing a non-"authority" figure the initial barrier of effective communication is removed thus improving chances for successful negotiations. With increasing pressures for financial restraint, while still maintaining service, it has become essential that such a referral service becomes part of the overall administrative framework for handling noise complaints. Essentially, St. Stephen's would play a more active role in the process and, with established clear guidelines, the efficiency in handling complaints would improve substantially.

At time of writing this proposal had not been approved by City Council but generally accepted by a City interdepartmental streamlining committee as a recommendation to be considered.

# Air Conditioning Device Noise Control: Ontario Model By-law and Toronto By-Law

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### Introduction

The Ontario Ministry of the Environment and Energy created an ad hoc advisory committee to develop modifications to the model municipal noise control by-law regarding noise from air conditioners and heat pumps. Members included representatives from local and provincial government, Ontario Hydro, Toronto Hydro, builders, a consumer association, manufacturers, installers and consultants. Over the past five years they struggled with the issues of sound level limits, enforcement and installation guidelines. The outcome was a comprehensive model noise by-law and a detailed guide for the installation of units including checklists for both the installer and the enforcement agency. The City of Toronto sat on the committee but pursued a different approach, partly due to the existence of their own noise control by-law which had been in use for several years. The City examined changes to their by-law in light of the work by the Committee. The paper summarizes the background and details of the two noise control documents.

### History and Philosophy

The Ontario Ministry of the Environment and Energy updated its noise guideline entitled "NPC-116 Residential Air conditioners" which was part of the Model Municipal Noise Control By-Law issued in 1978. This was part of a general updating of a number of documents in the by-law. The new document entitled "NPC-216 Residential Air Conditioning Devices" was finalized in 1993 after five years of work by an advisory committee formed as the result of a symposium in 1988.

The revisions to the document followed the general philosophy of the other NPC series guidelines. This includes setting sound level limits for stationary sources on an hourly basis related to the ambient sound environment. The source descriptor is the energy equivalent sound level (Leq) and the limit is set for the source alone. The noise control guidelines are intended to be used mainly for enforcement but are useful also for design of new installations.

Recently, the ambient sound environment has been categorized by the use of "class area" designations.

The City of Toronto passed its comprehensive noise by-law (44-75) in 1975, preceding that of MOEE by three years, but the background work of each group overlapped somewhat. With respect to air conditioning devices which fell under the "source equipment" category, the philosophy was to limit the amount by which the ambient could be raised due to the source contribution. The descriptor is the L90. Thus it is not necessary to identify the sound level of the source alone in a complaint investigation, only the increase due to the source. The by-law was intended as an enforcement tool.

### Sound Level Limits

The Province deals only with urban areas at present and sets both a general sound level limit and specific sound level limits. The more lenient in any situation applies. In addition, a sound emission limit is also given. The general sound level limit is 5 dBA greater than the ambient due to road traffic. This applies to a one hour period between 7 a.m. and 9 p.m., the specific time to be determined by the noise control officer. This limit was set as a result of a survey conducted during daytime and evening hours. Even though complaints may originate at night, the annoyance threshold was found to be 5 dBA over the day/evening sound environment.

Specific hourly sound level limits are 50 dBA in Class 1 areas (major population centre), 45 dBA in Class 2 areas (urban with quiet evening) and 55 dBA in areas for which air conditioning was a mandatory noise control requirement for new developments.

The sound emission limits apply to manufacturers and are 8.0 bels ARI Sound Rating for units built in 1991 and 7.6 bels for units built after that. It applies to 3.5 ton central units and smaller. No provincial limits apply to units made before 1991.

The City of Toronto originally set an allowable L90 increase of 2 dBA in any one hour period. The latest revision has changed this to an L90 increase of 5 dBA in any time period. In essence, this means that the source cannot produce more than the L90 plus 3 dBA where previously it was the L90 less 2 dBA. It is also proposed to set a sound emission limit.

Compared to the provincial limits, the City felt that certain areas of the City would require limits lower than the provincial minimum of 45 dBA. For other areas where ambients are higher, the limits are generally more stringent under the City by-law than the provincial.

### Enforcement

During a complaint investigation, the provincial model by-law requires various measurements. To be most efficient, a procedure is given whereby the ambient is measured (20 minutes minimum) and then a short (less than one minute) measurement with the unit in operation is taken. If the combined level does not exceed the limit, no further measurements are needed. If so, additional measurements with the source in operation (20 minutes minimum) are required. In all cases, the investigator must inhibit sounds other than that of road traffic and the source in question. Calculations are then done to separate the source from the ambient and to determine the magnitude of the excess, if any.

The City is required only to take measurements with and without the source operating. The time period is at the discretion of the investigator. Since the L90 setting is used, there is no need to inhibit other sounds. A simple subtraction identifies an excess of not.

### Design

Because the Province has used road traffic Leq as the ambient, it is possible for designers to calculate the ambient sound level for a given installation. Alternatively, the specific limits could be used as a conservative approach, provided the area can be appropriately categorized by class designation. Or, field measurements could be conducted.

In the City, the L90 is difficult to estimate or calculate and monitoring is probably the only solution. The City provides data from their monitoring program to assist in this regard. However, a generally more stringent limit would be set than for Provincial limits because the time period of investigation is 24 hours a day.

**Installation**

The MOEE has produced a set of installation guidelines to accompany NPC-216. The guidelines include a checklist for installers to use at the time of sale. It enables the appropriate bel rating to be determined in order to comply with the sound level limit. In addition, the guideline discusses other alternatives if the conventional air cooling condensing unit alone does not meet the limit.

The City of Toronto considered and rejected a permit system for air conditioners. Instead, it encourages discussion and education in order to deal with potential problems at an early stage. After the fact. City inspectors have offered guidance in providing solutions to complaints.

**Summary and Conclusion**

Two different approaches have been taken to air conditioner noise control; that being the sound descriptor, time of measurement and duration of measurement. Both are workable but require the use of properly trained noise control officers. This is necessary to accommodate the varying ambient sound levels which occur in urban areas.

As by-laws, they are primarily oriented to enforcement but by the fact that specific limits are given they are also useful to designers of new installations.

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# DEVELOPMENTAL PLASTICITY OF CENTRAL AUDITORY PATHWAYS: FREQUENCY REPRESENTATION AFTER NEONATAL HIGH FREQUENCY HEARING LOSS

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## INTRODUCTION

Cochleotopic (or tonotopic) organization is the systematic representation of the sensory epithelium of the cochlea within central auditory pathways including cortex. This central mapping of the sensory surface is a feature of all sensory systems. The organization of these sensory maps can be significantly modified by abnormal patterns of excitatory input, particularly during early stages of development. This has been shown most extensively in studies on the visual system [1] and in the somatosensory system [2,3]. More recently, studies in the auditory system have revealed that alterations to cochleotopic maps in primary auditory cortex can result from cochlear haircell lesions [4,5]. Our previous work has shown that ototoxic poisoning of the basal cochlear region (i.e. partial deafferentation) in newborn kittens results in the development of major changes to the frequency map in primary auditory cortex. Thus, regions of cortex which would normally contain neurons coding high frequencies (activity originating at cochlear base) has neurons tuned to lower frequencies. It appears that the establishment of cochleotopic maps in auditory cortex depends on the integrity of the pattern of ascending input from the cochlea. In this study, we ask if this reorganization is a feature only of the auditory cortex or whether it also exists at the midbrain level (inferior colliculus; IC).

## MATERIALS AND METHODS

Newborn chinchilla pups were treated with the ototoxic aminoglycoside amikacin, 400 mg/kg/day for 2-4 days s.c., resulting in bilateral lesions to the base of the cochlea. We monitored auditory threshold elevation using auditory brainstem evoked responses to tone-pip stimuli (ABR audiograms). At maturity (6 months) the subjects were used in single unit electrophysiological recording studies in which the frequency representation (the cochleotopic map) at the level of IC was determined. (All procedures were carried out within the guidelines of the Canadian Council on Animal Care).

## RESULTS AND DISCUSSION

Figure 1 shows the cochleotopic or tonotopic map in IC of the normal chinchilla. Dorsally, neurons respond best to low frequencies of sound; in ventral areas, they respond best to high frequencies. Figures 2 & 3 show the frequency maps in IC of two subjects treated with amikacin, i.e. with long-term neonatal high frequency hearing loss (as indicated by the ABR audiograms). Note in both these examples that the frequency region corresponding to the border of the cochlear lesion, and therefore the high frequency cut-off of the audiogram, is over- represented. In the subject of Figure 2, this cut-off is at 10 kHz; note the "expanded" region (cross-hatched area) containing neurons all tuned 10 kHz. In Figure 3 the cut-off slope of the audiogram is at about 5kHz; again and the IC contains a larger than normal region in which all cells are tuned to 5 kHz. It should be noted, however, that the tuning and threshold characteristics of the neurons in these expanded regions are pathological, i.e. thresholds are elevated and frequency tuning is abnormally broad (this is consistent with previous work on the threshold and tuning properties of neurons from damaged cochleas, [e.g. 6,7,8].

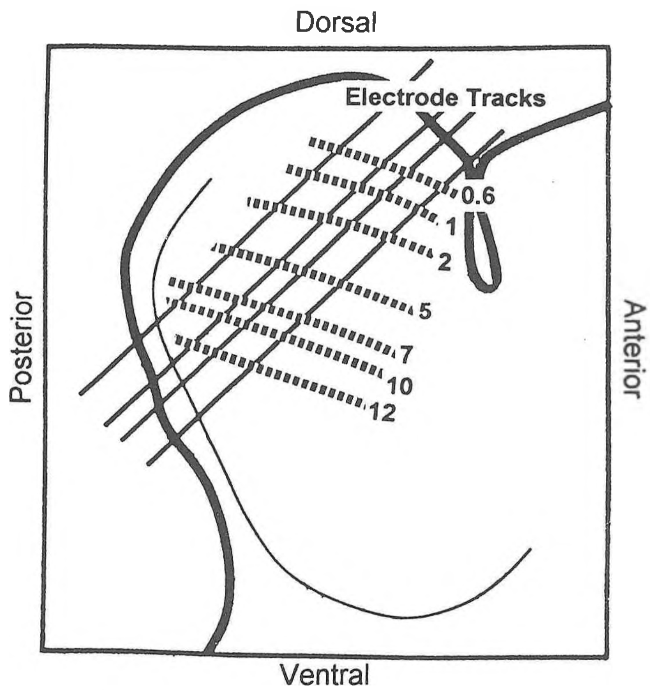
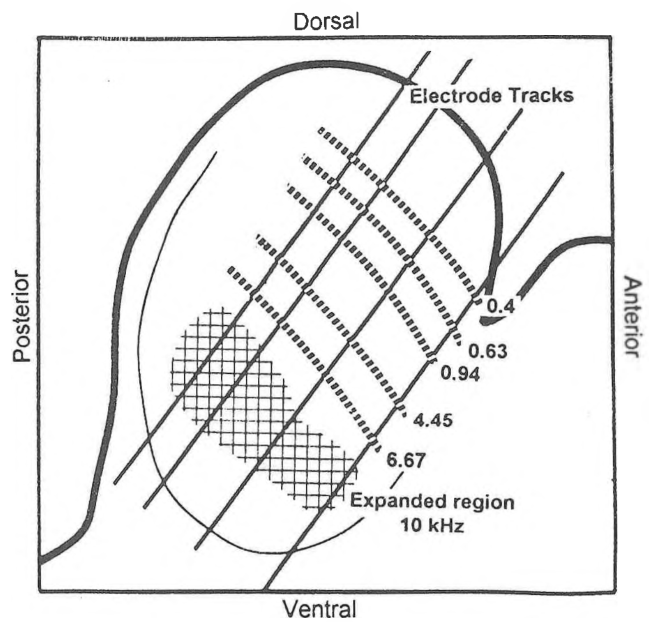



Figure 1. Saggital section through inferior colliculus indicating cochleotopic organization in the normal chinchilla midbrain.



