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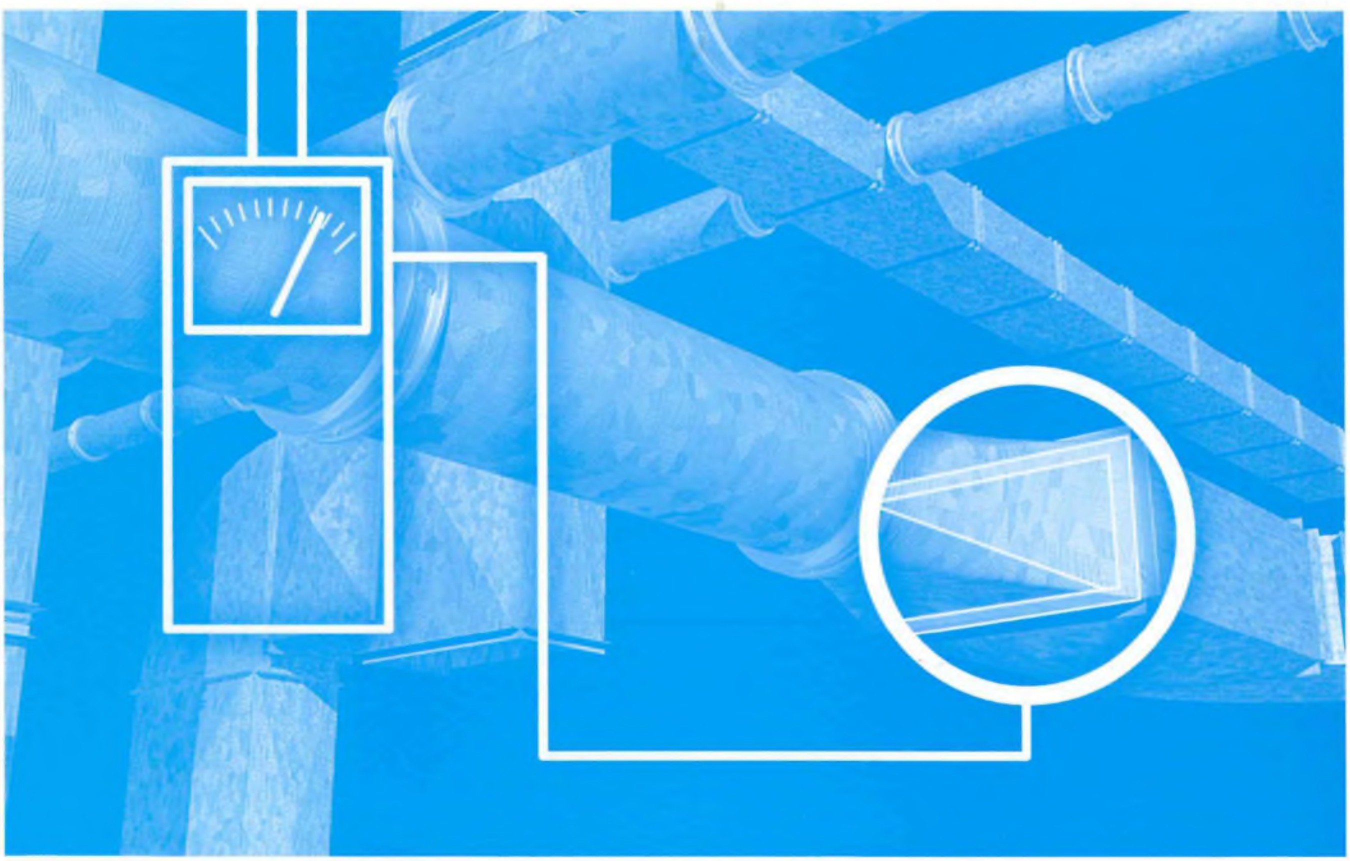

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EDITORIAL / ÉDITORIAL

We had a highly successful annual conference in Alliston, ON, in 2001. It was successful on a number of accounts. A complete account of the conference by Dalila Guisti, the conference chair, is included in this issue of the journal.

The membership agreed to an increase in membership fees so that the journal could be funded adequately in its endeavours. For instance, the membership didn't want to pose any restrictions in the size of each article submitted to the journal. I'd like to personally thank the membership for according this freedom to the editorial board of the journal.

This year's conference will be held in Charlottetown, PEI, and the local organizing committee is expected to make the conference as memorable as previous venues have been. The information about the conference, included in this issue, is periodically updated on our website: www.caa-aca.ca, due to the yeoman efforts of its maintainer, David Stredulinsky. Please visit the site regularly and pass on your comments to Dr. Stredulinsky. The registration form for the conference will be available some time in April on our website.

Finally, we have been talking about increasing our journal content from four to six issues per year. I am quite optimistic about such a venture. We are researching the financial ramifications of the proposal. In addition, it also needs your active participation such as submitting research articles, case studies and any other items of interest. I hope to submit a final report about the proposal in this year's AGM at Charlottetown, PEI. If all goes well, the year 2003 might be the start of this new venture. We can only hope!

Ramani Ramakrishnan

Nous avons eu une conférence annuelle très réussie en 2001 à Alliston, Ontario. Un compte rendu complet de la conférence fourni par Dalila Guisti, responsable de la conférence, est inclus dans cette édition du journal.

Les membres se sont mis d'accord sur l'augmentation des honoraires d'adhésion de sorte que le journal puisse être financé convenablement dans ses efforts. Par exemple, les membres n'ont pas voulu poser de restriction sur la taille des articles soumis au journal. Je voudrais remercier personnellement les membres d'avoir accorder cette liberté au bureau de rédaction du journal.

La conférence de cette année se tiendra à Charlottetown, PEI et on s'attend à ce que le comité d'organisation local rends cette conférence aussi mémorable que les rendez-vous précédents l'ont été. Les informations sur la conférence, incluse dans ce journal, sont périodiquement mises à jour sur notre site Internet: www.caa-aca.ca, grâce aux efforts de David Stredulinsky. Veuillez visiter le site régulièrement et transmettez vos commentaires à Dr. Stredulinsky. Le formulaire d'inscription à la conférence sera disponible au mois d'avril sur notre site internet.

En conclusion, nous avons invoqué la possibilité d'augmenter le nombre d'issue de notre journal de quatre à six par an. Je suis tout à fait optimiste au sujet d'une telle entreprise. Nous cherchons les ramifications financières de la proposition. De plus, on a également besoin de votre participation active telle que soumettre des articles de recherches, des études de cas et tout autres éléments d'intérêt. J'espère soumettre un rapport final au sujet de cette proposition à l'AGM de cette année à Charlottetown, PEI. Si tout va bien, l'année 2003 pourrait être le début de cette nouvelle entreprise. Nous pouvons seulement espérer!

Ramani Ramakrishnan

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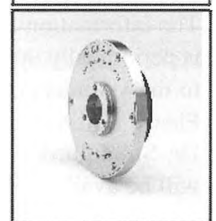

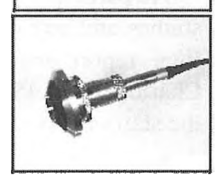

DUCT MODES CONVERSION DUE TO THE PRESENCE OF LINERS

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ABSTRACT

This paper deals with the classical problem of transmission-reflection and modes conversion due to the presence of acoustic liners in duct. The model consists of an infinite rectangular duct with an unlined and lined section. The approach is based on the modal calculation of the total acoustic sound power in a duct by the determination of the modal coefficients of the transmitted waves at the impedance discontinuity junction. For a given propagating mode (plane wave or higher order mode), incident from the rigid duct, the total sound power is calculated in the lined section by summing the acoustic power over all generated modes. The example studied here shows how the conversion affects the nature of the incident mode as a function of the admittance of the liner, frequency and mode number. It also allows for the quantification of the attenuation provided by an acoustic duct liner.

SOMMAIRE

Le recours a des revêtements de parois absorbants le long d'un conduit est un moyen naturel de réduire le bruit généré par des machines tournantes (turboréacteurs, circuits de ventilation, etc.). D'un point de vue scientifique, ce contexte pose plusieurs problèmes fondamentaux tant sur le plan de la compréhension des phénomènes que sur celui de leur modélisation. Il faut en effet pouvoir préciser les aspects liés à la propagation guidée en présence de parois absorbantes. Cet article s'inscrit dans ce contexte et discute les effets liés à la conversion des modes qui est causée par la présence d'une discontinuité d'impédance en conduit. Pour ce faire, un model de conduit infini tridimensionnel a été étudié ou une partie est traitée par un revêtement acoustique. L'approche est basée sur un calcul modal de la puissance acoustique en conduit après détermination des coefficients modaux des modes transmis. Pour un mode de propagation donné (ondes planes ou modes supérieures), dans la section rigide du conduit, la puissance acoustique totale est calculée dans la section traitée en sommant les puissances des modes générés par la discontinuité d'impédance. L'exemple étudié ici montre comment cette conversion affecte la nature du mode incident en fonction de l'impédance du revêtement acoustique, de la fréquence et du mode. Il permet aussi de calculer l'atténuation apportée par un revêtement acoustique en conduit.

1. INTRODUCTION

The presence of acoustic treatments inside a duct induces a discontinuity problem which has been subject for many studies and researches in the 70's. One of the first complete studies in the subject was undertaken by Zorumski [1] who solved the case of a lined duct with a known acoustic impedance. Later on, Lansing et al. [2] studied the effect of the impedance of the duct walls on the transmission-reflection coefficients and on the radiation from the end of a baffled duct. Koch [3] used the Wiener-Hopf technique to study the effect of a finite layer of an acoustic material in a two dimensional duct on the propagation of the acoustic modes. He found that the acoustic field had considerably changed because of the conversion of the modes due to the presence of the liner. The above studies mainly dealt with semi-infinite duct having a simple geometry. The difficulties arise when the geometry or the shape of the duct is no longer straight due to the analytical nature of these developments. The present study offers the advantage of considering both the plane waves and the higher modes that propagate in the duct. It also enables the calculation of the attenuation from the transmitted modal sound power of each generated mode at the impedance junction.

The model consists of an infinite rectangular duct with an unlined and lined section. This model was chosen, even though complicated due to the presence of the lined section, as it seems to provide more realistic results of the attenuation provided by an acoustic liner, compared to the case of fully lined duct. The presence of a lined section in duct induces a discontinuity problem that is solved here. When a propagating acoustic duct mode arrives at an impedance discontinuity, it is partly transmitted into the lined section as a series of modes and partly reflected back.