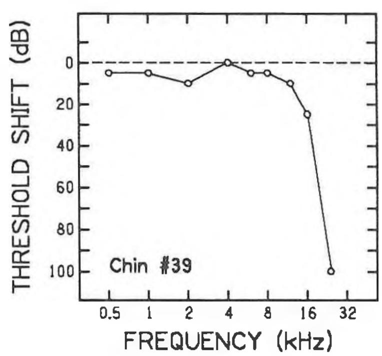
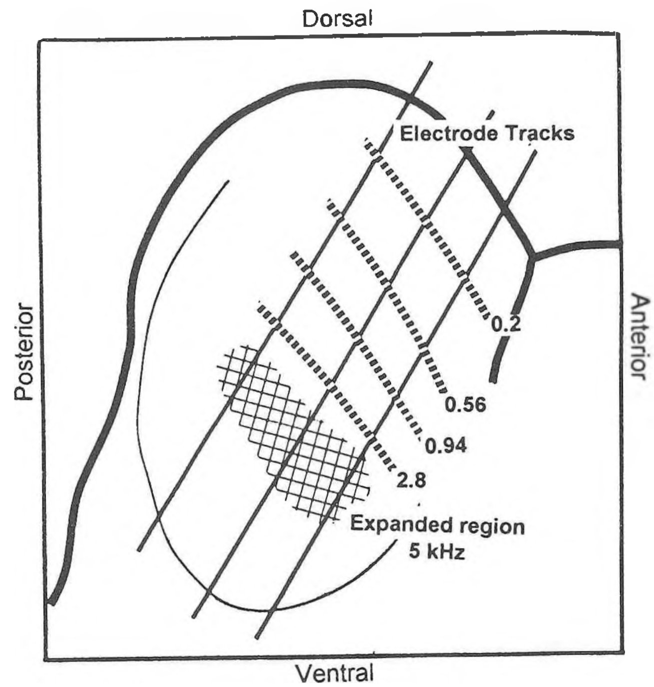




Figure 2.
Abnormal cochleo-
topic organization
in inferior colliculus
of amikacin treated
chinchilla.



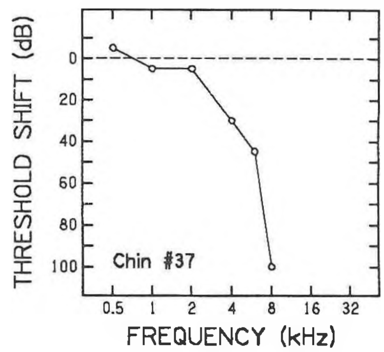




Figure 3.
Abnormal cochleo-
topic organization
in inferior colliculus
of amikacin treated
chinchilla.

These experiments give us new information in relation to the basic mechanisms of plasticity of the central nervous system. The integrity of the cochlea is important for the development of central frequency maps; changes to auditory sensory input, particularly from an early age can significantly alter cochleotopic maps not just at cortex but at the midbrain level. Furthermore, our experimental subjects are animal models of human sensorineural hearing loss. Thus we believe that the changes to central cochleotopic maps that we observe are to be found in humans with long-term cochlear hearing loss. There are still many important questions in relation to this developmental plasticity. First, is there a developmental critical period when the central auditory pathways are most susceptible to changes in cochlear afferent input? Secondly, are these changes reversible? The answers to both questions have direct relevance to the treatment and rehabilitation of hearing loss, particularly in infants.

## ACKNOWLEDGMENTS

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# La relation entre la largeur des filtres auditifs et les seuils d'audibilité

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Dans le cadre de la mise au point d'une méthode clinique de mesure de la sélectivité fréquentielle, nous avons entrepris d'adapter la mesure des filtres auditifs aux contraintes de l'examen clinique. La procédure a été optimisée auprès d'auditeurs normaux et des données normatives ont été recueillies [1]. Le présent projet visait à raffiner la stratégie d'examen auprès de personnes malentendantes, auprès desquelles une première collecte de données a permis d'identifier certaines difficultés méthodologiques [2]. Les questions suivantes ont été abordées: comment traiter les données lorsque l'asymétrie des filtres est telle que le calcul de la pente supérieure du filtre donne un résultat aberrant? A partir de quelle degré de perte auditive une personne est-elle susceptible de montrer une perte significative de sélectivité, d'une part, et une perte complète de sélectivité d'autre part?

La mesure des filtres auditifs est fondée sur le principe selon lequel le seuil masqué ($P_s$) variant en fonction des fréquences de coupure d'un bruit à échancrure est directement proportionnel à la puissance du bruit qui passe à travers le filtre. L'équation 1 définit cette relation:

$$P_s = K F_c N_0 \left[ \int_{-\infty}^{0} W_l(g)\, dg + \int_{0}^{\infty} W_u(g)\, dg \right] \qquad (1)$$

où $F_c$ est la fréquence centre du filtre, $N_0$, le niveau spectral du bruit, K, une constante de proportionnalité, g, la fréquence de coupure normalisée de l'échancrure. $W_l(g)$ et $W_u(g)$ définissent respectivement les branches inférieure et supérieure du filtre d'après les équations suivantes:

$$W_l(g) = (1-r)(1+p_l g)e^{-p_l g}+r \qquad (2)$$

et

$$W_u(g) = (1-r)(1+p_u g)e^{-p_u g}+r \qquad (3)$$

Dans ces équations, $P_l$ et $P_u$ correspondent à la pente inférieure et supérieure du filtre et r, à sa gamme dynamique. L'effet de l'asymétrie du filtre sur le calcul des pentes à partir des seuils masqués est illustré à la Figure 1. On y a simulé l'effet d'un rapport $p_u/p_l = 3$ sur la mesure des seuils masqués en présence de 6 conditions d'échancrure. On note que les seuils masqués déterminés par les deux branches combinées du filtre auditif sont assimilables à ceux qui auraient résultés de la seule influence de la branche inférieure; par contre, ils ne permettent pas de déduire les caractéristiques de la branche supérieure (equation 3) et, par conséquent, la largeur elle-même du filtre. Cette illustration, bien que théorique, démontre la nécessité de procéder à une mesure distincte des caractéristiques de la branche supérieure du filtre auditif dans les cas de forte asymétrie. Cette contrainte a été prise en compte dans la procédure d'examen décrite plus bas. Par ailleurs, la question de la relation entre élargissement du filtre et perte d'audition a gouverné le recrutement des participants à l'étude.

## Méthodologie

### Participants

Les 31 participants à l'étude ont été recrutés parmi les bénéficiaires des services de réadaptation audiologique de l'Institut Raymond-Dewar de Montréal. Les critères d'exclusions étaient les suivants: (a) écart entre seuils tonals aériens et osseux supérieur à 10 dB; (b) tympanogramme anormal; (c) seuil d'audition maximal entre 0.25 et 4 kHz supérieur à 75 dB HL [3]; (d) âge supérieur à 65 ans ou inférieure à 18 ans.

Les critères d'inclusions étaient les suivants: montrer des évidences d'atteinte neuro-sensorielle et une courbe audiométrique descendante (écart entre la moyenne des seuils à 2, 3 et 4 kH et la moyenne des seuils à 0.25, 0.5 et 1 kHz supérieur à 10 dB).

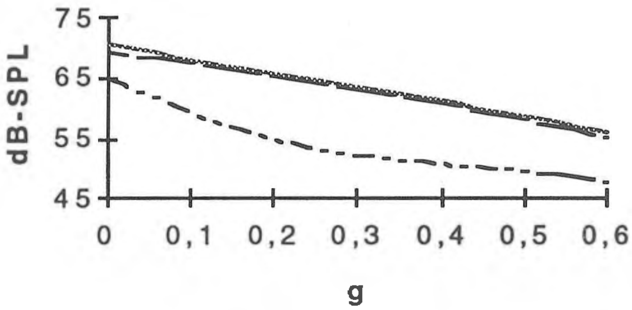


Figure 1. Seuils masqués déterminés par la branche inférieure ($P_{sl}$ - trait interrompu), supérieure ($P_{su}$ - trait achuré) et par les deux branches combinées (Ps - trame) du filtre en fonction de g, lorsque $P_l$=7, $P_u$=21, r=20 dB et K=0 dB.

### Procédure

Le dispositif et la procédure de mesure des filtres étaient identiques à ceux utilités lors de la mise au point du protocole simplifié [1], sauf pour la prise en compte de l'asymétrie du filtre. La méthode de Békésy servait à la mesure des seuils masqués pour 4 conditions d'échancrure symétriques par rapport à la fréquence du signal et 2 conditions asymétriques. Le niveau de densité spectral du bruit ($N_0$) était d'abord fixé à 40 dB/Hz si le seuil absolu était inférieur à 55 dB HL; si le seuil masqué en condition d'absence d'échancrure ($g_l=g_u=0.0$) était de 10 dB supérieur au seuil absolu, les 5 autres conditions d'échancrure étaient alors testées. Si l'écart était inférieur à 10 dB, $N_0$ était fixé à 50 dB/Hz; si l'écart entre seuil masqué et seuil absolu était inférieur à 5 dB, l'examen était terminé. Dans le cas contraire, on procédait à la mesure des seuils masqués pour les 5 autres conditions d'échancrure.

Le traitement numérique des données en fonction des équations 1, 2 et 3 suivait la procédure proposée par Glaberg et Moore [4]. La sélectivité des filtres auditifs a été caractérisée par le calcul de la largeur des filtres rectangulaires équivalents (ERB). Les valeurs obtenues à $N_0$=50 dB/Hz ont été converties en valeurs estimées pour $N_0$=40 dB/Hz [4], puisque les valeurs normatives ont été recueillies dans cette condition [1].

Lors du traitement des données, l'absence de sélectivité fréquentielle a été définie par un écart inférieur à 3 dB entre le seuil masqué obtenu en l'absence d'échancrure ($g_l=g_u=0.0$) et le seuil masqué obtenu en présence d'une échancrure maximale ($g_l=g_u=0.5$).

Indépendament du niveau spectral utilisé, lorsque le calcul de $p_u$ donnait une valeur aberrante, c'est-à-dire plus grande que celle obtenue chez des auditeurs normaux [1], des mesures complémentaires étaient effectuées avec un bruit passe-haut dont la fréquence de coupure, $g_u$, variait de 0.0 à 0.5 par pas de 0.1. Les seuils masqués ainsi obtenus ont été introduits dans l'équation 3.

**Résultats et discussion**

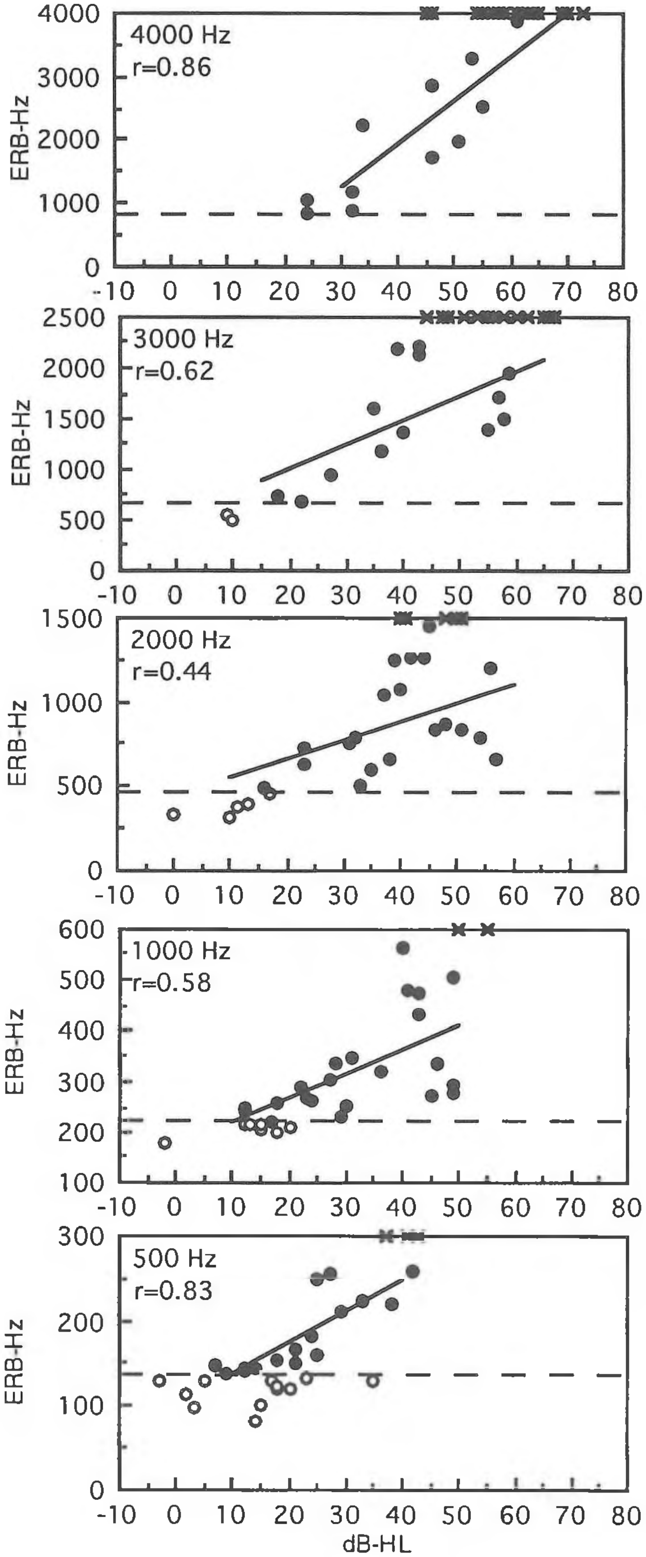


Figure 2. Distribution des valeurs de ERB en fonction des seuils audiométriques à 5 fréquences; la droite pointillée indique le 95ème centile des ERB dans une population normale [1]; les cercles vides correspondent aux cas de ERB normaux, les cerlces pleins, aux cas de ERB anormalement élargis et les "x" aux cas d'absence de filtre auditif (N=31).

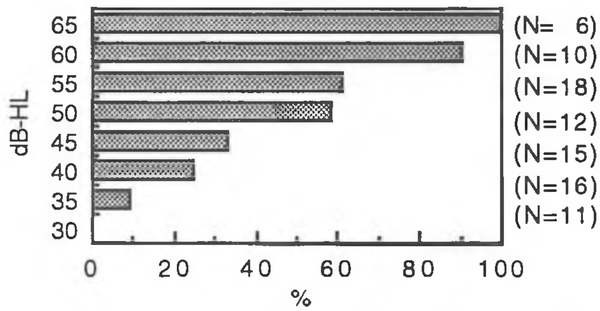


Figure 3. Proportion des cas d'absence de sélectivité en fonction des seuils d'audition (à 5 fréquences audiométriques) regroupés par classes de 5 dB.

Le problème de valeurs de pente $p_u$ aberrantes s'est posé dans 21 cas sur 155 mesures et a été solutionné de façon satisfaisante dans 11 d'entre eux par la procédure décrite plus haut. Les autres cas correspondaient à une absence de masquage à 50 dB/Hz ou à une valeur extrême d'asymétrie du filtre.

La Figure 2 montre les résultats obtenus en termes de largeurs des filtres auditifs. Aucune corrélation signficative ne pouvait être obtenue entre valeurs de ERB et seuils absolus lorsque les ceux-ci étaient inférieurs ou égaux à 15 dB HL [1]. Cependant, on observe une relation systématique et statistiquement significative ($p<0.05$) chez les personnes présentant des filtres anormaux à 5 fréquences différentes (Fig. 2). Toutefois, à seuil égal, on observe des variations inter-individuelles importantes, en particulier à 1, 2 et 3 kHz. On remarque que des pertes significatives de sélectivité (valeurs supérieures au 95ème centile de la population normale) apparaissent à des seuils absolus voisins de 20 dB HL. Il s'agirait bien d'une frontière en-deçà de laquelle la mesure clinique des filtres auditifs présenterait généralement peu d'intérêt (sauf dans les cas de plaintes explicites de difficultés d'audition dans le bruit).

Par ailleurs, l'absence de sélectivité est observée dans certains cas pour lesquels le seuil absolu n'est que de 40 dB HL. Comme le montre la Figure 3, la probabilité d'absence de filtre croît de façon systématique à partir de cette valeur. A des seuils de 50 à 59 dB HL est associée une probabilité voisine de 50% d'absence de sélectivité. Les données reproduites à la Figure 3 suggèrent en outre qu'il est inutile d'entreprendre des mesures de sélectivité fréquentielle dans les cas où les seuils absolus atteignent ou dépassent 65 dB HL.

La poursuite de notre collecte de données sur un plus grand nombre de personnes malentendantes offrira des assises encore plus solides aux estimations rapportées ici. Nos résultats offrent néanmoins des réponses satisfaisantes à nos questions de départ.

Etude subventionnée par l'I.R.S.S.T. (#N/D PE-90-13).

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# FREQUENCY AND THE SENSATION OF SOUNDS

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Psychoacoustics deals with the biological response to an auditory stimulus. This biological response, however, depends on the characteristics of this physical stimulus in a complex manner. For example, it would be an oversimplification to state that loudness of a pure tone depends solely on the amplitude of a sound pressure wave, and that pitch depends solely on the frequency. The physical quantities sound intensity, sound duration and frequency combine in a subtle manner to produce the psychoacoustic quantities loudness and pitch. In an attempt to isolate the effects of each physical variable, experiments have been conducted in the past to determine the relationship between:

a) loudness and intensity, with frequency held constant;
b) pitch and frequency, with intensity held constant.

The relationship between loudness/pitch and intensity/frequency has also been investigated using experiments on auditory discrimination. Discrimination experiments involve determining how much change in intensity (or frequency) is required to elicit a change in biological response. For example, a tone of 100 Hz can probably not be distinguished from a tone of 101 Hz at all relevant sound intensities, as long as the intensity is held constant between the two tones. Commonly one calculates the Weber Fraction, which is the change in intensity/frequency divided by the reference intensity/frequency, required to produce a change in sensation. The Weber fraction is calculated over a range of intensities (or frequencies). There are two types of Weber Fraction with which we shall be concerned, corresponding to Experiments a) and b) above:

c) fractional changes in intensity, with frequency held constant;
d) fractional changes in frequency, with intensity held constant.

We shall return to these experiments after giving a little theoretical background.

We have been developing, over a number of years, an entropic or informational approach towards quantifying human sensation. A sequence of papers have been published detailing our advances of this method [1,2]. A single, master equation of three parameters has been derived relating the variables biological response, constant sensory stimulus intensity, and time since onset of the sensory stimulus. This equation accounts quantitatively for nearly all experimental results relating the three sensory variables. The published form of this equation deals only with steady sensory inputs (constant intensity over time), but the equation functions over almost all sensory modalities. In particular, we are now interested in its predictive abilities in auditory psychophysics.

The entropy equation, in its simplest form, may be written as follows:

$$F = \tfrac{1}{2} k \ln(1 + \beta I^n / t), \qquad (1)$$

where $F$ is biological response, $I$ is auditory stimulus intensity, and $t$, the time since onset of stimulus, must be $\leq t_{max}$. The exponent, $n$, to which $I$ is raised, can be identified with the Steven's exponent appearing in the power law of sensation. For audition, this exponent has a mean value of about 0.3 if $I$ is measured in units of sound intensity. The response $F$, may be either impulse firing rates in auditory ganglion cells or loudness, with appropriate change in time scale. The derivation of this equation is given in [1,2,3]. Notice that the time variable divides into the intensity variable in the numerator. For constant $I$, this has the effect of decreasing the "effective intensity" over time and, consequently, decreasing the response $F$, as seen by Equation (1). A monotonic decreasing $F$ with time corresponds to adaptation to a stimulus of constant intensity. In audition, $F$ would represent either loudness decreasing over time, or the decrease of the firing rate in the primary sensory afferent neurons attached to the hair cells. This equation has been applied to the experiment of Yates et al. on the guinea pig auditory ganglion cells. The parameters $k$, $\beta$ and $n$ were obtained from curve fitting the data of Yates et al. [4]; the analysis can be found in reference [3].

For magnitude estimation experiments, the sound stimulus is applied for a constant duration of time, say $t'$. Equation (1) then takes on the following form:

$$F = \tfrac{1}{2} k \ln(1 + \beta I^n / t'). \qquad (2)$$

By setting $\gamma = \beta / t'$, we obtain

$$F = \tfrac{1}{2} k \ln(1 + \gamma I^n). \qquad (3)$$

Notice that, for small values of $\gamma I^n$, we can expand the right hand side of Equation (3) in a Taylor series of the form $\ln(1 + x) \approx x$ to get

$$F = (\tfrac{1}{2} k\gamma) I^n. \qquad (4)$$

This equation is recognized as the power law of sensation. One can now appreciate why the parameter $n$ in Equation (1) can be identified with the Steven's exponent in the power law of sensation. If we now let $\gamma I^n$ become large, we can approximate Equation (3) with the form $\ln(1 + x) \approx \ln(x)$ to obtain

$$F = (\tfrac{1}{2} kn) \ln(I) + const. \qquad (5)$$

which is the logarithmic law of sensation. Although it has been observed that, for most of the physiological range of $I$, *both* the power and the logarithmic laws hold to a high degree of approximation, both laws systematically deviate from the data at, respectively, larger and smaller intensities. Equation (3) is a more general law of sensation embracing both the power and the logarithmic laws. Equation (3) describes all published data of the loudness-intensity type whether or not they conform to a straight line when plotted in a semilog or full log plot. To wit, consider the magnitude estimate data of Luce and Mo [5], which do not conform to a straight line with either type of graph, but are fitted well by Equation (3). The analysis of the data can be found in reference [3].

Returning to Equation (3), taking the derivative of $F$ with respect to $I$ and replacing the differentials with finite differences, one can arrive at an expression for the Weber Fraction describing discrimination experiments. Reference [3] provides the derivation of the equation, which has the following form,

$$\frac{\Delta I}{I} = \frac{2\Delta F / k}{n}(1+\gamma^{-1}I^{-n}). \tag{6}$$

Equation (6) can be applied to analyze the experiments of Riesz [6], who performed auditory intensity discrimination experiments (Experiment c) on human subjects. In fact, the equation he used to describe the data matches Equation (6) term for term. Although Riesz did not derive the equation he used, offering the equation only as an empirical fit for the data, we have now been able to derive the equation theoretically. There are various other auditory experiments involving intensity embraced by the seminal Equation (1), including the SDLB effect and some of Békésy's results.

We now wish to incorporate a frequency variable into Equation (1) in order to account for frequency effects in the sensation of sounds. Notice that, if we were to utilize Equation (1) to describe pitch sensation, the time variable becomes "extraneous", in the sense that one does not adapt to the frequency of sound. In other words, the time variable, in pitch sensation, is used for something other than adaptation. We can associate the inverse of time with frequency. For a pure tone, inverse time will simply be the frequency of the tone. For a more complex tone, Schouten's theory of hearing postulates that a non-linear filter in the ear allows the ear to pick up the fundamental frequency of oscillation of a complex tone (what Schouten calls a *residue*) [7]. The modified equation would now have the following form, where $f$ is the relevant frequency:

$$F = \tfrac{1}{2}k\ln(1+\beta I^{n} f). \tag{7}$$

The work of Linsay and Norman [8] provides an excellent of test of Equation (7). At a fixed sound intensity, they determined experimentally how pitch changes with frequency (Experiment b). Furthermore, the empirical equation they used to fit their data is identical to Equation (7),

$$F = 2410.\ln(1+1.6\times 10^{-3} f), \tag{8}$$

providing some confirmation of the validity of our inverse time equation.

An additional equation can be derived to account for frequency discrimination experiments (Experiment d). The derivation is mathematically identical to the derivation of Equation (6) and the Weber Fraction function for frequency takes on a similar form to it:

$$\frac{\Delta f}{f} = 2\frac{\Delta F}{k}[1+(\beta I^{n} f)^{-1}]. \tag{9}$$

This equation provides good prediction of the data of Shower and Biddulph [9], who did experiments to determine the Weber Fractions of frequency as a function of frequency for constant sound intensity. The equation also predicts that $\Delta f \ / \ f$ will diminish for increasing $I$ as shown by Shower and Biddulph.

In summary, while experiments a) through d) have been analyzed by the experimenters themselves, their method of analysis is often empirical or applicable only to their own experiments. We now offer a unified approach to the study of auditory psychophysics by proposing that a *single* equation can account for experiments a) to d). To do so, we have reinterpreted a variable in our original equation so that it can now account for frequency effects in the sound stimulus.

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# APPLICATION OF AN AUDITORY MODEL TO THE COMPUTER SIMULATION OF HEARING IMPAIRMENT: PRELIMINARY RESULTS

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## 1. INTRODUCTION

A new computational model of the auditory periphery has been recently reported by the first two authors [1, 2]. The model is particularly attractive for hearing research in that it enables practical simulation of auditory nonlinearities and feedback mechanisms, and the study of the deterioration of these processes in the hearing impaired.

In this article, we present the results of an exploratory study designed to simulate the perceptual consequences associated with cochlear hearing loss. For this purpose, we combine the auditory model to a state-of-the-art recurrent neural network classifier [5] to form an auditory-based automatic speech recognition system. Preliminary speech recognition results for normal and impaired operations of the auditory model are then compared.

The ultimate goal of this work would be to guide the development of signal processing strategies for hearing aids. The idea is to find new ways of processing input speech so that, when passed through a model of impaired cochlea, the observed auditory nerve firing patterns and/or speech recognition scores would be as close as possible to those observed for unprocessed speech passed through a normal cochlea.

## 2. THE AUDITORY MODEL

The structure of the model closely follows the general architecture of the peripheral ear as shown in Fig. 1.