The modal coefficients of the transmitted modes are determined by solving a set of modal equations grouped in a matrix form. The total modal sound power is calculated after the determination of the modal coefficients of the transmitted modes at the impedance discontinuity junction. For a given propagating mode (plane wave or higher order mode), incident from the rigid duct, the total sound power is calculated in the lined section by summing the acoustic power over all generated modes. The example studied here shows how the conversion affects the nature of the incident mode in function of the admittance of the liner, frequency and the mode number. It also allows quantifying the attenuation provided by an acoustic duct liner.

2. THEORY

A modal theory of propagation and attenuation of sound in a three-dimensional rectangular duct has been fully described in ref. [4]. The two first sections of this paragraph give a summary of the sound propagation in unlined and lined duct with formulation to calculate the modal coefficients of the transmitted modes.

The duct system studied here is shown in Figure 1.

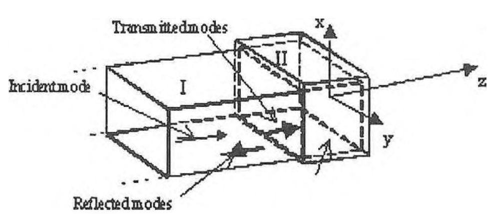

Figure 1: Three-dimensional infinite rectangular duct with an unlined and acoustically lined sections

The first section of the duct (section I) has rigid walls; the second section (section II) has all four walls treated with an acoustic liner with a known normalized acoustic admittance A. Let now consider an acoustic mode (m_0, n_0) incident from the rigid section I with a known amplitude $A_{m_0 n_0}$.

When the acoustic mode arrives at the admittance discontinuity junction, between the lined and unlined sections of the duct, it will be partly transmitted into the lined section as a series of modes with complex amplitudes $\hat{A}_{qp}$ and partly reflected back into the rigid section I with complex modal amplitudes $\hat{B}_{mn}$. The acoustic energy in the incident mode is thus partially transmitted into the lined section and partially reflected back. The modal coefficients of the transmitted waves are determined in section 2.2.

2.1 Propagation and attenuation in the duct

The acoustic field inside the duct is determined by solving the Helmholtz equation for the linear case and in the absence of sources and flow:

$$\Delta P + k^2 P = 0 \qquad (1)$$

where P is the acoustic pressure, $k = \omega / c_0$ is the wave number, ω the angular frequency and ρ, c_0 are the ambient density and speed of sound respectively. The sidewall boundary conditions are:

$$\frac{\partial P}{\partial \vec{n}} = \begin{cases} 0 & ; in\ section\ I \\ i k A P & ; in\ section\ II \end{cases} \qquad (2)$$

at $x = \pm l_x / 2$ and $y = \pm l_y / 2$

where A is the normalized wall admittance of a "locally reacting" boundary and $\vec{n}$ is the outward normal.

The general solution for the pressure field in sections I and II of the duct is given by,

$$P_I(x,y,z) = A_{m_0 n_0} \Psi(K_{m_0} x) \Psi(K_{n_0} y) e^{-i K_{m_0 n_0} z} + \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} \hat{B}_{mn} \Psi(K_m x) \Psi(K_n y) e^{i K_{mn} z} \qquad (3)$$

where,

$$\begin{cases} \Psi(K_m x) = {}^{\cos}_{\sin}(K_m x) \\ \Psi(K_n y) = {}^{\cos}_{\sin}(K_n y) \end{cases} \qquad (4)$$

are the eigenfunctions. The term $e^{i\omega t}$ is implicit throughout the paper. A cosine function is used for the even modes and a sine function for the odd modes. The transverse wave numbers $K_m = (m-1)\pi / l_x$ and $K_n = (n-1)\pi / l_y$ are determined by the boundary conditions (2) and where l_x and l_y are the cross-sectional dimensions of the duct in the x and y direction respectively, and m, n are integers different from zero.

The axial wave number is given by the following dispersion equation $K_{mn}^2 = k^2 - (K_m^2 + K_n^2)$

The propagation of the waves in the axial direction is possible as long as the axial wave number $K_{mn}^2 > 0$. According to the dispersion equation, this is true for $\omega > c_0\sqrt{K_m^2 + K_n^2} = \omega_{mn}^c$. Below this "cut-off" frequency ω_{mn}^c, the axial wave number K_{mn} becomes a purely imaginary number, and the propagation factors in equation (4) turn into $e^{-|K_{mn}z|}$; which means the amplitudes of these modes decay exponentially with axial distance from the source: they are "cut-off". Note that the mode (m_0, n_0) is just one particular mode over all possible (m,n) modes.

The general solution, for the pressure field in Section II, of Helmohltz equation (1) with boundary conditions (2) is (assuming that reflections from the end of the duct are neglected)

$$P_{II}(x,y,z) = \sum_{q=1}^{\infty}\sum_{p=1}^{\infty} \hat{A}_{qp}\ \hat{\Psi}(\hat{K}_q x)\,\hat{\Psi}(\hat{K}_p y)\, e^{-i\hat{K}_{qp}z} \tag{5}$$

where,

$$\begin{cases} \hat{\Psi}(\hat{K}_q x) = {}^{\cos}_{\sin}(\hat{K}_q x) \\ \hat{\Psi}(\hat{K}_p y) = {}^{\cos}_{\sin}(\hat{K}_p y) \end{cases} \tag{6}$$

are the complexes eigenfunctions. The transverse wave numbers $\hat{K}_q = \hat{\mu}_q \pi / l_x$ and $\hat{K}_p = \hat{\mu}_p \pi / l_y$ are determined by the boundary condition (2). The axial wave number is given by the dispersion equation: $\hat{K}_{qp}^2 = k^2 - (\hat{K}_q^2 + \hat{K}_p^2)$, where $\hat{\mu}_q$ and $\hat{\mu}_p$ are complex numbers. Note, in this case, the "cut-off notion" has no physical meaning. Assuming $\hat{K}_{qp} = (\alpha \pm i\beta)k$, α is the non-dimensional axial wave number and β is the damping factor of the mode. β should be positive for a mode propagating in $z>0$ direction. This means that we should look for a solution of the dispersion equation that provides attenuation.

Solving equation (5) by the method of separation of variables and imposing the boundary conditions (2) leads to the following characteristic equations

$$(\hat{K}_e l_k/2)\ {}^{\tan}_{\cot}(\hat{K}_e l_k/2) = \pm i k A l_k/2 \tag{7}$$

where the term in tangent is used for even modes and the one with cotangent for odd modes. The e and k indices represent q or p and x or y respectively depending on the propagation direction.

The axial and transverse wave numbers were computed using a numerical scheme developed in ref. [2, 3], where the characteristic equation is transformed into a first order non-linear differential equation. The differential equation is integrated using a Runge-Kutta algorithm with appropriate initial values. The transverse wave numbers then are used to compute the axial wave number using the dispersion equation.

2.2 Calculation of the Transmitted Modal Coefficients

The acoustic pressure P and the acoustic velocity V are related by the momentum equation

$$\nabla\overline{P} = -ik\rho c_0 \overline{V} \tag{8}$$

Using equations (3) and (5), the axial velocity in both sections (I and II) can be written as

$$\begin{cases} V_I(x,y,z) = (1/k\rho c_0)\left\{A_{m_0n_0} K_{m_0n_0} e^{-iK_{m_0n_0}z}\,\Psi(K_{m_0}x)\,\Psi(K_{n_0}y) - \sum_{m=1}^{\infty}\sum_{n=1}^{\infty} \hat{B}_{mn} K_{mn} e^{iK_{mn}z}\,\Psi(K_m x)\Psi(K_n y)\right\} \\ V_{II}(x,y,z) = (1/k\rho c_0)\sum_{q=1}^{\infty}\sum_{p=1}^{\infty} \hat{A}_{qp}\hat{K}_{qp}\,\hat{\Psi}(\hat{K}_q x)\,\hat{\Psi}(\hat{K}_p y)\, e^{-i\hat{K}_{qp}z} \end{cases} \tag{9}$$

The unknown amplitudes $\hat{A}_{qp}$ and $\hat{B}_{mn}$ in equations (3), (5) and (9) are determined from a system of linear equations obtained by applying the continuities conditions: the pressures and axial velocities in the two sections of the duct must be equal at the junction $z = 0$ $P_I(x,y,0) = P_{II}(x,y,0)$ and $V_I(x,y,0) = V_{II}(x,y,0)$

Thus, by substituting equations (3), (5) and (9) into these two equations, and using the orthogonality properties of the eigenfunctions, the following system for the transmitted modal coefficients is obtained

$$\sum_{q=1}^{\infty}\sum_{p=1}^{\infty} \hat{A}_{qp}\,\hat{I}_{m'q}\,\hat{J}_{n'p}\,(\hat{K}_{qp} + K_{m'n'}) = A_{m_0n_0}\frac{(1+\delta_{m_01})}{2}\frac{(1+\delta_{n_01})}{2}\delta_{m_0m'}\,\delta_{n_0n'}\,(K_{m'n'} + K_{m_0n_0}) \tag{10}$$

where δ is the Kronecker delta and

$$\hat{I}_{ij} = \frac{1}{l_x} \int_{-l_x/2}^{l_x/2} \Psi(K_i\, x)\, \hat{\Psi}(\hat{K}_j\, x) dx \qquad (11)$$

$$\hat{J}_{ij} = \frac{1}{l_y} \int_{-l_y/2}^{l_y/2} \Psi(K_i\, y)\, \hat{\Psi}(\hat{K}_j\, y) dy \qquad (12)$$

Equations (11) and (12) are solved analytically. The system indices, m and n, vary from 1 to **M** and 1 to **N** respectively, while q, p vary from 1 to **Q** and 1 to **P** respectively after truncation. Therefore, we have $[\mathbf{Q}\,.\,\mathbf{P}]$ complex equations and $[\mathbf{Q}\,.\,\mathbf{P}]$ complex unknown which are the transmitted modal coefficients. **M** and **N** are the total propagating modes in a duct following each transverse direction. The final linear system (10) takes the final matrix form $[\mathbf{a}].[\mathbf{X}]=[\mathbf{b}]$

where,

- $[\mathbf{a}]$ complex vector which contains the modal transmitted coefficients to be determined,
- $[\mathbf{X}]$ complex matrix which depends on the modes (m,n) and on the eigenvalues of the system,
- $[\mathbf{b}]$ known vector which depends on the incident mode (m_0,n_0) and its amplitude $A_{m_0 n_0}$.

The final matrix $[\mathbf{X}]$ is square and the dimension of the system is multiplied by 2 to account for the complex numbers, therefore the final matrix dimensions are $[2.\mathbf{Q}.\mathbf{P}\,,\,2.\mathbf{Q}.\mathbf{P}]$. Further, the truncation is performed at $\mathbf{Q}=\mathbf{M}+2$ and $\mathbf{P}=\mathbf{N}+2$. This truncation was checked when calculating all possible transmitted and reflected coefficients at the discontinuity junction for any incident mode (m_0,n_0). The determination of the modal coefficients and sound powers of the transmitted modes will show that it's worthless and time consuming to consider a number of modes (generated in section II) greater than the limit chosen above. An LU decomposition algorithm with matrix inversion was used to solve the matrix system.