The different stages of the ascending path can be equivalently represented as lumped-element analog circuits or as wave digital filters (WDFs) [2]. The input $P(t)$ is digitized speech or other incident acoustic wave. This signal is processed by a WDF module representing the sound transformation through the outer ear, middle ear and cochlea. The middle ear stage includes a time-variant capacitive element $C_{st}(t)$ modelling the variable acoustic compliance of the stapes suspension in response to stapedial muscle contractions. The cochlear stage is based on the classical 1-D transmission line model of basilar membrane (BM) motion, extended to account for the mechanical effects of the outer hair cells (OHCs). By injecting energy in phase with BM velocity at low levels, the OHC circuit leads to auditory filters with level-dependent frequency selectivity and sensitivity. At low input levels, the BM is sharply tuned and highly sensitive. At high input levels, the BM is broadly tuned and the characteristic frequency shifts by about half an octave to a lower frequency. Over the full input range, the BM shows 31 dB of dynamic compression near the characteristic frequency. These properties are in broad agreement with physiological observations [3].

The ascending path is completed by the inner hair cell (IHC) transduction model of Meddis [4], implemented as a wave digital filter. There are N WDF inner hair cell modules, one per BM segment, using the parameters of a medium-rate fibre. The input $s_n(t)$ to the IHC indexed $n$ is assumed to be proportional to the velocity $i_n(t)$ of the BM segment $n$ to which it is paired (Fig. 1). The fluid-cilia coupling gain

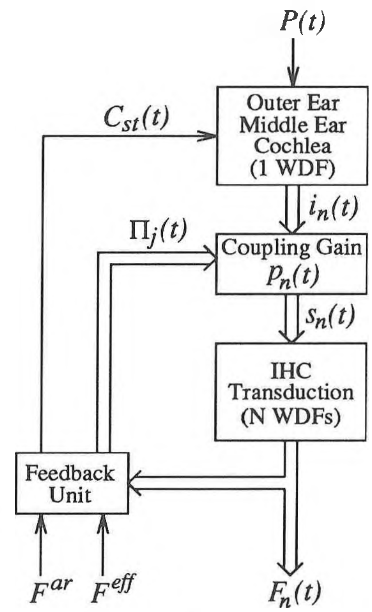


Figure 1. Block diagram of the auditory model.

$p_n(t)$ is made time and space variant as discussed below. The model output $F_n(t)$ is the instantaneous firing rate of the N tonotopically-arrayed IHC afferent fibres.

The model also includes a simple feedback unit simulating the dynamics of the descending paths to the peripheral ear. The acoustic reflex is assumed to be a regulation system whose goal is to maintain the average firing rate to a constant target rate of $F^{ar}$. The control function is a slow modulation of $C_{st}(t)$, leading to a decrease in middle-ear transmission by up to 15 dB below 1000 Hz. The OHC efferent system is also assumed to be a firing rate regulation system. The control function is taken as a slow modulation of the coupling gain $p_n(t)$. The N fibres at the output of the IHC stage are grouped into J contiguous bands, and regulation is applied independently in each band with a target rate of $F^{eff}$. The gain control command $\Pi_j(t)$ is then spatially interpolated to yield $p_n(t)$, leading to an inhibition of cochlear output equivalent to a reduction in input level by up to 24 dB.

The auditory model was applied to the analysis of speech data by computing auditory nerve cochleograms [1]. The OHC circuit provides level compression and spectral sharpening in the low energy portions of an utterance. This results in a better resolution of the formant structure for weak vowels and nasals, and an enhancement of fricative noise. The descending paths lead to further dynamic compression.

## 3. THE RECURRENT NEURAL NETWORK

The structure of the recurrent neural network is shown in Fig. 2 and described in detail in [5]. The input and output

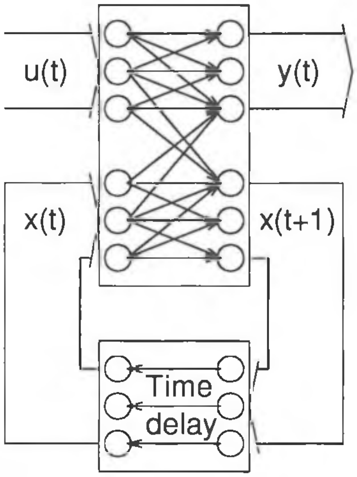


Figure 2. Block diagram of the recurrent neural network.

vectors are divided into external and internal portions. The external input vector $u(t)$ is a sequence of frames (16 ms duration) of parameterized speech of 48 dimensions from the auditory model. The components of $u(t)$ are as follows:

$$u_i(t) = \begin{cases} C_{st}(t) & \text{for } i = 0 \\ \Pi_i(t) & \text{for } 1 \leq i \leq \mathrm{J} \\ F_{i-\mathrm{J}}(t) & \text{for } \mathrm{J}+1 \leq i \leq \mathrm{J}+\mathrm{N} \end{cases}$$

where J=3 and N=44. Thus, both the tonotopic distribution of firing activity in the ascending path and the control commands from the descending paths are fed to the recognition stage. The external output vector $y(t)$ has 61 dimensions, one per symbol of the phone set of the TIMIT speech database [7]. The internal output forms a state vector $x(t)$ of 160 dimensions and is fed back to the input in the next time frame. These recurrent connections allow contextual information to be accumulated over time in the state vector. The recognition process can then use this information to make a more accurate classification.

The training data consists of 8 utterances (the si and sx sentences) from each of the 420 different speakers of the training portion of the TIMIT database. Training proceeds by unfolding the network in time over several frames of speech, comparing the external outputs to the target hand-labelled phone symbols, and adjusting the weights of the network so as to maximize a log-likelyhood cost function.

The test data consists of 8 utterances from each of the 210 different speakers of the test portion of the TIMIT database. The external outputs $y(t)$ are interpreted as phone probabilities for the specified speech frame. The most likely sequence of phone symbols is then computed from the frame by frame phone probabilities using dynamic programming. Final phone recognition results are obtained by comparing these machine-labelled symbols to the target hand-labelled symbols.

## 4. RECOGNITION RESULTS

Recognition results based on two modes of operation of the auditory model, normal and impaired, are reported in Table 1. The first number is the percentage of hand-labelled phone symbols correctly detected. The last number is the recognition accuracy, defined as 100% minus the percentage of insertion, substitution and deletion errors, and is the most important performance measure in this table.

When the auditory model is operated in its normal mode, the recognition accuracy is 61.8%. Over 2/3 of all errors are substitution errors. Inspection of the confusion matrix revealed that the most common substitution errors are between phones from the same broad class (e.g. /z/ vs /s/, /m/ vs /n/) and often involves nearby vowels on the vowel triangle (e.g. /ih/ vs /ix/, /ax/ vs /ix/). There are relatively few errors across broad classes.

| mode | correct | insert. | subst. | delet. | accur. |
|---|---|---|---|---|---|
| normal | 66.1% | 4.2% | 26.5% | 7.4% | 61.8% |
| impaired | 58.7% | 4.9% | 32.4% | 8.9% | 53.8% |

Table 1. Recognition results

The impaired mode of operation of the auditory model consisted of disconnecting the OHC circuit, in effect simulating a total loss of OHCs. There is a loss of sensitivity and frequency resolution at low levels. The descending paths have also been cut off in this mode, although the control commands have been calculated and fed to the recognition stage for a fairer comparison with the normal mode. The neural network has been re-trained and re-tested with this new data. The recognition accuracy has dropped to 53.8%, a decrease of 8.0% in absolute terms with respect to the normal mode. All major types of errors have increased, but particularly substitution errors. Inspection of the confusion matrix revealed that recognition performance has decreased for all classes of phones. Nasals and non-sibilant fricatives are the most affected. Vowels, affricates and sibilant fricatives are the least affected. Amongst vowels, there is a tendency for the confusions to occur between vowels having their first formant in similar frequency regions, and this is also observed in hearing-impaired listeners [6].

## 5. FUTURE WORK

There remains many aspects to consider before we can achieve the ultimate goal of guiding the development of signal-processing hearing aids. In the short-term, recognition results will have to be repeated for more selective cochlear lesions than used here, both in quiet and in noise. In the longer term, further validation of the auditory model is needed by comparing the model output to physiological data for normal and impaired ears. The recognition stage will also have to be reviewed to ensure that the observed decrement in scores in the impaired mode closely follows the pattern seen in hearing-impaired listeners in psychoacoustical experiments.

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# Suspension Seat Adjustment Parameters and their Influence on Measured Vibration Response: Preliminary Results.

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## INTRODUCTION

It is well known that the use of suspension seats which are adequately adapted to specific vehicles can greatly reduce the driver's exposure to whole-body vibration provided that they are properly adjusted such as to optimize the performance of the suspension elements. In recent years, mechanical and air suspension seats have been equipped with several controls , some of them having a direct effect on the seat's ability to attenuate terrain-induced vibration. The driver, forming an integral part of the seat itself, its influence on the seat performance cannot be neglected, nor can the manner in which the seat is adjusted in terms of stiffness (weight control), height, backrest angle and damping. All of these factors are bound to influence the manner by which the cab vibrations are transmitted to the driver and the occurrence of shocks resulting from the seat-driver system hitting the limiting stops. For this reason, it apppears relevant to try to quantify how these various adjustment parameters influence the overall seat performance under various types of excitations, in an attempt to set guidelines for the users of suspension seats as to how exactly they should adjust their seat to get the most performance out of their product. At the same time, these results are used in order to validate a model to predict the behaviour of a seat under various conditions.

## METHODOLOGY

Three types of mechanical and air suspension seats have been selected for the tests, although the results are presented here only for the ISRI model 6000 mechanical suspension seat. This seat includes a cross-linkage mechanism, a helicoidal spring and an inclined shock absorber. Seat adjustments are provided for weight, backrest angle, height, fore-aft position, cushion angle, armrest angle and for locking the longitudinal isolator. The seat is loaded with a rigid mass of 63.6 kg, consisting of sand bags, although tests using human subjects are anticipated in the future. For the testing, the weight setting is adjusted at three positions corresponding to mid-ride, 25 mm below mid-ride(low) and 25 mm above mid-ride (high). The total stroke of the suspension is 80 mm. Other adjustments include a backrest angle of 0 and 14 degrees, and a height setting of 275 and 300 mm. The longitudinal isolator is blocked during the tests.

The seat-mass system is installed on a vertical vibration simulator consisting of a platform driven by two hydraulic cylinders having a stroke of 200 mm. This system is driven by a hydraulic pump having a 19 l/min capacity at 21 MPa. The simulator responds to displacement signals up to 25 Hz with a 2 $ms^{-2}$ acceleration achievable only for frequencies exceeding 1.5 Hz. Various safety features are incorporated on the simulator to allow the testing to be conducted using human subjects.

Various excitation signals are used to conduct the tests. These consist of a sinusoidal log sweep excitation providing a constant 25 mm displacement in the 0.5 to 2 Hz range and a constant peak acceleration of 3.95 $ms^{-2}$ in the 2 to 10 Hz range, classes 1 and 2 random excitations designed for agricultural tractors in the ISO 5007 technical report [1], classes I and II random excitations defined in the French standard AFNOR E 58-074 [2] and an IRSST random excitation recorded on fork-lift trucks. These displacement signals are digitized and drive the simulator using a computer.

For the various seat adjustment combinations and the various excitations, vibration signals are recorded using accelerometers fixed on the simulator platform, at the suspension and at the load-seat interface. At each measurement point, the frequency spectrum is recorded in the 0 to 25 Hz range. Using these, the overall unweighted acceleration was computed in the 0.5 to 20 Hz range for each of the settings and measurement positions. The weighted acceleration was then computed, first using the vertical weighting defined in the current ISO 2631/1 standard [3] from 1 to 20 Hz, then using the newly proposed weighting $W_b$ from 0.5 to 20 Hz as presented in the new draft proposal [4] for replacement of the current ISO 2631/1. From these, the "Seat Effective Amplitude Transmissibility" (S.E.A.T.) is formed, representing the ratio of the overall weighted acceleration measured on the seat or the suspension to that measured at the base on the platform. This factor provides an estimate of the seat's ability to attenuate whole-body vibration.

Finally, the suspension and the seat's transmissibility are formed representing the ratio of the frequency spectrum measured at the suspension and on the seat to that measured at the base in the range 0 to 25 Hz. From these, the resonant frequency of the mass-seat system and the transmissibility at resonance are determined for each measurement condition.

## RESULTS

Tables 1 to 3 list the maximum and minimum S.E.A.T. values measured at the suspension and at the seat for different types of excitation when the weight setting is adjusted at mid-ride and 25 mm above and below mid-ride. The S.E.A.T. values are obtained by applying the ISO 2631/1 weighting for vertical vibration appearing in the current version of the standard. The variation of S.E.A.T. values appearing in the tables may be attributed to the variations in height and backrest angle adjustments since these were the only parameters changed between low and high values. These include a series of four tests conducted at a height of 275 and 300 mm and a backrest angle of 0 and 14 degrees.

**Influence of weight setting**
A comparison of the S.E.A.T. values appearing in the three tables shows that values are significantly lower when the seat is adjusted according to the mass on the seat (i.e. mid-ride). The performance of the seat can be greatly deteriorated if the weight setting is not adjusted properly. For example, under ISO 1 excitation, the S.E.A.T. value would range from 0.398 at mid-ride to 0.706 when made too stiff, constituting an increase of 77%. Under Class I excitation, the value could range from .758 at mid-ride to values exceeding 1.0, making the seat unacceptable for this class of vehicle.

**Influence of type of excitation**
The lowest S.E.A.T. values occur using ISO 1 excitation while the highest occur using Class I, and this for all three weight settings. This is understandable considering that ISO 1 excitation peaks at 3.15 Hz while Class I peaks at 2 Hz. The resonant frequency of the mass-seat lying between 1.2 and 1.4 Hz, very little attenuation is provided at 2 Hz, while attenuation improves at higher frequencies. These results illustrate quite nicely that a seat which is effective in one vehicle will not necessarily be effective in another vehicle characterized by a different vibration class.

**Influence of height and backrest angle**
It is difficult to dissociate the combined effect of height and backrest angle on the S.E.A.T.. This effect is shown clearly in the tables and explains the % variation between the minimum and maximum S.E.A.T. values. Depending on the excitation and the measurement position (seat or suspension), variations on the order of 10 to 50% may be noted, which would have a significant effect on the estimated performance of the seat and the suspension.

Table 1. Maximum and minimum S.E.A.T. values measured at the suspension and the seat at mid-ride

| Excitation | | S.E.A.T. at suspension | | | S.E.A.T. at seat | | |
|---|---|---|---|---|---|---|---|
| Type | $a_w(ms^{-2})$ | Min. | Max. | % variation | Min. | Max. | % variation |
| Sinusoidal | 2.02 | .391 | .475 | 21.5 | .452 | .504 | 11.5 |
| ISO 1 | 2.26 | .382 | .431 | 12.8 | .398 | .471 | 18.3 |
| ISO 2 | 1.58 | .566 | .596 | 5.3 | .591 | .669 | 13.2 |
| Class I | 1.84 | .701 | .759 | 8.3 | .714 | .758 | 6.2 |
| Class II | 1.59 | .646 | .660 | 2.2 | .661 | .713 | 7.9 |
| IRSST | 1.45 | .598 | .652 | 9.0 | .617 | .644 | 4.4 |

Table 2. Maximum and minimum S.E.A.T. values measured at the suspension and the seat 25 mm below mid-ride

| Excitation | | S.E.A.T. at suspension | | | S.E.A.T. at seat | | |
|---|---|---|---|---|---|---|---|
| Type | $a_w(ms^{-2})$ | Min. | Max. | % variation | Min. | Max. | % variation |
| Sinusoidal | 2.02 | .493 | .586 | 18.9 | .556 | .654 | 17.6 |
| ISO 1 | 2.26 | .406 | .470 | 15.8 | .424 | .546 | 28.8 |
| ISO 2 | 1.58 | .599 | .657 | 9.7 | .633 | .747 | 18.0 |
| Class I | 1.84 | 1.02 | 1.11 | 8.8 | 1.06 | 1.30 | 22.6 |
| Class II | 1.59 | .967 | 1.01 | 4.4 | 1.05 | 1.19 | 13.3 |
| IRSST | 1.45 | .652 | .682 | 4.6 | .685 | .716 | 4.5 |

Table 3. Maximum and minimum S.E.A.T. values measured at the suspension and the seat 25 mm above mid-ride

| Excitation | | S.E.A.T. at suspension | | | S.E.A.T. at seat | | |
|---|---|---|---|---|---|---|---|
| Type | $a_w(ms^{-2})$ | Min. | Max. | % variation | Min. | Max. | % variation |
| Sinusoidal | 2.02 | .716 | 1.05 | 46.6 | .753 | .939 | 24.7 |
| ISO 1 | 2.26 | .635 | .729 | 14.8 | .706 | .833 | 18.0 |
| ISO 2 | 1.58 | .788 | .965 | 22.5 | .927 | 1.19 | 28.4 |
| Class I | 1.84 | 1.18 | 1.20 | 1.7 | 1.16 | 1.22 | 5.2 |
| Class II | 1.59 | 1.10 | 1.15 | 4.5 | 1.09 | 1.20 | 10.1 |
| IRSST | 1.45 | .969 | 1.03 | 6.3 | .904 | 1.14 | 26.1 |

## CONCLUSION

This preliminary study , although limited since only a rigid load was used, provides some insight on the various factors likely to influence the performance of a suspension seat under different types of excitation. It is seen that the height and backrest angle do affect the overall vibration transmissibility to some extent but not necessarily in an orderly manner. On the other hand, the type of excitation and the stiffness or weight adjustment do have a significant effect on the overall seat performance, where depending on the setting, the S.E.A.T. value can be reduced by as much as 70% with respect to some other setting.

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[2] Norme expérimentale AFNOR E 58-074 (1981). Engins de terrassement-Spécifications pour le mesurage en laboratoire des vibrations verticales transmises au conducteur par l'assise du siège. 10pp.

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[4] Fourth Committee Draft on ISO/CD 2631 (1993). Mechanical Vibration and Shock-Guide to the evaluation of human exposure to whole-body vibration. 27pp.

# RESPONSE OF THE HAND-ARM SYSTEM TO VIBRATION

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A complex of vascular, neurological and musculo-skeletal disturbances, often referred to as the hand-arm vibration syndrome, may occur among operators of hand-held power tools. The driving-point mechanical impedance of the hand-arm has been extensively investigated to enhance an understanding of its biodynamic response to vibration excitations, and to permit development of effective vibration isolators. Although, the impedance characteristics of the hand-arm have, invariably, been measured on human subjects under carefully controlled test conditions, considerable deviations exist among the impedance data reported by different investigators. These deviations may be attributed to the strong dependence of the biodynamic response of the hand-arm on: (*i*) grip and thrust forces; (*ii*) posture; (*iii*) anthropometric parameters of the hand-arm; (*iv*) amplitude and nature of vibration excitation; and (*v*) the nonlinear properties of the biological materials.

In this study, the test procedures and measured driving-point mechanical impedance data reported in published studies are analyzed to synthesize the envelopes of mean values of impedance magnitude and phase, as a function of the excitation frequency. The weighted mean values, derived for each of the three orthogonal directions of vibration specified by ISO 5349 [1], are considered to characterize the idealized driving-point impedance of the human hand-arm system. A lumped-parameter four-degrees-of-freedom (DOF) model of the hand-arm system is developed using the *idealized* impedance characteristics in conjunction with a nonlinear programming based optimization algorithm.

## SYNTHESIS OF DRIVING-POINT IMPEDANCE DATA

Driving-point mechanical impedance, widely used to describe the dynamic properties of the hand-arm system, is defined as the ratio of the complex driving force to the complex velocity measured at the driving point. Mechanical compliance and accelerance have also been employed to describe the biodynamic response of the hand-arm system. The mechanical impedance, compliance or accelerance properties of the human hand-arm have been measured using different experimental methods, number of male subjects, amplitude and nature of vibration excitations, grip force, response variables, handle size and elbow angle. The range of test variables, employed in various studies, are summarized below:

| | |
|---|---|
| Direction of Vibration: | $X_h$ and / or $Y_h$ and / or $Z_h$ |
| Test Handle: | Circular (Diameter 19-45 mm)<br>Elliptical (31x 42 mm) |
| Grip Force: | Constant magnitude (10-200 N) |
| Frequency Range: | From 8-200 Hz to 20-2000 Hz |
| Nature of Vibration: | Sine sweep, pseudo-random, random or impulse of varying magnitudes. |
| Response Variable: | Impedance, compliance, accelerance |
| Elbow Angle: | Ranging from 50-180 degrees |
| Number of Subjects: | Ranging from 1 to 75 |

In view of above variations and the expected discrepancies among the results, the driving-point impedance characteristics reported for similar test conditions were analyzed to derive the range of *idealized* values of impedance magnitude and phase. Based upon the most common range of variables used in different studies, 9 impedance data reported in the published studies [2-9], were selected for synthesis. The reported data were selected when: (*i*) both magnitude and phase were reported in the 20-500 Hz frequency range; (*ii*) data was reported for either one or more orthogonal axes conforming with ISO 5349; (*iii*) the grip force was in 25-50 N range; and (*iv*) elbow angle was close to 90°.

The driving-point impedance data, reported in selected studies, are compared for all three orthogonal axes. In order to compare the data, the reported compliance data was converted to the equivalent impedance. Figure 1 illustrates the differences and similarities among the magnitude and phase data reported for $Z_h$ direction. The two data sets reported by Gurram [8], and Reynolds [6] were obtained using sine-sweep and random excitations, and different handle sizes, respectively. Although most data sets exhibit a impedance magnitude peak in the 20-50 Hz frequency band, the magnitude data reported in [2,4,9] differ considerably from the entire ensemble in the frequency range 20-50 Hz, and above 500 Hz. While the impedance phase data does not indicate a definite trend, the data reported in [4,7,8 (random excitation only)] evidently form the outliers.

### Sources of Variability

Although all the studies included in this analysis employed similar range of vibration frequencies, closely controlled grip forces, elbow angles, direction of vibration, and four or more male subjects, the comparison of the reported data revealed large variations in the impedance magnitude and phase. While these variations are quite disturbing and lead to questions concerning the validity of measurement procedures, it is extremely difficult to identify the sources of variability. The variations in hand-arm impedance evidenced in one investigation due to changes in a single parameter are therefore examined to determine some of the sources of variability, summarized below:

- Impedance magnitude measured under random vibrations is similar to that measured under sinusoidal excitations, while the phase response measured under random excitations tends to be somewhat greater [8].
- The $X_h$ component of impedance is relatively insensitive to magnitude of sinusoidal excitation at frequencies above 250 Hz, the impedance magnitude and phase at lower frequencies however vary with the level of excitation [2].
- The impedance magnitude measured in all directions using a large diameter handle tends to be larger than that measured using a smaller diameter handle. The influence of handle size on the phase response, however, is insignificant [6].

### Most Probable Values of Human Hand-Arm Impedance

The differences in mean values of impedance, and the patterns of agreements or disagreements among various studies serve to justify the exclusion of data that form the outliers from the synthesis. The range of most probable values of impedance magnitude and phase of the human hand-arm are derived from the envelopes of the mean values reported in selected studies for all three axes in the 10-1000 Hz frequency range. The envelop curves, smoothened using segmental cubic spline functions, define the range of mean values of all the selected data sets and may be considered to characterize the range of *idealized* values of the human male hand-arm impedance in all three directions. Figure 2 illustrates the range of $Z_h$ component of *idealized* or most probable impedance magnitude and phase values. The impedance data that fall within the range of *idealized* values may be considered acceptable representations of the human hand-arm impedance. The weighted mean values of data sets included in this analysis are computed and illustrated as the central dotted curve in Fig. 2.

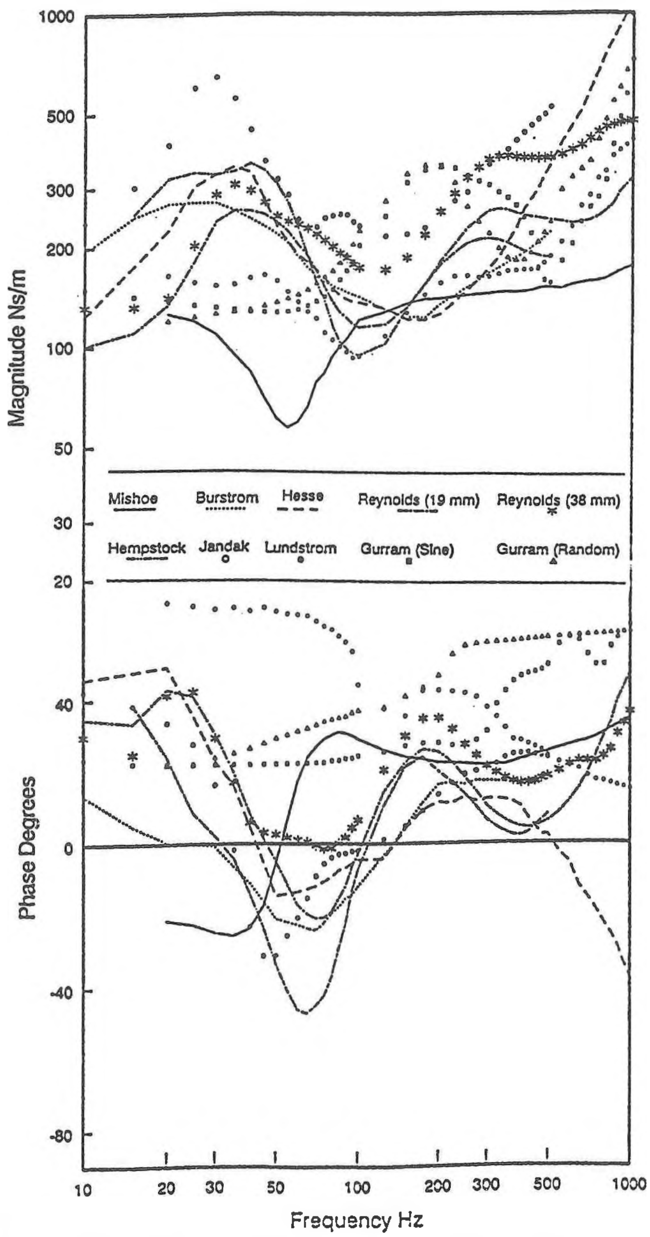


Fig. 1: Comparison of $Z_h$ component of driving-point impedance.