2.3 Sound power calculation

The modal axial acoustic velocity of a propagating mode, following $z>0$ direction, is given by:

$$\mathrm{V_{mn}} = \frac{1}{\rho\, c_0}\left(\frac{K_{mn}}{k}\right) P_{mn} \qquad (13)$$

and the modal acoustic intensity is given by,

$$I_{mn} = \overline{P_{mn}\, V_{mn}} = \frac{1}{2}\mathrm{Re}\left\{P_{mn}\, V_{mn}^*\right\} \qquad (14)$$

where V_{mn}^* denotes the complex conjugate of V_{mn}.

The sound power is obtained by integration of the intensity over the duct section (ref. [5, 6]):

$$\Pi = \iint_S I\, dS \qquad (15)$$

where S is the duct cross section area.

Rigid duct case

For an incident mode with a given index (m,n), P_{mn} is given by the solution of the wave equation. The modal acoustic sound power in a rigid duct is:

$$\Pi_{mn} = \frac{1}{2}\left(\frac{S}{\varepsilon_m \varepsilon_n}\right)\left(\frac{1}{\rho c_0}\right)\left|A_{mn}\right|^2\left(\frac{K_{mn}}{k}\right) \qquad (16)$$

with K_{mn} real, S the cross section area of the duct, and $\varepsilon_i = \begin{cases} 1 & ; if\ i=1 \\ 2 & ; if\ i>1 \end{cases}$

Lined duct case

The modal acoustic sound power is obtained in similar way as for the rigid duct case. The modal sound power, for a given generated mode in section II, is given by:

$$\Pi_{qp}^{I} = \frac{1}{2}\left(\frac{1}{\rho\, c_0}\right)\left|\hat{A}_{qp}\right|^2 I_q\, I_p\, \mathrm{Re}\left\{\frac{\hat{K}_{qp}}{k}\right\} e^{-2\mathrm{Im}(\hat{K}_{qp})z} \qquad (17)$$

with

$$I_q = \int_{-l_x/2}^{l_x/2} {\scriptstyle\cos \atop \scriptstyle\sin}\left(\hat{K}_q\, x\right) {\scriptstyle\cos \atop \scriptstyle\sin}\left(\hat{K}_q^*\, x\right) dx \qquad (18)$$

and

$$I_p = \int_{-l_y/2}^{l_y/2} {\scriptstyle\cos \atop \scriptstyle\sin}\left(\hat{K}_p\, y\right) {\scriptstyle\cos \atop \scriptstyle\sin}\left(\hat{K}_p^*\, y\right) dy \qquad (19)$$

The integrals are calculated analytically by using the trigonometry functions' properties as follows:

$$\int_{-l_x/2}^{l_x/2} {\scriptstyle\cos \atop \scriptstyle\sin}\left(\hat{K}\, x\right) {\scriptstyle\cos \atop \scriptstyle\sin}\left(\hat{K}^*\, x\right) dx = \frac{1}{2}\left\{\frac{sh\left[\mathrm{Im}(\hat{K}l)\right]}{\mathrm{Im}(\hat{K}l)} \pm \frac{\sin\left[\mathrm{Re}(\hat{K}l)\right]}{\mathrm{Re}(\hat{K}l)}\right\} \qquad (20)$$

The total transmitted sound power in region II is the sum of the modal acoustic power of each generated mode, and is given by:

$$\Pi_t^{II} = \sum_{q=1}^{\infty}\sum_{p=1}^{\infty} \Pi_{qp}^{II} \qquad (21)$$

Finally, the expression of the total attenuation provided by the acoustic treatment over a certain length L is given by:

$$\Delta\Pi_t^{II}(dB) = 10\log_{10}\left(\sum_{q=1}^{\mathbf{Q}}\sum_{p=1}^{\mathbf{P}} \Pi_{qp}^{II}(L) \Big/ \sum_{q=1}^{\mathbf{Q}}\sum_{p=1}^{\mathbf{P}} \Pi_{qp}^{II}(0)\right) \qquad (22)$$

3. APPLICATION

An example of application, which deals with noise suppression from a turboshaft engine inlet, is discussed in this section. The aim is to attenuate the high frequency noise, generated by the compressor (fundamental mode of blade passing frequency), by lining the inlet with a composite material. The frequency of concerns is situated between 8 and 10 kHz. The liner consists of a solid backplate and a layer of feltmetal material separated by honeycomb structures. The normalized acoustic admittance of the used liner is referred to R1. It has been determined experimentally by using an impedance tube measurement.

Results from a two-dimensional case is shown in Figure 3 and Figure 4 where the total transmitted non-dimensional sound power $\left(\Pi_t^{II}/\Pi_1\right)$ for the least attenuated incident mode (fundamental mode) is plotted versus the frequency with a 500 Hz step for a weak and strong constant normalized admittance values $A=(.1,+.1)$ and $A=(1.,+.1)$ respectively, and at different location $z=0$ m (at the discontinuity junction), $z=0.1$ m and $z=0.2$ m. The duct width is 0.2 m, $\mathbf{M}=12$ at $f_{\max}=10\,kHz$, and the system truncated at $\mathbf{Q}=20$.

For a weak admittance value, the graph in Figure 3 at $z=0.1$ m, shows that the transmission is complete with almost no reflections. However for the same mode, Figure 4 shows a much stronger reflection at the discontinuity junction. The graphs of this figure exhibit a wavy behavior, especially at $z=0$ m, which due to the successive contribution of the reflected modes occurring at the discontinuity junction while the frequency increases.
The results from an incident mode 3 are shown in Figure 4 for a strong constant normalized admittance value.
It is clear that the sound power ratio increases with the frequency at a constant admittance value. This representation also allows to globally quantifying the attenuation provided by the liner.

A three-dimensional square duct of 0.1 m by 0.1 m cross section was considered in the calculation of the total transmitted power $\left(\Pi_t^{II}/\Pi_{m0n0}\right)$. Figure 5 and Figure 6 show the calculation results for modes (1,1) and (2,2) for different admittances (constant weak and strong admittance value and a variable admittance R1). The same behaviors as observed in the two-dimensional case described above occur here.

Finally, the graphs of Figure 7 and Figure 8 show the total modal attenuation in dB, provided by the liner, versus the lined length for different incident modes and at $Kl=29.3$. The model duct had 0.1 m by 0.1 m cross section, and the matrix system has been truncated at $\mathbf{Q}=\mathbf{P}=20$ for the calculation of the transmitted coefficients, that means, almost a double number of modes in the lined section (section II) than the propagating modes in section I was considered at a maximum frequency of 10 kHz where $\mathbf{M}=\mathbf{N}=12$.
It can be seen that liner with normalized admittance R1 provides better attenuation in the frequency range of concerns.

4. CONCLUSION

The objective is to provide formulations that allow the quantification of the attenuation provided by a liner that has a known acoustic admittance. The modal analytical approach described here permits an understanding of the modes conversion phenomena. The total transmitted sound power has been calculated for different wall admittance values representing a weak, strong and optimal "R1" attenuation at different locations.

5. REFERENCES

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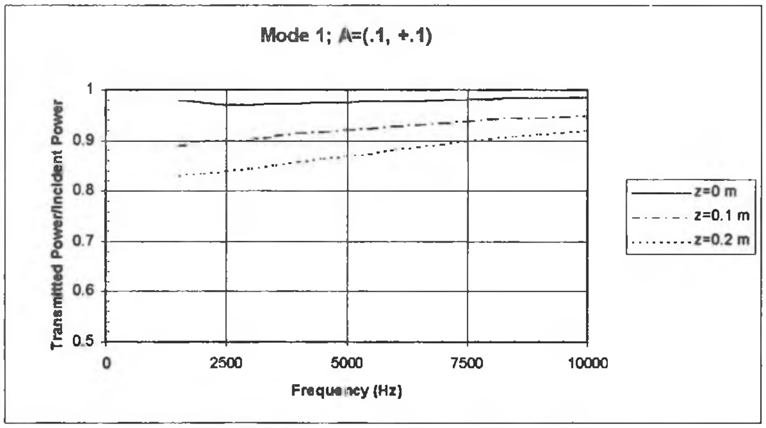


Figure 2: Total transmitted to incident power ratio $\left(\Pi_t^{II}/\Pi_{m0}\right)$ *as function of frequency for incident mode 1 and a normalized acoustic admittance* $A=(.1,+.1)$

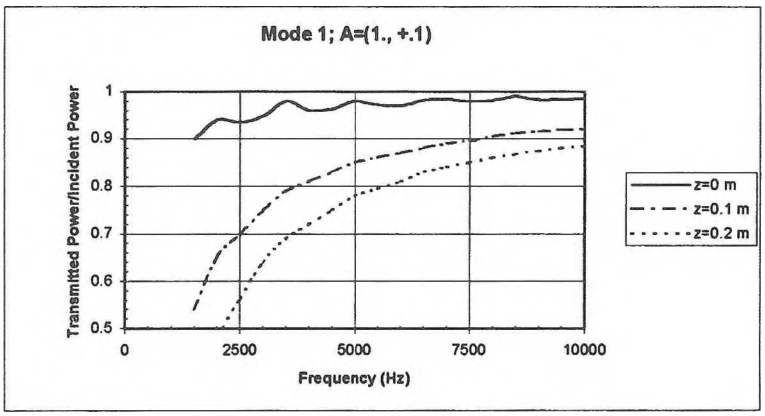


Figure 3: Total transmitted to incident power ratio $\left(\Pi_t^{II}/\Pi_{m0}\right)$ *as function of frequency for incident mode 1 and a normalized acoustic admittance* $A=(1.,+.1)$

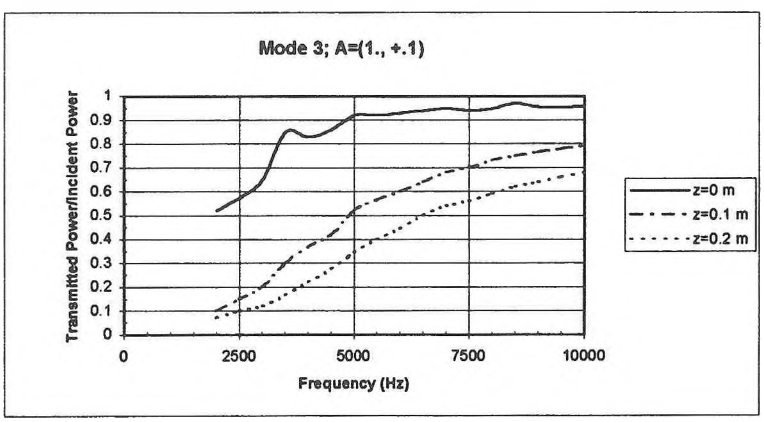


Figure 4: Total transmitted to incident power ratio $\left(\Pi_t^{II}/\Pi_{m0}\right)$ *as function of frequency for incident mode 3 and a normalized acoustic admittance* $A=(1.,+.1)$