**Hand-Arm Impedance Model**

A four-DOF lumped-parameter model of the hand-arm system is formulated for each orthogonal axis using the *idealized* impedance magnitude and phase values. A constrained optimization function, comprising the magnitude and phase erros between the *idealized* and model response, is formaulated for excitations in the 10-1000 Hz frequency range [8]. A sequential search algorithm is used to derive the hand-arm impedance model parameters for each orthogonal axis. A comparison of the model response (dashed curve) with the *idealized* values, shown in Fig. 2, reveals good agreement in both the impedance magnitude and phase response.

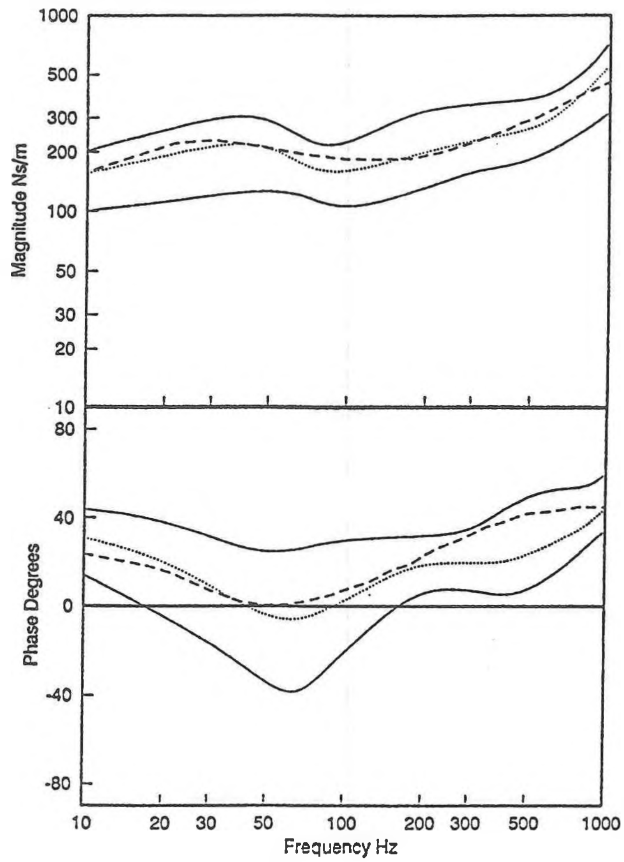


Fig. 2: Mean, range of *idealized* values, and the model response of impedance in the $Z_h$ direction.

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# THE EFFECT OF MECHANICAL SHOCK FREQUENCY AND AMPLITUDE ON SPINAL TRANSMISSION AND INTERNAL PRESSURE

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## 1. Introduction

When the human body is subject to vibration or impact, it demonstrates a dynamic response. The displacement of tissues and the forces transmitted by them alters as a function of time. A useful method to assess the potentially harmful effects is to measure the relative displacements and hence stresses of different regions of the body in response to vibration amplitude and frequency. Transmission of acceleration can be expressed in terms of a transfer function that defines the relative magnitude and phase relationship of the output acceleration in a particular region (for example, the spine) compared with the input acceleration (for example, at the seat). Knowledge of acceleration transfer functions provide insight into behaviour of the body sub-systems, and enables assessment of input acceleration levels and frequencies where a particular tissue is more likely to be damaged. A substantial body of knowledge has been reported concerning the transmission of vibration. However, little is known about the repeated impact environment and the dynamic response of individual body segments to vibration and impact. This paper reports some initial findings of the spinal and internal pressure responses to impact accelerations at the seat.

## 2. Methods

To determine health indices that could be used to develop a dose-effect model for minimizing the effects of repeated impact in army vehicles, a series of pilot experiments were conducted using the multi-axis ride simulator (MARS) at Fort Rucker, Alabama. These involved a series of seven 5.5 minute exposures to various shock frequencies, ranging from 2 to 11 Hz, and shock amplitudes, ranging from 0.5 to 3.0 g for each biodynamic axis (+x, +y and -z), with approximately 5 minutes rest between exposures. Each biodynamic axis was tested on a seperate day.

Ten male subjects between 20 and 40 years of age and within one standard deviation of the mean for height and weight, based on standard military data, were recruited from U.S. Army personnel assigned to Fort Rucker. All had experience with motion, either tactical ground vehicles (TGVs) or air transport. A solid metal seat with a bean-bag cushion and no backrest was securely mounted on the MARS. The seat was adjusted so the subject's feet rested comfortably on the MARS table with the knees and hips at approximately 90°.

Acceleration was measured at the seat and over the spinous process of lumbar (L2-4) and thoracic (T1-3) vertebrae using miniature Entran accelerometers (weight 0.3 gm; range of ± 10 g or ± 25 g). A skin to vertebrae inverse transfer function was generated using the method of Hinz et al. (1986) and applied to z- and y-axis spinal accelerations to correct measured accelerations for the elastic properties of the skin and subcutaneous tissues. The corrected accelerations represent motion of the vertebrae. Acceleration data were bandpass filtered at 0.5 to 60 Hz to eliminate baseline drift and high frequency noise.

Internal pressure was measured by a specially constructed rectal pressure probe, 50 cm in length and terminated in an Entran (model EPB-140W-5S) miniature pressure transducer (range ± 5 psi). The transducer and wiring were covered with 2 layers of heat shrink tubing 20 cm in length to provide a suitable degree of strength and flexibility. The probe was inserted by the subject to a depth of 15 centimeters beyond the anal sphincter. Internal pressure data were high pass filtered at 0.5 Hz to remove baseline fluctuations, and low pass filtered at 60 Hz to conform with seat acceleration data.

An acceleration peak detection and transmission ratio program was written to identify discrete shock events. The positive peak value of seat acceleration was first identified, followed by the acceleration minima (negative peak) occurring within a prescribed shock window of 250 msec. The program then analyzed spinal acceleration and internal pressure data to identify a corresponding positive and negative peak occurring within each shock window, and the time delay between the seat shock and peak response. Transmission ratios between seat acceleration and internal pressure were expressed as peak output pressure to acceleration input in units of $mmHg.s^2.m^{-1}$

## 3. Spinal Transmission of Shocks

Despite a range of responses between individuals, the shape of the mean transmission curves for seat to lumbar spine, seat to thoracic spine, and seat acceleration to internal pressure were remarkably similar for a given shock axis. There were, however, differences in magnitude and shape of acceleration responses between axes (x, y and z). The largest spinal responses were to -z-axis shocks. Maximal responses in the -z-axis were not, however, due to the initial shock input (the seat dropping out from below the subject), but rather to the secondary impact when the seat and subject were reunited. Acceleration responses to +y-axis shocks were higher in magnitude than to +x-axis shocks.

Spinal transmission curves for the x- and z- axis response to +x-axis shocks demonstrated non-linearity. At 3 g, there was a clear peak at 5 Hz in both the thoracic and lumbar spine and a smaller peak noticeable at 8 Hz. At 2 g and 1 g, however, the peak response shifted to 4 Hz and there was no second peak at 8 Hz. This pattern is clearly different from the dominant frequency of 1-2 Hz and diminishing response above 2 Hz suggested by ISO 2631 (1982) for sinusoidal vibration. Transmission of shock peaks was highest for 3 g shocks, followed by 2 g and then 1 g shocks. As the spine flexes in response to a shock, there are both x-and z-axis components due to curvature of the spine. The z-axis spinal response to x-axis shock inputs was higher than the x-axis response, likely due to rotation of vertebrae and whip-like motions.

In the y-axis, the shape of spinal acceleration response curves for lumbar and thoracic locations were similar for all shock amplitudes and for both y-and z-axis responses. The non-linearity of the response to x-axis shocks was not apparent for y-axis shocks. The dominant frequency was the lowest measured frequency (4 Hz for 3 g and 2 g; 2 Hz for 1 g). It is impossible to know whether the true dominant frequency is lower than these, since lower shock frequencies could not be measured. ISO 2631 suggests that the dominant frequency for sinusoidal vibration in the y-axis is 1-2 Hz. We cannot tell from this data whether the same is true for response to shocks. The z-axis spinal response to a y-axis shock was very small.

A non-linear response was apparent in the -z-axis, with highest responses recorded for higher amplitude shocks. The shape of spinal acceleration response curves for 3 g and 2 g shocks resembled that of the y-axis, with a dominant frequency of 4 Hz and a decreasing response at higher frequencies. Again, it is impossible to know whether a lower frequency shock will produce a higher response. For 1 g shocks, however, the dominant frequency was 4 Hz, not 2 Hz. This concurs with the ISO 2631 suggestion that the most sensitive frequencies for sinusoidal vibration are 4-8 Hz. The British Standards (BS 6841, 1987), which are meant to accommodate shocks, suggest that frequencies as high as 10 Hz are important in the z-axis. The DRI, designed for single large +z-axis impacts, suggests the dominant frequency is 11 Hz. Results from this study with -z-axis shocks, suggest the dominant frequency is lower than that in current standards.

The x-axis response at the spine to z-axis shocks at the seat was of similar magnitude and shape to the spinal z-axis response. As with x-axis shocks, the z-axis shocks caused a whip-like effect in the body resulting from forward flexion. At low frequencies, the x- and z-axis response at the lumbar spine averaged 2.5 times the seat acceleration. The upper thoracic spine showed a lesser amplification of seat input, approximately half that at the lumbar spine. Clearly, the body is a complex system that does not respond as a single mass to shock inputs.

When a subject impacts the seat following a -z-axis shock, the seat accelerometer registered a very high-frequency component (approximately 150 Hz). A corresponding high-frequency response (30-90 Hz) was transmitted to accelerometers at the spine. Data in the literature, however, suggests that the body damps these high frequency components (Fairley and Griffin, 1989). Our results showed higher spinal responses to 2 and 4 Hz shocks, compared with 11 Hz shocks. However, the much higher frequencies of 30-90 Hz were not being damped by the spine. It might be suggested that these high frequency components are skin movement. However, the vertebra-skin transfer function did not remove these. In addition, examination of internal pressure responses also showed similar high-frequency components. It may be that for a large enough impact the body responds as a non-linear system with transmission of high frequency components.

## 4. Internal Pressure Response to Shocks

The frequency dependence of internal pressure responses were remarkably similar to spinal acceleration responses, with a dominant peak commonly observed at 4 Hz (2 and 3 g) or 2 Hz (1 g) and a secondary peak at 8 Hz (3 g). The largest peak internal pressure response was measured in the z-axis (7.6 $mmHg.s^2.m^{-1}$ at 4 Hz). The lowest peak magnitudes were in response to y-axis shocks (0.86 $mmHg.s^2.m^{-1}$ at 4 Hz), however double peaks were often observed in the y-axis. The magnitude of internal pressure response to a 3 g impact in the z-axis varied from approximately 30 mmHg to 230 mmHg. By contrast, when subjects were asked to cough, they generated internal pressures of approximately 130 mmHg, laughing produced pressures of 80-90 mmHg, bearing down produced instantaneous pressures of 225 mmHg and sustained pressures of 130 mmHg. Some internal pressure magnitudes produced by 3 g shocks exceeded pressures that could be voluntarily produced by subjects.

## 5. Conclusions

The spinal acceleration and internal pressure responses to seat acceleration demonstrate clear non-linearities in the human response to mechanical shocks. The response curves are dependent on both shock magnitude and frequency.

These experiments highlight some limitations of current standards for vibration and shock. New standards for shocks should consider the non-linear response of the body and account for differences in the response to each axis (x, y and z) and different directions in each axes (+ and -). The shape of such a dose-response curve must also be magnitude dependent.

Further studies with shocks of lower frequency (<4 Hz for 3 and 2 g, and <2 Hz for 1 g) and higher magnitude (>3 g) are required to establish complete response curves with well defined maxima. It is also necessary to investigate the response to +z and -x shocks.

## 6. Note

This work was supported by the U.S. Army Medical Research and Development Command under Contract No. DAMD17-91-C-1115. The views, opinions and/or findings contained in this report are those of the authors and should not be construed as an official Department of the Army position, policy or decision unless so designated by other documentation. In the conduct of research where humans are the subjects, the investigators adhered to the policies regarding protection of human subjects as prescribed by 45 CFR 46 (Protection of Human Subjects).

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The Canadian Acoustical Association
L'Association Canadienne d'Acoustique

# MINUTES OF THE BOARD OF DIRECTOR'S MEETING

**10:00 a.m., May 16th, 1993**
**Chateau Laurier Hotel, Ottawa**

| | | | | |
|---|---|---|---|---|
| Present: | D. Chapman | M. Roland Mieszkowski | E. Bolstad | C. Laroche |
| | W. Sydenborgh | D. Jamieson | F. Laville | B. Dunn |
| | A. Behar | T. Nightingale | D. Quirt | |
| | J. Hemingway | R. Ramakrishnan | S. Abel | |

Regrets: M. Hodgson S. Forshaw D. Wicker

The Meeting was called to order at 10:07 hours.

1) President's Report: The CAA Brochure has been updated by M. Mieszkowski and F. Laville and 2,000 copies printed. The Brochure will require updating after the Conference each year. Brochures will be distributed amongst the Officers and Directors and inserted into Canadian Acoustics.

2) Executive Secretary's Report: The Mailing List has now been transferred to WordPerfect, enabling better label printing, better sorting capability and compatibility with the Journal. Of the total Mailing List of 491, 185 have not paid. It was agreed that Final Notices would be sent to all who have not paid. Those remaining unpaid on July 31st will be removed from the Mailing List. It was recommended that use of the PO Box address be minimised, particularly with reference to Prizes.

3) Treasurer's Report: The Association is in good financial health. The Capital Fund has increased, due in part to the non-award of Prizes. Interest Rates on CAA GIC's have dropped, and will likely continue to do so. Difficulty has been experienced bringing Journal Advertising Revenue into proper accounting procedures. New accounting procedures are being discussed with the new Associate Editor in charge of advertising to remedy this situation.

4) Editor's Report: There continues to be a lack of Papers submitted to the Journal. Questions were raised concerning the lateness of the recent issue. It was agreed that this was understandable considering the recent addition to the Editor's family!

5) Membership/Recruitment: A printed proposal regarding the Duties of the Membership Chair person was circulated. The target for membership is 600 members in 5 years. The role of the Directors in grass roots recruitment was discussed.

6) Awards Committee: As well as publication in the Journal, 300 posters have been sent to Universities advertising Prizes. This technique was considered successful in reaching the non-member population.

Motion: "In order to be considered for a Prize, the applicant must be a member of CAA"
Proposed: Bruce Dunn, Seconded: Sharon Abel. Carried

*Directors' Awards:* 4 Papers are eligible for the Directors' Awards this year.

*Bell:*
3 applications were received and a recipient has been selected. An Application Form and Description of the Bell prize was submitted.

Motion: "That the Application Form and Description of the Bell Prize be accepted, with the final sentence modified to read: The winner of the prize will be announced at the Fall meeting of the Canadian Acoustical Association" Proposed: Don Jamieson, Seconded: Sharon Abel, Carried

*Fessenden:*
3 applications, but the winner has not as yet been determined. The winner will be announced at the October Meeting.

*Eckel* :No applications

*Student Oral:*
Local Organizing Committee to be judges. Abstracts and Supervisor's signatures are required, otherwise applications will be rejected.

*Science Fair:*
Many thanks to Michel Brochud for judging on behalf of CAA at this year's Science Fair in Riviere-du-Loup.

*Postdoctoral:*
1 applicant, as the application is of an acceptable standard, the Prize will be awarded this year.

7) Acoustics Week Reports:

*Vancouver, 1992*
At the last minute, the 1992 meeting was switched from Winnipeg to Vancouver. Of the $2,000 seed money, $493.64 was returned to CAA. Student Travel to the symposium was subsidised. 58 Registrants and 13 Students attended.

*Toronto, 1993*
Seminars will not be directly organized by the local committee. 2 Plenary Sessions are planned, plus 3 parallel Invited Sessions. On the Friday, John Manuel will Chair a Session entitled "Origins of CAA". There will be no awards at the Banquet, these will be at a special Awards Lunch on the Friday. The Board of Directors Meting will be at 1:00 pm on the Tuesday. Student travel will not be subsidised as no formal CAA policy exists.

> Motion: "Ramani Ramakrishnan and Alberto Behar will produce a proposal for Student travel subsidy for presentation at the October Board of Directors meeting". Proposed: Ramani Ramakrishnan, Seconded: Alberto Behar, Carried

*Ottawa, 1994*
The Symposium will be from October 18th to 20th at the Citadel Inn. A Vibration Seminar is planned. An Exhibition will be held. A guided tour of the IRC Acoustics Facility is planned for the Friday.

*1995*
Either Windsor or Montreal/Sherbrooke/Quebec City were suggested.

8) Nomination Committee: Several suggestions for Officers and Directors were made.

9) Other Business: None

10) By-Law Review: A draft By-Law was submitted for discussion. As many comments were forthcoming, it was agreed that comments in writing should be sent to Winston Sydenborgh or Alberto Behar by July 1st, 1993.

The Meeting was adjourned at 15:00 hours.

# NEWS/INFORMATIONS

## CONFERENCES

**7th International Symposium in Audiological Medicine 1993:** September 19 - 2, 1993, Cardiff, Wales, UK. Genetic Hearing Loss, Training in Audiology for Primary Physicians, Investigation of Vertigo - how and why? Details: Dr. D. Stephens, Welsh Hearing Institute, University Hospital of Wales, Cardiff CF4 4XW, Wales

**126th Meeting of the Acoustical Society of America:** October 4 - 8, 1993, Denver, Colorado, USA. Contact: Acoustical Society of America, 500 Sunnyside Boulevard, Woodbury, NY 11797.

**November 9 - 10, 1993:** "Progress in acoustics, noise and vibration control," Australian Acoustical Society Annual Conference, Glenelg, SA, Australia. Contact: AAS Annual Conference 1993, Dept. of Mechanical Engineering, University of Adelaide, GPO Box 498, Adelaide, SA 5001, Australia. Fax: +61 8 2240464.

**Third French Congress on Acoustics**: Toulouse (France), May 2 - 6, 1994. Mail should be sent to: Secrétariat du Troisième C.F.A., Université Toulouse-le-Mirail, FRANCE, Tel. (33) 61 50 44 68, Fax. (33) 61 50 42 09.

**127th Meeting of the Acoustical Society of America:** August 23-25, 1994, Cambridge, Massachusetts, USA. Contact: Elaine Moran, Acoustical Society of America, 500 Sunnyside Blvd., Woodbury, NY 11797, USA. Tel. +1 (516) 576-2360, Fax. +1 (516) 349-7669.

**5th Western Pacific Regional Acoustics Conference:** August 23 - 25, 1994, Seoul, Korea. Contact: Conference Secretariat, Tel. +82 2 361-2783, Fax. +822 365-4668.

**INTER-NOISE 94:** The 1994 International congress on Noise Control Engineering, Yokohama, Japan, from August 29 to 31, 1994. Contact: Inter-Noise 94 - Congress Secretariat, Sone Lab. R.I.E.C., Tohoku University, 2-1-1 Katahira, Aoba-Ku, Sendai, 980 Japan. Fax: +81-22 263-9848, 81-22-224-7889. E-Mail: in94@riec.tohoku.ac.ip.

**128th Meeting of the Acoustical society of America:** November 28 - December 2, 1994, Austin, Texas, USA. Contact: Elaine Moran, Acoustical Society of America, 500 Sunnyside Blvd., Woodbury, NY 11797, USA. Tel. +1 (516) 576-2360, Fax. +1 (516) 349-7669.

**INTER-NOISE 95:** July 10-12, 1995, Newport Beach, California, USA. Contact: Intstitute of Noise Control Engineering, P.O. Box 3206, Arlington Branch, Poughkeepsie, NY 12603, USA. Tel. (914) 462-4006, Fax. (914) 473-9325.

## CONFERENCES

**7e symposium international d'audiologie**: Cardiff, pays de Galles, Grande-Bretagne, du 19 au 22 septembre 1993. Sujets à l''ordre du jour: la perte auditive d'origine génétique, la formation en audiologie pour médecins, les vertiges. Renseignements: D. D. Stephens, Welsh Hearing Institute, University Hospital of Wales, Cardiff CF4 4XW, Wales.

**126e rencontre de l'Acoustical Society of America**: Denver, Colorado, du 4 au 8 octobre 1993. Renseignements: Acoustical Society of America, 500 Sunnyside Boulevard, Woodbury, NY 11797, USA.

**Conférence annuelle de la Australian Acoustical Society:** Glenelg, Australia, les 9 et 10 novembre 1993, sur le thème de l'évolution des techniques d'acoustique et de maîtrise du bruit et des vibrations. Renseignements: AAS Annual Conference 1993, Department of Mechanical Engineering, University of Adelaide, GPO Box 498, Adelaide, SA 5001, Australie; télécopieur +61 8 2240464.

**3e Congrès français d'acoustique:** Toulouse, France, du 2 au 6 mai 1994. Renseignements: Secrétariat du Troisième C.F.A., Université Toulouse-Le-Mirail (C.P.R.S.), 5, allée Antonio Machado, 31058 Toulouse Cédex, France. Téléphone (33) 61 50 44 68; télécopieur (33) 61 50 42 09.

**127e rencontre de l'Acoustical Society of America:** Cambridge, Massachesetts, du 5 au 9 juin 1994. Renseignements: Elaine Moran, Acoustical Society of America, 500 Sunnyside Boulevard, Woodbury, NY 11797, USA. Téléphone (516) 576-2360; télécopieur (516) 349-7669.

**5e conférence des pays du Pacifique ouest sur l'acoustique:** Séoul, Corée, du 23 au 25 août 1994. Renseignements: Elaine Moran, Acoustical Society of America, 500 Sunnyside Boulevard, Woodbury, NY 11797, USA. Téléphone (516) 576-2360; télécopieur +82 2 365-4668

**Conférence Inter-Noise 94**: Yokohama, Japon, du 29 au 31 août 1994. Renseignements: Inter-Noise 94, Congress Secretariat, Sone Lab. R.I.E.C., Tohoku University, 2-1-1 Katahira, Aoba-Ku, Sendai, 980 Japon. Télécopieur +81-22-263-9848; +81-22-224-7889; courrier électronique en 94 @ riec.tohoku.ac.ip.

**128e rencontre de l'Acoustical Society of America:** Austin, Texas, du 28 novembre au 2 décembre 1994. Renseignements: Elaine Moran, Acoustical Society of America, 500 Sunnyside Boulevard, Woodbury NY 11797, USA. Téléphone (516) 576-2360; télécopieur (516) 349-7669.

**Conférence Inter-Noie 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 (914) 473-9325.

## COURSES

**Industrial Hygiene:** September 27-October 1, 1993, University of Toronto, The Department of Chemical Engineering and Applied Chemistry. Mail to: Julie Mendonca, Department of Chemical Engineering & Applied Chemistry, Unversity of Toronto, 200 College Street, Toronto, Ontario, M5S 1A4.

**Vibration Seminars:** Goldman Machinery Dynamics Corp. is offering 3-day seminars on machinery vibration in upstate New York on September 13-15, 1993. For further information, contact: Steve Goldman, Goldman Machinery Dynamics Corp., 6 Mallard Dr., W. Nyack, NY 10994, USA. Tel. (914) 634-0674.

**Strategies and Techniques in Noise Control**: October 28, 1993. The 1986 Draft Regulation Prescribing Noise as a Designated Substance, stresses engineering controls, rather than HPD's to reduce worker noise exposures below 90 dBA. Although this legislation is still "pending", we know from the nature of the "holdup" that the requirements of the Regulation (when enacted) will not be any less. This session will review some basic physics of sound and then evaluate: noise sources and spectra; sound power and sound pressure levels, instrumentation for sound and vibration measurement and analysis, sound level standards and surveys: ANSI, CSA, MOL; noise control strategies: vibration isolation, surface treatments, barriers and enclosures. There will be a number of calculations of predicted noise reduction: attendees are requested to bring a calculator to this session. Attendance at this session will facilitate in-house noise solutions and create a more knowledgeable consumer of noise control produces and services.