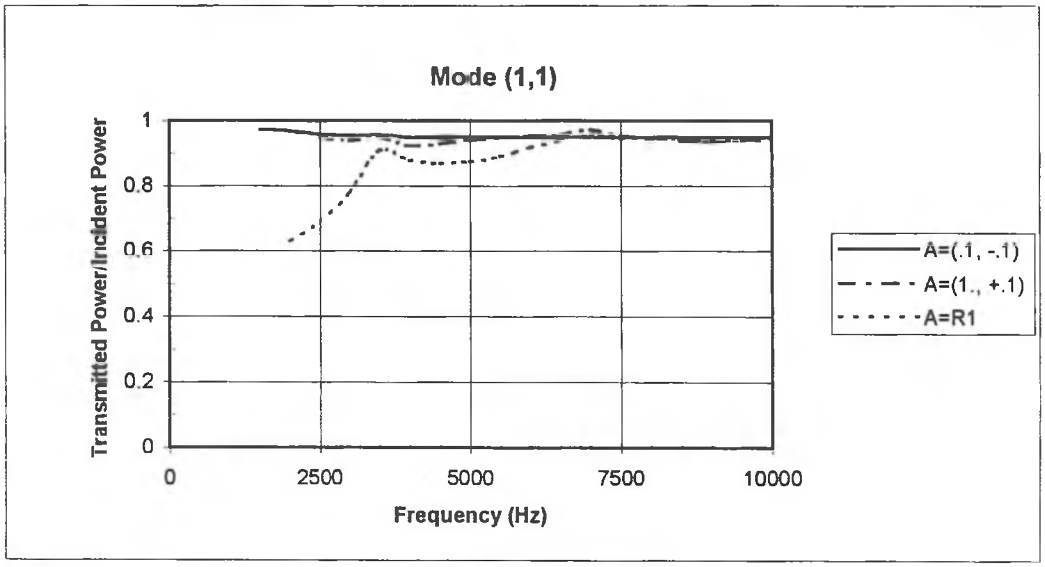


Figure 5: Total transmitted to incident power ratio $\left(\Pi_t^{II}/\Pi_{m0n0}\right)$ as function of frequency for incident mode (1, 1) and different normalized acoustic admittance values.

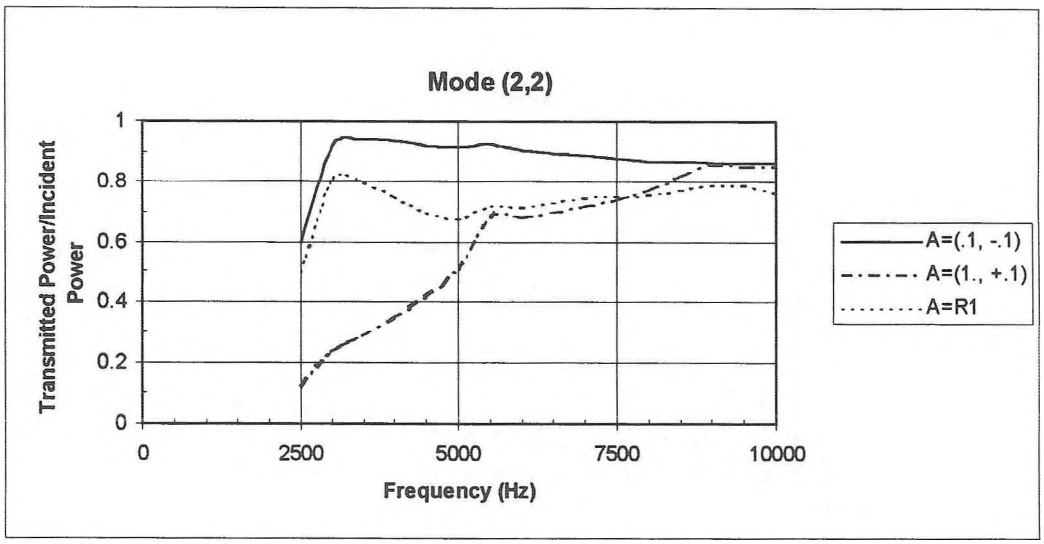


Figure 6: Total transmitted to incident power ratio $\left(\Pi_t^{II}/\Pi_{m0n0}\right)$ as function of frequency for incident mode (2, 2) and different normalized acoustic admittance values.

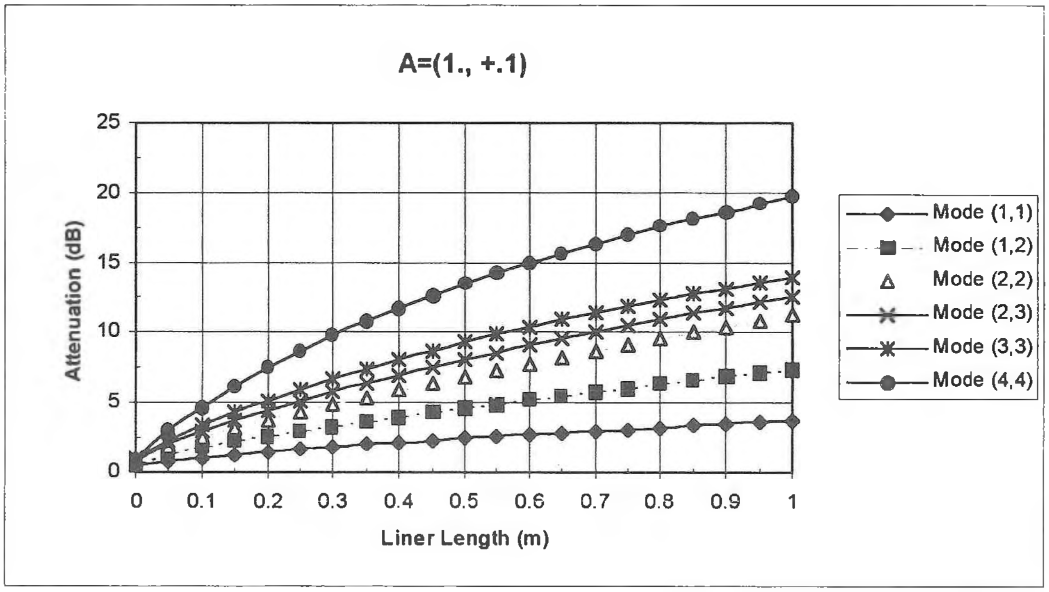


Figure 7: Modal attenuation $\left(\Pi_t^{II}/\Pi_i\right)$ as function of liner length for different incidents modes and a normalized acoustic admittance $A=(1.,+.1)$

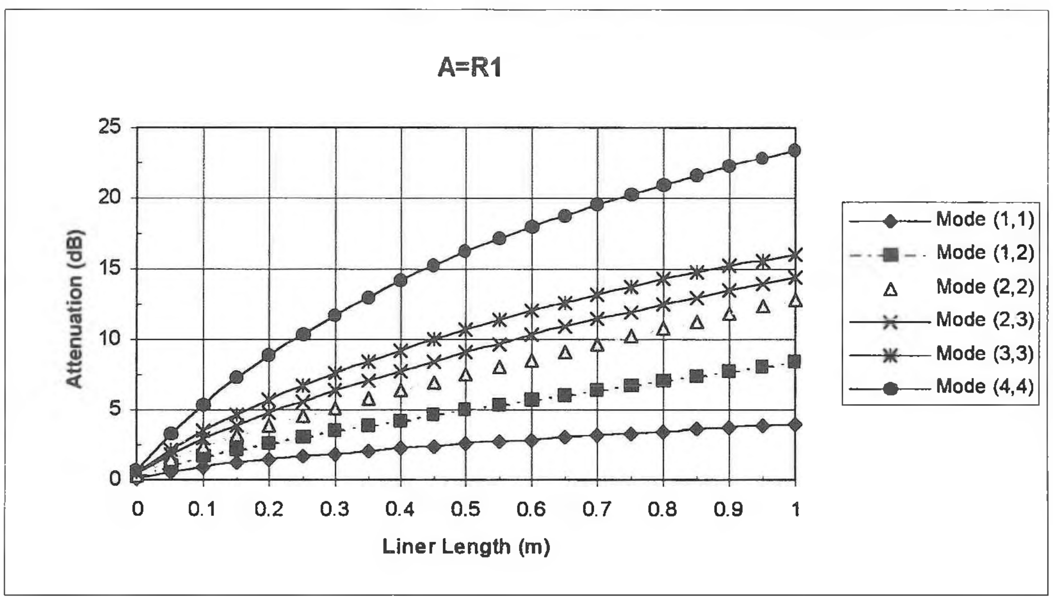
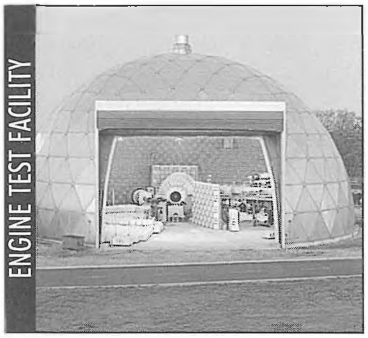


Figure 8: Modal attenuation $\left(\Pi_t^{II}/\Pi_i\right)$ in function of the distance for different incidents modes and a normalized acoustic admittance $A=R1$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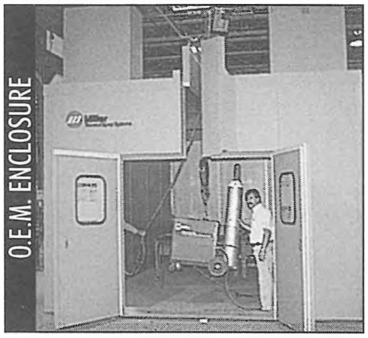
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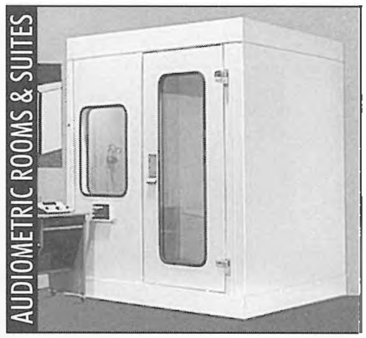
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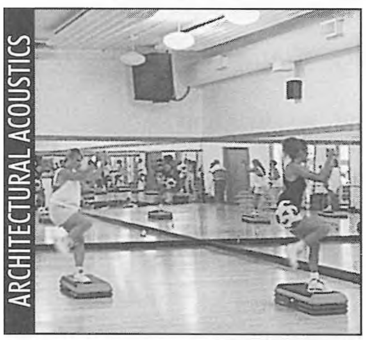
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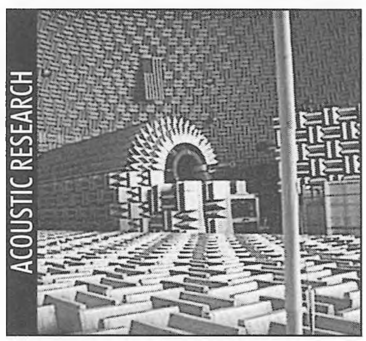
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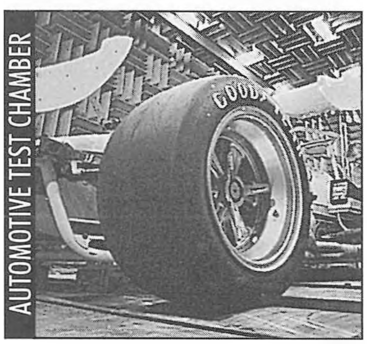
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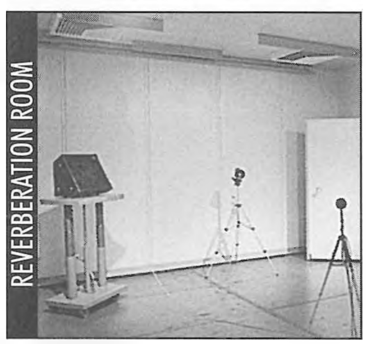
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LOW FREQUENCY ACOUSTIC TEST CELL FOR THE EVALUATION OF CIRCUMAURAL HEADSETS AND HEARING PROTECTION

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ABSTRACT

Active noise reduction (ANR) technology, based on feedback signal processing, is being applied in commercial communication headsets and provides noise reductions up to 10-20 dB between 50 Hz and 400 Hz. There are, however, many acoustical designs and computational difficulties associated with feedback designs which limit their performance. Current research in feedforward design offers the opportunity for significant improvement in ANR performance. To support this current research in ANR feedforward algorithm development and evaluation, a low frequency acoustic test cell (LFATC) has been designed to provide a uniform and precisely controlled low frequency acoustic measurement environment. The LFATC design is based on the original work of E.A.G. Shaw and G.J. Theisson at the National Research Council of Canada [1] and a prototype LFATC developed by J.G. Ryan, E.A.G. Shaw, A. J. Brammer, and T.G. Zang [2]. The design analysis of the LFATC is based both on a lumped parameter model and a one-dimensional standing wave model. The acoustic performance of this test cell, including a simple floor vibration isolation system, is evaluated experimentally over a wide range of sound pressure levels. A representative set of measurements with a prototype ANR headset illustrates the application of the LFATC.

SOMMAIRE

La technique de réduction active de bruit, basée sur le traitement de signal rétroactif, est appliquée aux écouteurs de communication commerciaux et rend possible des réductions de bruit jusqu'à 10-20 dB entre 50 Hz et 400 Hz. Il ya cependant bien des obstacles de performance acoustique et de calculs à surmonter avec les systèmes rétroactifs. La recherche actuelle sur les modèles "feedforward" promet une augmentation significative de la performance des systèmes de réduction active de bruit. Pour aider cette recherche vers le développement et l'évaluation d'un algorithme "feedforward", une cellule de test acoustique à basse fréquence (LFATC) a été construite pour fournir un environnement de mesure acoustique uniforme et contrôllé avec précision. L'idée de cette cellule est basée sur le travail original de E.A.G. Shaw et G.J. Theisson au Conseil National de Recherches Canada [1] et sur un prototype de LFATC créé par J.G. Ryan, E.A.G. Shaw, A.J. Brammer et T.G. Zang [2]. L'analyse de fonctionnement de notre cellule est basée sur un modèle de paramètres groupés et un modèle d'onde unidimensionnelle stationnaire. La performance acoustique de cette cellule, comprenant un sytème simple de plancher pour isolation vibrationnelle, est evaluée expérimentalement à travers un intervalle large de niveaux de pression sonore. Un groupe représentatif des mesures avec une paire prototype d'écouteurs à réduction active de bruit illustre l'application de notre cellule.

1. INTRODUCTION

The exposure of the human auditory system to high levels of low frequency noise constitutes a serious problem in modern society. The objectives of this paper are to: (1) describe a specially designed low cost, compact laboratory measurement facility for the evaluation of low frequency Active Noise Reduction (ANR) hearing protection systems; and (2) demonstrate the accuracy of the acoustical measurement system for a range of experimental conditions. Passive noise reduction afforded by communication headsets and hearing protectors is a function of the frequency of the environmental noise and is limited in the low frequency range. Figure 1 shows representative passive noise attenuation measurements for hearing protectors compiled from flat plate testing and low frequency acoustic test cell (LFATC) measurements. Passive noise attenuation typically reaches 30 dB above 1kHz but decreases to less than 10 dB below 100Hz. These relatively small reductions are inadequate for hearing protection in high noise level environments dominated by low frequency noise. The adverse effects of high intensity low frequency sound, including the effect of the upward spread in masking at higher frequencies, are summarized in reference [3].

Active Noise Reduction (ANR), based on feedback designs, has been developed and implemented in commercially available headsets over the past decade. Figure 2 illustrates representative objective measurements reported for current technology ANR feedback headsets compiled from LFATC testing and MIRE testing. Included in Figure 2 are upper and lower noise reduction performance bands for feedback headsets from [4]. Reductions of the order of 10-20 dB over a frequency range of 50-400 Hz are measured for stationary broadband white noise introduced through multiple speaker systems in reverberant rooms. It is well known that the performance of feedback ANR headsets is highly dependent on the specific characteristics of the incident noise and that the feedback circuitry and signal processing often adds to the noise in the mid-frequency speech communication bands [5]. Due to the limitations of feedback ANR headsets, recent research has focused on development of feedforward ANR based on least-mean-squared (LMS) adaptive filters [6-11]. The schematic of a feedforward ANR system is shown in Figure 3, and Figure 2 provides sample active noise reduction measurements of feedforward ANR in response to pure tones, from [6].

The variable spatial and temporal acoustic environments of reverberant testing room facilities are inadequate to support the precision measurements required for research and development investigations in advanced feedforward ANR algorithms and prototype headset designs. The need is to establish a highly controlled noise field in which the performance and stability of specific ANR algorithms, in conjunction with actual headset hardware configurations, can be accurately measured and compared. The source noise fields must include a wide range of conditions from deterministic

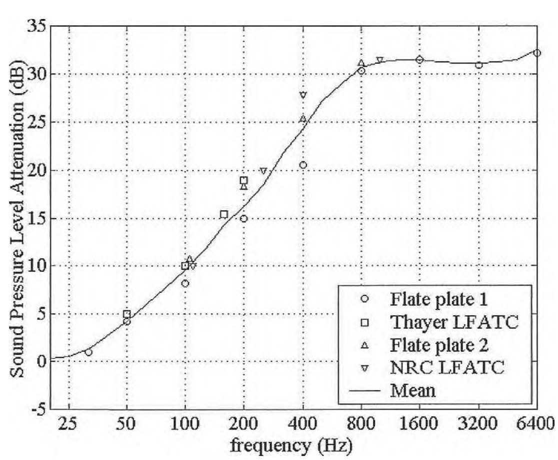

Figure 1. Representative passive noise attenuation in flat plate testing and LFATC testing of hearing protectors.

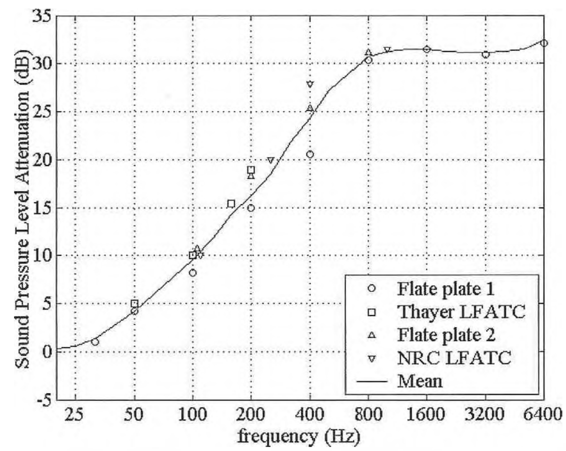

Figure 2. Representative active noise reduction in MIRE and LFATC testing of feedback and feedforward headsets.

discrete frequency tones to broadband non-stationary and narrowband impulsive sources.

The development of a low frequency acoustic test cell (LFATC) has been an important component in the feedforward ANR research of Dr. Anthony Brammer and his colleagues at The National Research Council (NRC), Ottawa, Canada. Their prototype LFATC design was based on the original work of Shaw and Theisson [1] and is described in reference [2]. Results of the research are reported in references [6-9].

In 1998, the authors initiated research in feedforward LMS algorithms for low frequency noise reduction in communication headsets as part of a Unites States Air Force sponsored program. A major goal of the Thayer research has been to develop and evaluate the performance and stability of innovative adaptive leaky LMS algorithms based on Lyapunov tuning principles [10]. As was the case for the NRC program, this research requires precise acoustic measurements under rigorously controlled experimental conditions involving high intensity noise levels up to 140 dB, a wide dynamic range, and real world, highly non-stationary jet aircraft noise. The design and construction of the Thayer LFATC are presented in this paper together with examples of the measurements obtained with this facility. A comprehensive description of the experimental results of this research program is given in reference [11].

The modified LFATC specifically addresses the need for vibration isolation, providing a sufficiently low noise floor to demonstrate feedforward ANR over a wide dynamic range. Moreover, the design extends the cut off frequency of the LFATC up to 200 Hz, as compared to 100 Hz in the NRC prototype LFATC reported in [2]. Additional design and fabrication refinements provide a functional laboratory measurement system for testing communication headsets.

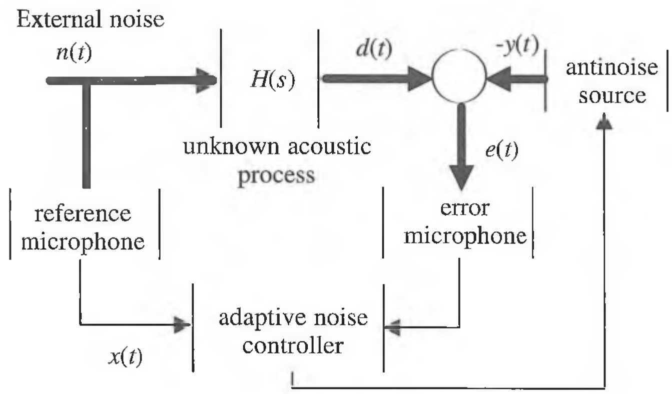

Figure 3. Single channel feedforward active noise control. Thick lines represent acoustic signals, thin lines represent electrical signals.