**Industrial Audiometry and the Effective Hearing Conservation Program**: December 8-10, 1993. The 1986 Draft Noise Regulation stipulates that where workers are exposed to a daily TWA noise exposure of 85 dBA or greater, a Hearing Conservation Program (HCP) is required; where a weekly TWA is 85 dBA or greater, the HCP must include audiometric tests. This three-day course will provide the background necessary to introduce an effective HCP into your workplace. As well, it will offer the specific training required to meet the definition of "competent audiometric tester" as specified by the Code for Audiometry of Noise Exposed workers. Curriculum will include anatomy of the ear, noise-induced hearing loss, workers' compensation, ethics of audiometry hearing protection, legal requirements, etoscopy and audiometry laboratories. Participants are encouraged to bring their own audiometer, if one is available in-house. For further information contact Dr. Alan D. Stuart, summer Program Coordinator, The Penn State Graduate Program in Acoustics, P.O. Box 30, State College, PA 16804 Tel: (814) 863-4128 or FAX (814) 865-3119.

**Applied Noise & Vibration Control**: November 16-19 in Chicago, ILL. Contact: Education Section, ASHRAE, 1791 Tullie Circle NE, Atlanta, GA 30329, Tel: (404) 636-8400, Fax: (404) 321-5478.

## COURS

**Industrial Hygiene:** université de Toronto, du 27 septembre au 1 octobre 1993. Renseignements: Julie Mendonca, Department of Chemical Engineering and Applied Chemistry, University of Toronto, 200 rue College, Toronto, (Ontario) M5S 1A4.

**Vibration Seminar:** séminaire d'une durée de 3 jours sur les vibrations produites par les machines, due 13 au 15 septembre 1993. Renseignements: Steve Goldman, Goldman Machinery Dynamics Corp., 6 Mallard Dr., West Nyack, New York 10994, U.S.A. Téléphone (914) 634-0674

**Strategies and Techniques in Noise Control:** le 28 octobre 1993. Le projet de loi américain de 1986 qualifiant le bruit de substance désignée privilégie les moyens extérieurs techniques, aux dépens des protecteurs auriculaires individuels, pour amener l'exposition des travailleurs sous le seuil des 90 dBA. Bien que ce texte ne soit pas encore devenu loi, nous savons, de par les raisons du retard de son adoption, que les exigences de la loi, une fois celle-ci adoptée, seront aussi sévères, sinon plus, que celles du projet de loi. Ce séminaire d'une journée passera en revue les sujets suivants; physique fondamentale du bruit; sources et spectres des bruits; puissance sonore et niveaux de pression sonore; instruments de mesure et d'analyse des sons et des vibrations; normes ANSI, CSA et MOL de niveau sonore; mesures de niveau sonore; stratégies de réduction du bruit; isolation des vibrations, traitements des surfaces, barrières et enccintes. Les participants seront appelés à estimer des niveaux de bruit après réduction (il est conseillé d'apporter sa propre calculatrice). Le but du sèminaire est d'aider les participants à élaborer d'eux-mêmes des stratégies de réduction du bruit et à prendre des décisions éclairées lors du choix de produits et de services pour la réduction du bruit. Renseignments: Dr. Alan D. Stuart, Summer Program Coordinator, The Penn State Graduate Program in Acoustics, P.O. Box 30, State College, P.A. 16804, USA; téléphone (814) 863-4128, télécopieur (814) 865-3119.

**Industrial Audiometry and the Effective Hearing Conservation Program:** du 8 au 10 décembre 1993. Le projet de loi américain de 1986 sur le bruit stipule l'existence d'un programme de préservation de l'ouïe sur les lieux de travail où l'exposition au bruit est de 85 dBA et plus, et des tests audiométriques lorsque l\exposition est de 85 dBA et plus. Ce cours de trois jours devrait fournir aux participants les éléments de base nécessaires à la mise sur pied d'un programme de préservation de l'ouïe sur leurs lieux de travail. Les participants recevront la formation d'évaluateur compétent en audiométrie, telle que prescrite dans le Code for Audiometry of Noise Exposed Workers. Les sujets suivants seront abordés: anatomie de l'oreille, perte d'audition due au bruit, audiométrie, protection de l'ouïe, programmes d'indemnisation des travailleurs, règlements, laboratoires d'étoscopie et d'audiomètrie. On demande aux participants d'apporter, si possible, laur propre audiomètre. Renseignements: Dr. Alan D. Stuart, Summer Program Coordinator, The Penn State Graduate Program in Acoustics, P.O. Box 30, State College, PA 16804, U.S.A., téléphone (814) 863-4128, télécopieur (814) 865-3119.

**Applied Noise and Vibration Control:** Chicago, Illinois, du 16 au 19 novembre 1993. Renseignements: Education Section, ASHRAE, 1791 Tullie Circle, NE, Atlanta, GA 30329, téléphone (404) 636-8400, télécopieur (404) 321-5478.

## NEW PRODUCTS

**Elsevier's Dictionary of Noise and Noise Control,** Robert Serre, Editor, Elsevier Science Publishers, B.V., Sara Burgerhartstraat 25, P.O. Box 211, 100 AE Amsterdam, The Netherlands. This book is 206 pages which includes definitions in the English language of 1175 keywords used in noise and noise control. There is a large number of cross-reference to the keywords.

The final series of proposed changes intended for the inclusion in the 1995 editions of the National Building Code, the National Fire Code the canadian Farm Building Code and the Canadian Plumbing Code will be released for public reivew August 3, 1993. It will be made available in a document titled "National Building and Fire Codes of Canada 1990, National Farm Building and Building Codes 1990; Final Series of Proposed Changes." Copies can be obtained free of charge from:

Public Review
Canadian Codes Centre
Institute for Research in Construction
National Research council of Canada
Ottawa, Ontario
K1A 0R6

or for further information you may contact:

Canadian Codes Centre
Tel. (613) 993-9960
Fax. (613) 952-4040

## NOUVEAUX PRODUITS

La maison Elsevier vient de pulbier un dictionnaire sur le bruit et les techniques de lutte contre le bruit. Le **Elsevier's Dictionary of Noise and Noise Control** compte dans ses 206 pages 1175 définitions, avec renvois multiples, de termes fréquemment utilisés dans la domaine. On peut se procurer se dictionnaire auprès de Elsevier Science Publishers B.V., Sara Burgerhartstraat 25, P.O. Box 211, 100 AE Amsterdam, Pays-Bas.

A partir du 3 août 1993, le public pourra examiner la dernière série de modifications proposées pour les éditions 1995 du code national du bâtiment, du Code national de prévention des incendies, du code du bâtiment agricole du Canada et du Code canadien de la plomberie. On peut se procurer gratuitement le texte de ces modifications en en faisant la demande à l'addresse suivante:

Examen du public
Centre canadien des codes
Institut de recherche en construction
Conseil national de recherches du Canada
Ottawa, Ontario
K1A 0R6

Renseignments:

Centre canadien des codes
Téléphone (613) 993-9960
Télécopieur (613) 952-4040

# The Canadian Acoustical Association
# l'Association Canadienne d'Acoustique

## ANNUAL GENERAL MEETING/ASSEMBLE GENERALE ANNUELLE

| | |
|---|---|
| Date / Date: | October 7 / 7 octobre 1993 |
| Time / Heure: | 3:20 P.M. / 15h20 |
| Place / Endroit: | Delta Chelsea Inn<br>Toronto, Ontario |

## AGENDA / ORDRE DU JOUR

1. Welcome (D. Chapman)
2. Review of Minutes of the last AGM (D. Chapman)
3. President's Report (D. Chapman)
4. Executive Secretary's Report (J. Hemingway)
5. Treasurer's Report ( E. Bolstad)
6. Editor's Report, Canadian Acoustics (M. Hodgson)
7. Membership/Recruitment Report (W. Sydenborgh)
8. Awards Committee Report (B. Dunn)
9. Meeting Reports

   AWC '93 Toronto (S. Abel)
   AWC '94 Ottawa (T. Nightingale)
   AWC '95 ?
10. Other Business

    Arising from the Board of Directors Meeting
    New Business from the floor
11. Elections (B. Dunn)

# THIRD INTERNATIONAL CONGRESS ON AIR- AND STRUCTURE-BORNE SOUND AND VIBRATION

June 13-15, 1994
Hotel Le Chateau Champlain
Montréal, CANADA

The Third International Congress on Sound and Vibration is the third in the series. The first and second were held in the USA in March 6-8, 1990 and March 4-6, 1992. There will be a program of invited and contributed papers. In addition there will be several keynote papers by experts in important areas covered in the Congress. The Congress is sponsored by Concordia University, Montréal, Canada in cooperation with the Canadian Acoustical Association and the Acoustical Society of America and societies or institutes in many other countries. The programme chairman is Professor Richard Guy (Montréal), the scientific chairman is Dr. G. Krishnappa (Ottawa) and Dr. Malcolm J. Crocker (Auburn, USA) is the general chairman.

Contributed papers covering theoretical and experimental research in the following areas are solicited:

* *Aeroacoustics and Atmospheric Sound*
* *Sound Intensity*
* *Modal Analysis*
* *Statistical Energy Analysis*
* *Interaction of Fluid Motion and Sound*
* *Passive and Active Damping*
* *Underwater Acoustics*
* *Boundary Element Methods*
* *Diagnostics & Condition Monitoring*
* *Material Characterization & Non-destructive Evaluation*
* *Active Noise and Vibration Control*
* *Sound Radiation & Scattering*
* *Finite Element Analysis*
* *Human Response to Sound and Vibration*

Abstracts of contributed papers proposed for presentation at the Congress should be sent as soon as possible and must be received by the Congress Secretariat no later than December 31, 1993. Abstracts should be approximately 200 words in length. If the abstract is accepted the paper must be typed on special manuscript sheets which will be supplied by the Congress Secretariat. The complete manuscript will be printed in the Congress Proceedings, and must be received no later than February 28, 1994. The Congress Registration fee is US $290 prior to December 31, 1993.

Papers must be presented in English. Books of Proceedings will be available at the Congress. There will be a parallel cultural program involving visits to the many attractions in Montréal and Quebec city. Montréal is an historic city founded 350 years ago, it has a European flavour and a unique and attractive atmosphere.

Copies of the first and second proceedings are available at the domestic postpaid price of US $95 and US $125 respectively. Please add US $30 for each copy for overseas airmail.

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THIRD INTERNATIONAL CONGRESS ON AIR- AND STRUCTURE-BORNE SOUND AND VIBRATION
Montréal, CANADA, June 13-15, 1994

Name______________________________ Position________________

Address________________________________________

____________________ City________________ Country________________

I am interested in submitting an abstract entitled____________________________

________________________________________________

Please return the form to Congress Secretariat, Dept. Mechanical Engineering, 202 Ross Hall, Auburn University, AL 38649, USA (FAX: 205 844 3307, Tel: 205 844 3310).

# The Canadian Acoustical Association
# l'Association Canadienne d'Acoustique

## 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 complété(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*

### PRIX ETUDIANT 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, Hearing Health Care Research Unit, University of Western Ontario, London, ON N6G 1H1. Les récipiendaires antérieur(e)s sont:

*1990 Bradley Frankland, Dalhousie University*
*1991 Steven Donald Turnbull, University of New Brunswick*
*Fangxin Chen, University of Alberta*
*Leonard E. Cornelisse, University of Western Ontario*

### PRIX ETUDIANT 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*

### PRIX ETUDIANT 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 chaqun, sont décernés à des auteurs professionnels âgés de moins de 30 ans et de 30 ans et plus, respectivement. Coordonnatrice: Chantai Laroche, Département d'orthophonie et d'audiologie, Université d'Ottawa, 545 King Edward, Ottawa, ON K1N 7N5.

### PRIX DE PRESENTATION ETUDIANT

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, Ontario Hydro, Health and Safety Division, 1549 Victoria Street East, Whitby, ON L1N 9E3.

# 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*

### 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, Hearing Health Care Research Unit, University of Western Ontario, London, ON N6G 1H1. Past recipients are:

*1990 Bradley Frankland, Dalhousie University*
*1991 Steven Donald Turnbull, University of New Brunswick*
*Fangxin Chen, University of Alberta*
*Leonard E. Cornelisse, University of Western Ontario*

### 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*

### 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: Chantal Laroche, Département d'orthophonie et d'audiologie, Université d'Ottawa, 545 King Edward, Ottawa, ON K1N 7N5.

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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, Ontario Hydro, Health and Safety Division, 1549 Victoria Street East, Whitby, ON L1N 9E3.

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**Submissions**: The original manuscript and two copies should be sent to the Editor-in-Chief.

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