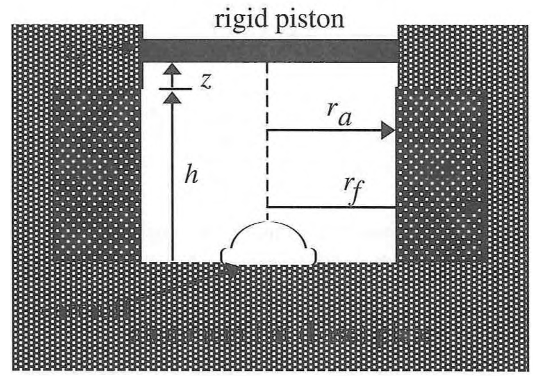

Figure 4. Low frequency acoustic test cell topology showing critical dimensions and placement of earmuff.

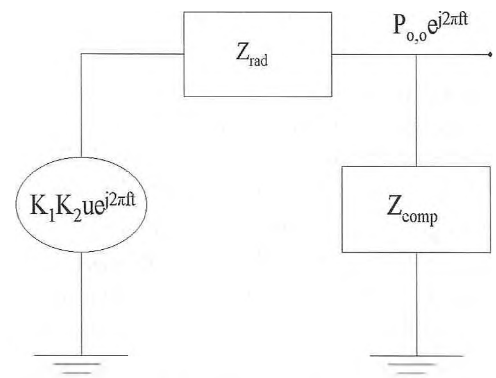

Figure 5. Lumped parameter model.

2. LFATC DESIGN

The specifications for the Thayer LFATC are as follows:

Frequency range:	20-200 Hz; uniform response
Dynamic range:	65 dB; Source levels, 75-140 dB
Noise floor:	50 dB in headset
Wall thickness:	Min. 50 dB transmission loss
Signal-to-noise ratio for experiments:	30 -50 dB in headset
Loudspeaker:	High fidelity with minimum distortion at up to 140 dB

These specifications are predicated on three principle performance requirements: (1) precise, direct sound pressure measurements over the frequency band for maximizing ANR performance without filters or equalizers; (2) evaluation of ANR algorithms over a 65 dB range of noise levels within the headset; and (3) operation in normal laboratory quiet environments.

Two models were employed in the LFATC design, namely a lumped parameter model and a one-dimensional standing wave model. Use of the two models is dictated by the predicted dimensions, i.e., lumped parameter, one-dimensional and multi-dimensional models are based on the ratio of wavelength, , to a major dimension [12]. The measured broadband response is represented by both the lumped parameter and the standing wave model. First, a given topology is assumed which establishes the physical relationships of the proposed system as shown in Figure 4.

The cylindrical test cell wall is made of aluminum (6061-TS), which provides both the required stiffness for the rigid wall assumption and the mass for the required high transmission loss. The inner wall is lined with a sound absorbing felt wool material to minimize higher frequency cross modes. This lining, which is made of 95% natural wool and has a density of 342 kg/m^3 and a thickness of 3.81 cm, is not included in the models. The acoustic medium is air at T= 30 C and humidity of 70%.

The critical dimensions for the topology are shown in Figure 4. The test cell is a right cylinder with critical dimensions of height of test volume, h, radius of air column, r_a, and radius of felt annulus, r_f. The test cell's excitation is provided by a speaker which is modeled as a rigid piston of radius r_a, at a height z. z_o is the mean position of the piston's radiating surface. The objective of this model is to determine the pressure, $P_{0,0}\, e^{2\pi ft}$, at the center of the base plate surface contacting the air column, $(h,\ r_a)$ = (0 cm, 0 cm), on which a prototype circumaural headset is to be mounted.

Figure 5 gives the equivalent lumped parameter model for the LFATC, where f is the frequency, $Ue^{2\pi ft}$ is the volume velocity (m/s), and K_1 is the constant speaker gain (N-sec/volt-m^2). Z_{rad} (N-s/m^5) is the radiation impedance into the LFATC and Z_{comp} is the compliance impedance associated with the test volume. The following equations are given in Beranek [12] for $kr_a < 0.5$, where k is the wave number:

$$Z_{rad} = 6\frac{\rho r_a^2 \pi^4 f^4}{c^3} \quad (1)$$

$$Z_{comp} = \frac{-j\pi(r_f^2 h + r_a^2 z_0)}{\rho c^2 2\pi f} \quad (2)$$

The estimated values for density, ρ , and speed of sound in air, c, provide a closer forecast of subsequent LFATC experimental calibration. Conversely, use of the LFATC is not limited to these experimental conditions.

Given the impedance expressions in eq. 1 and 2, the transfer function, $H(f)$, of the lumped parameter model is

$$H(f) = \frac{K_1 Z_{comp}}{Z_{comp} + Z_{rad}} \quad (3)$$

For standing wave pressures, the internal volume of the test cell is modeled as a one-dimensional wave-guide. This is an acceptable approximation as long as the shell diameter, $2r_f$, is less than one-tenth the wavelength, 0.1λ. The problem is to find a single ray's interaction with the test cell's termination flange and speaker diaphragm as it reflects between the source and the base flange, and then to integrate to capture all excitation generated by the source at steady state. The point source, P_s is modeled as

$$P_s(f) = K_2 K_1 U e^{j2\pi ft} \quad (4)$$

where K_2 is a unitless correction factor, which is necessitated by the fact that ambient values of the lumped-parameter and standing wave models must agree. Air has a linear attenuation factor, δ, that describes the attenuation of sound per unit length of a ray's travel. At each surface reflection, the surface pressure is the sum of the incident wave and the reflected wave. The reflected wave is reduced by the reflection coefficient, α, for the flange and β, for the diaphragm. The calculation of reflection coefficient for aluminum is

$$\alpha = \frac{Z_{aluminum} - Z_{air}}{Z_{aluminum} + Z_{air}} \quad (5)$$

As the ray passes between surfaces it is also phase shifted with respect to the source due to the distance traveled by a factor, $-kx$. x the distance between the diaphragm and terminal flange. The resultant infinite series is

$$P_f(t,f,x) = K_1 K_2 U \sum_{i=1}^{\infty} \delta^{2i-1}(\alpha^{i-1} + \alpha^i)\beta^{i-1} e^{j(2\pi ft - (2i-1)kx)} \quad (6)$$

whose magnitude for this instance can be simplified as an infinite geometric series to

$$P_f(f) = K_1 K_2 U \frac{(1+\alpha)\delta e^{-j2\pi f(h+z_o)/c}}{1 - \delta^2 \alpha\beta e^{-j4\pi f(h+z_o)/c}} \quad (7)$$

The net magnitude of the sound pressure level for the LFATC is expected to be some weighted combination of the two models. For example,

$$P = 20\log_{10}\left[\frac{\left(w_1|H(f)| + w_2|P_f(f)|\right)}{20\mu Pa}\right] \text{dB} \quad (8)$$

where w_1 and w_2 are weighting factors. This model is valid for frequencies below $\lambda/10$ or

$$f = \frac{c}{20 r_f}. \quad (9)$$

The first resonant frequency, f_r, for this model is

$$f_r = \frac{c}{2(h + z_o)}. \quad (10)$$

The design parameter values given in Table 1 were chosen to provide minimum clearance between the headset earmuff under test, and the noise source speaker. Figure 6 shows the predicted results of eq. 8 with weighting factors $w_1 = w_2 = 0.5$, which are compared with the actual experimental data as discussed in Section 4. Using the test cell dimensions of Table 1, a uniform low frequency plane wave pressure field is predicted for 0-200 Hz. The magnitude of the sound pressure field is predicted to decrease by about 6 dB in the 200-400 octave band. The first resonance, predicted by eq. 10, is at 1580 Hz, which is well beyond the measurement range of the cell.

3. LFATC DESCRIPTION AND ISOLATION

Based on model simulations, the LFATC, shown in Figures 7 and 8, is constructed following the pattern in Ryan et al. [2]. The LFATC is constructed from 1.27 cm thick aluminum (6061 – T5) plate and cylinder. It rests on an aluminum flat plate base of dimensions 43.18 x 43.18 x 10.16 cm. The upper chamber is assembled from a cylinder 12.7 cm long with an inside diameter of 22.86 cm. It is terminated with a 7.62-cm thick aluminum plug. The remaining 5.08 cm depth of the upper chamber is filled with felt, except for that portion occupied by the back of the speaker. The lower chamber's dimensions are given in Table 1 and its wall is covered by a 3.81-cm thick annulus of felt. The horizontal gasket, labeled damping in Ryan et al. [2], is omitted. A 15.24-cm diameter Rockford Fosgate 100-watt speaker,

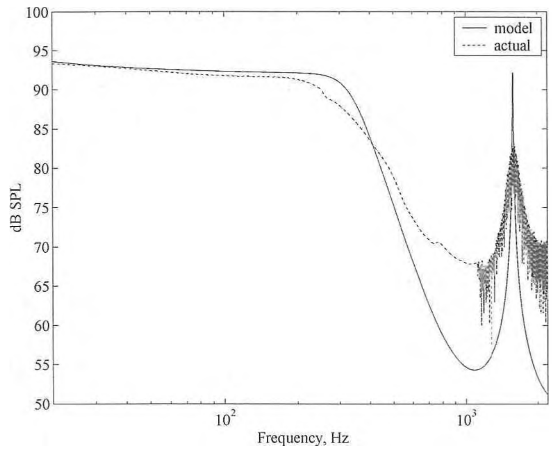

Figure 6. Comparison of model and experimental frequency response.

model FNQ1406, provides the sound source. A B&K (4190) precision microphone is mounted axially in the center of the base. Another precision microphone is mounted radially through the midpoint of the lower volume cylinder wall. This dual microphone arrangement allows the simultaneous measurement of acoustic pressures both internal and external to the communication headset.

A flat plate headset attachment is provided by two 0.32-cm eyebolts, mounted 5.08 cm radially from the axis to which precision dental ligatures are attached. The end of each ligature is attached to studs protruding from the head-set. This arrangement ensures a uniform seal pressure between the headset cuff and the flat plate.

Extensive rotating machinery in adjacent spaces and construction at a nearby facility necessitated the isolation of the LFATC from the floor of the laboratory. Accordingly, the LFATC is isolated from the floor by using two layers of air packing material, 30.48 cm in diameter, separated by a 50.8 x 63.5 x 1.27-cm brass plate. The test cell is supported on this isolation by a 50.8 x 63.5 x 1.27-cm plywood bed. The compressed height of the plywood bed from the floor is 45.72 cm.

4. LFATC CALIBRATION DATA

Figure 9 shows the noise floor within the test cell volume as measured at the base of the test cell, (h, r_a) = (0 cm, 0 cm), by the installed B&K microphone. This noise floor establishes the maximum performance level of active noise reduction measurable in the test cell. The 'without' vibration isolation indicates the noise floor obtained with the LFATC sitting directly on the lab floor. The vibration isolation provides a 45 dB reduction in the noise floor in the 50-60 Hz range and a lower threshold of 50 dB for the sound pressure level is satisfied above 50 Hz. At 20 Hz the noise floor limits the lowest sound pressure level to 60 dB. A test consisting of discrete pure tones was performed throughout the band of interest at the maximum design pressure level of 140 dB to verify the performance of the test cell. No difference was found in the noise floor as measured at the B&K microphone within a mounted headset and without a headset in place.

The predicted frequency response and characteristics of the gradient pressure field within the LFATC were verified

Table 1. Design parameters.

Parameter	Value
ρ_{air}	1.293 kg/m^3
c_{air}	334 m/s^2
r_a	7.62 cm
r_f	3.81 cm
h	7.62 cm
z_0	2.96 cm
$Z_{aluminum}$	1.7 x 10^7 Ns/m^5
$Z_{speaker}$	0.9 x 10^7 Ns/m^5
δ	0.996 m^{-1}

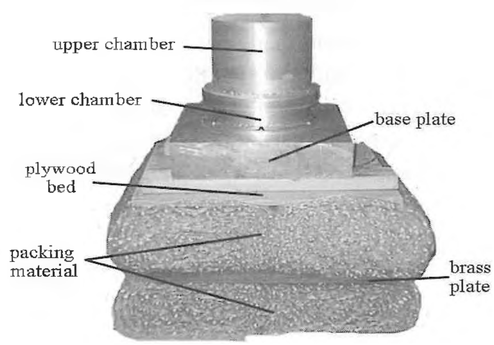

Figure 7. Assembled view of the LFATC.

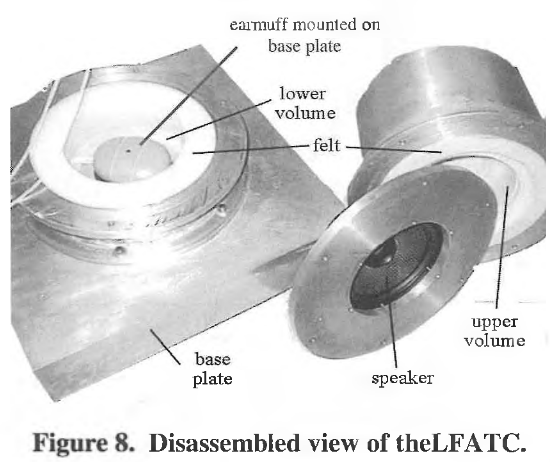

Figure 8. Disassembled view of theLFATC.

experimentally. The speaker was chosen to have sufficient power rating and dynamic range to make the transfer function magnitude a constant gain, K_I, in the frequency band of interest. The excitation for the experiment was a 2-volt peak-to-peak chirp from 1 to 10 KHz of 2 minutes duration. The results are shown in Figures 6, along with the theoretical frequency response predicted by eq. 8.

The transfer functions based on experimental data for three positions on the LFATC axis, (h, r_f) = (0 cm, 0 cm), (2.54 cm, 0 cm), and (5.08 cm, 0 cm), are shown in Figure 10. Experimentally, the quality factor, Q, is significantly lower than predicted due to the felt liner in the actual LFATC. A second experiment using a 0.2-volt peak-to-peak excitation, an order of magnitude lower than the previous experiment, was conducted to show linearity. These transfer functions, which are not shown, show no perceptible differences from those in Figure 10.

These results demonstrate a linear test volume exists between 20 and 200 Hz and 50 and 140 dB. Indeed, the results of this calibration, as seen in Figures 6 and 10, indicate that the LFATC has the properties to permit measurements up to 1000 Hz, or two or more octaves above the 50-200 Hz frequency range chosen for the experimental evaluation of ANR algorithms.

5. LFATC Practical Demonstration

As previously mentioned, the authors are performing research in low frequency, feedforward ANR algorithm development and evaluation for communication headsets. This research requires experimental comparison of candidate ANR algorithms that are variants of the LMS family of algorithms. Candidate algorithms have been developed based on a Lyapunov tuning method described by the authors in [10]. These algorithms are variants of the well-known leaky, normalized LMS (LNLMS) algorithm [13] and are designed to optimize both low frequency stability and active noise reduction performance over a large dynamic range for non-stationery noise sources. The Lyapunov tuned algorithms provide a time-varying leakage factor and step-size combination that retains acceptable stability of the weight update equation at low signal-to-noise ratio (SNR) on the measured reference input, while maximizing ANR performance at both low and high SNR. The Lyapunov tuning method also addresses the need to eliminate empirical tuning of LNLMS algorithms. The use of the LFATC is demonstrated by comparison of a Lyapunov-tuned algorithm to traditional normalized (NLMS) and fixed-leakage factor LNLMS algorithms in conjunction with a prototype headset, with highly non-stationary, F-16 aircraft cockpit noise. The F-16 noise is band limited to 50 Hz due to the high distortion levels of the small, commercially available, noise cancellation headset speakers operating at high reference sound pressure levels.

Low and high SNR performance comparisons of a Lyapunov-tuned leaky, normalized LMS algorithm to a fixed leakage factor LNLMS and NLMS algorithms are shown in Figures 11 and 12, respectively. Noise reduction performance is measured at the prototype headset's error microphone. The figures represent the ensemble average of four sets of independent measurements. In Figure 11, at low SNR the NLMS algorithm is unstable, due to weight drift induced by measurement noise on the reference input. Instability is indicated by a non-convergent sound pressure level (SPL) within the five-second sample. The constant leakage factor LNLMS filter is empirically tuned to regain stability at low SNR, resulting in a performance degradation of approximately 10 dB. The Lyapunov-tuned LMS filter is shown to provide stability, while enhancing noise reduction performance over the fixed-leaky LNLMS filter by 5 dB.

When these algorithms are applied at high SNR, as

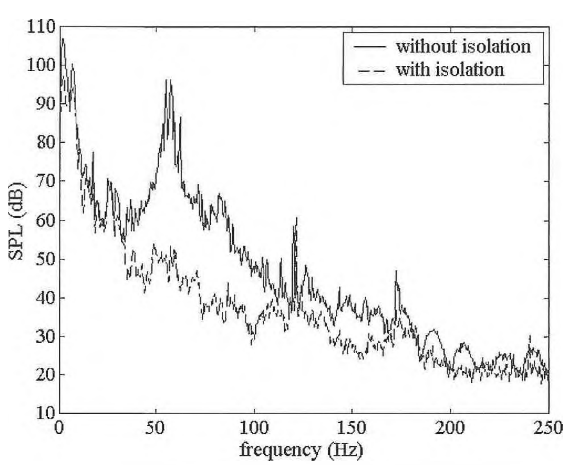

Figure 9. Noise floor measured at bottom center of the base plate at the B&K without and with vibration isolation.

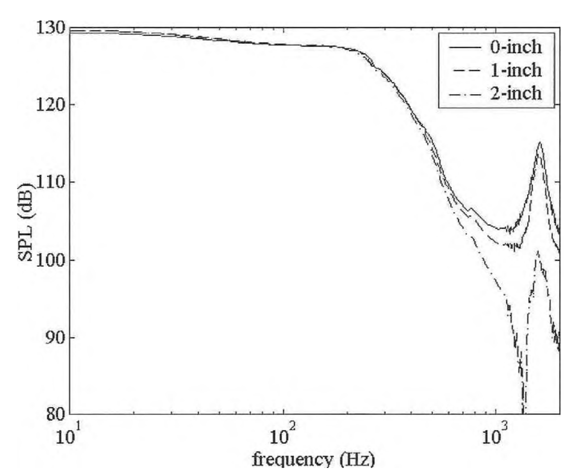

Figure 10. Linearity test at 3 locations from the bottom center of the base plate.

shown in Figure 12, the NLMS filter remains unstable, though the divergent behavior is imperceptible in the five-second sample. Nevertheless, the NLMS filter performance provides the "best case" target noise reduction performance for the family of normalized LMS algorithms. Again, the fixed-leaky LNLMS filter results in a performance degradation of 10 dB. The Lyapunov-tuned LMS filter is stable and provides no ANR performance degradation over the NLMS filter.

These results demonstrate that the Lyapunov-tuned filter has performance characteristics approaching the NLMS filter at high SNR, while retaining stability at minimal performance degradation at low SNR, compared to an empirically tuned fixed leakage factor LNLMS filter. In Figure 11, the minimum SPL attained is well above the 50 dB noise floor design specifications. The LFATC provides a precise

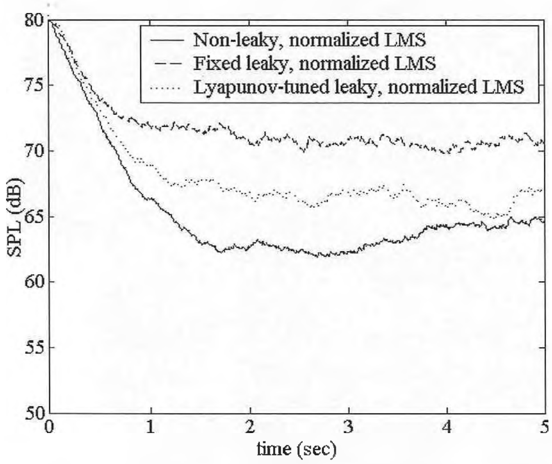

Figure 11. Comparison of three LMS algorithms canceling 80 dB F-16 noise inside the prototype headset. This represents the performance potential during low SNR, 35 dB.

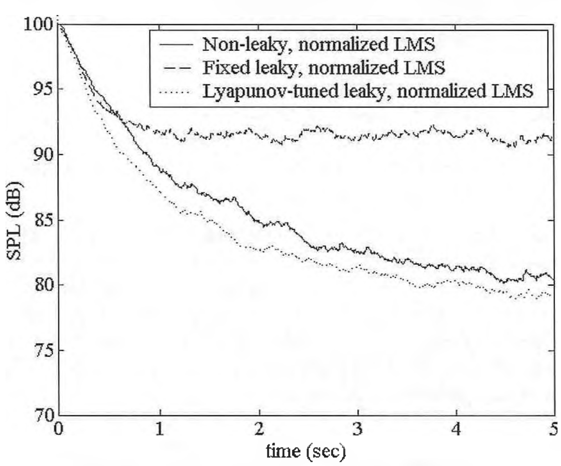

Figure 12. Comparison of three LMS algorithms canceling 100 dB F-16 noise inside the prototype headset. This represents the performance potential during high SNR, 55 dB.

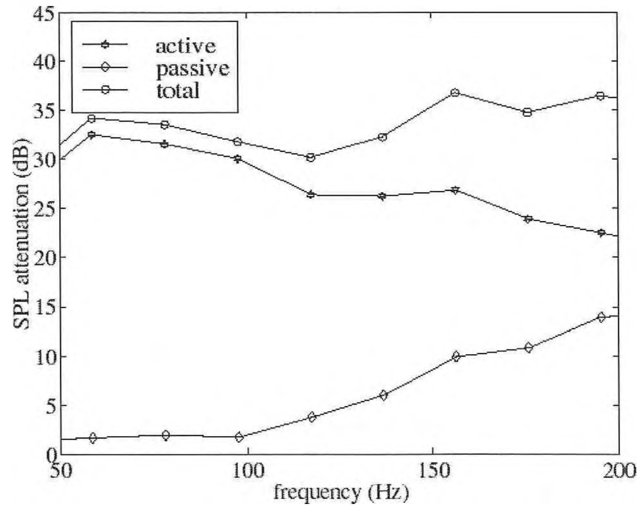

Figure 13. Prototype headset noise reduction performance at the error microphone.

evaluation of convergent, divergent, and steady state performance of prototype headsets and ANR algorithms as a function of SNR.

Figure 13 shows the passive attenuation, active noise reduction, and combined noise reduction performance of the Lyapunov-tuned leaky NLMS algorithm at the error microphone of the prototype headset in response to a sum of pure tones, {50, 63, 80, 100, 125, 160, 200} Hz. The passive attenuation values of the prototype headset were somewhat lower than those values shown in Figure 1 for commercial headsets due to wiring modifications. On the other hand, compared to the best-case active performance of feedback ANR headsets presented in Figure 2, measured active noise reduction using the Lyapunov tuned algorithm provides an additional 10-20 dB active noise reduction in the 50-200 Hz frequency band.

6. CONCLUSIONS

A linear acoustic test volume for the evaluation of low frequency active noise reduction in circumaural headsets and hearing protection has been designed, constructed, and validated experimentally. The LFATC volume dimensions are derived using two acoustic models in its design. These are a lumped-parameter model and a one-dimensional standing wave model. The vibration isolation system provides up to a 45 dB improvement in the noise floor within the bandwidth of interest. This LFATC has been used to evaluate feedforward ANR algorithms and communication headsets in the bandwidth of 50-200 Hz over a dynamic range of 65 dB.

ACKNOWLEDGEMENT

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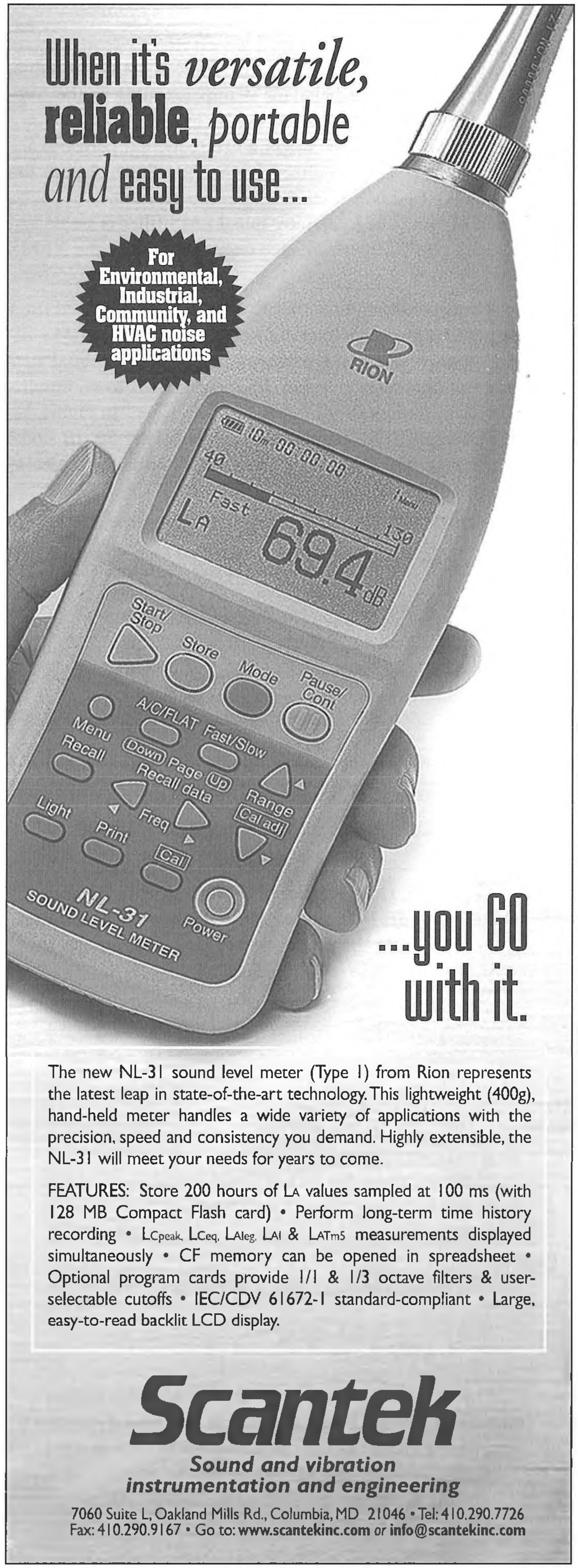

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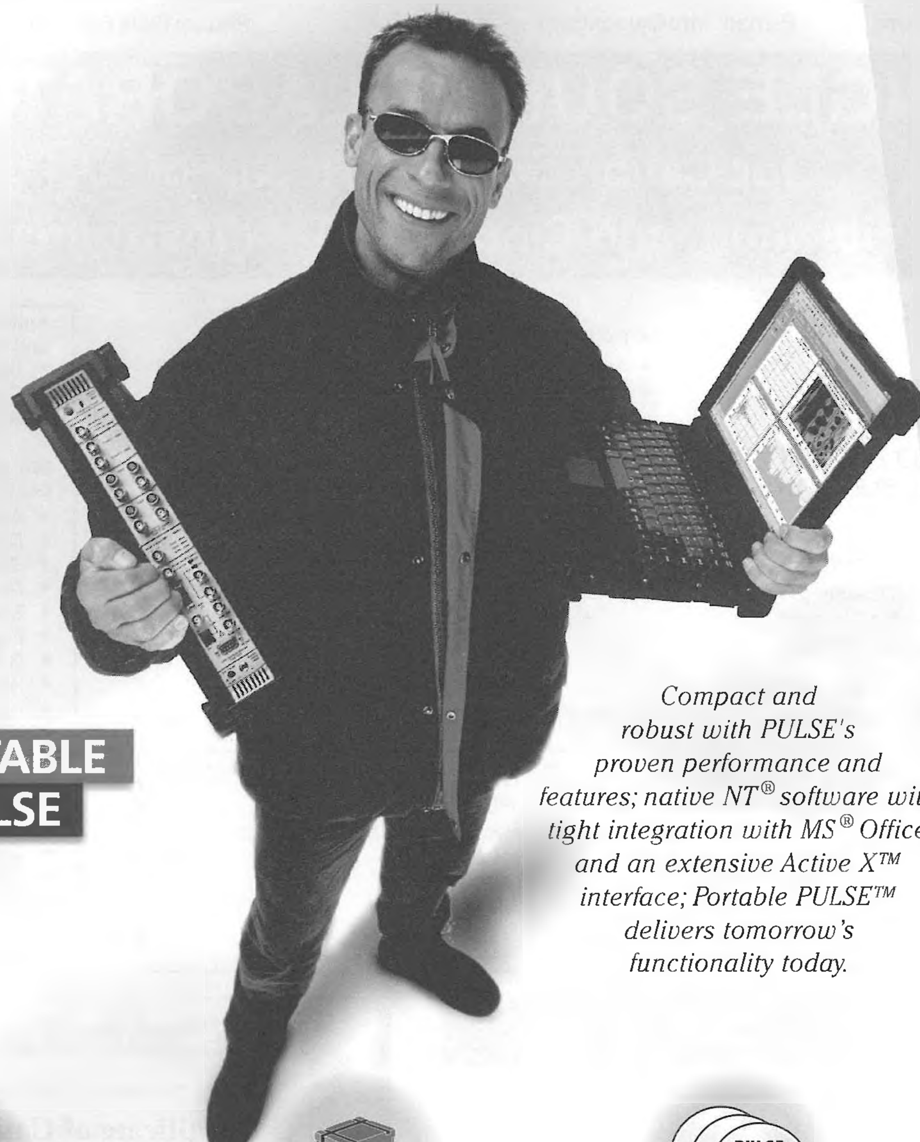

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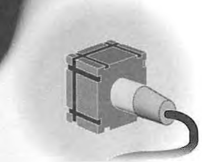

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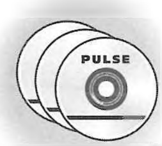

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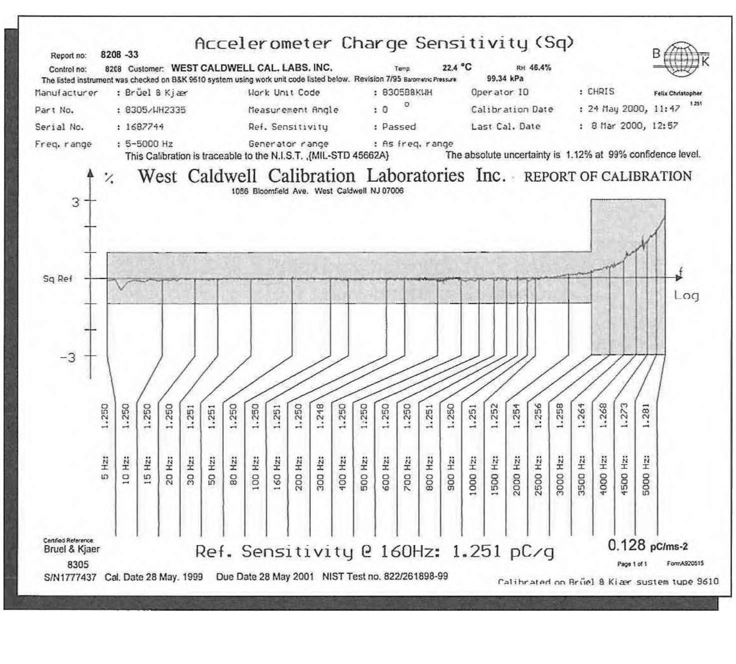

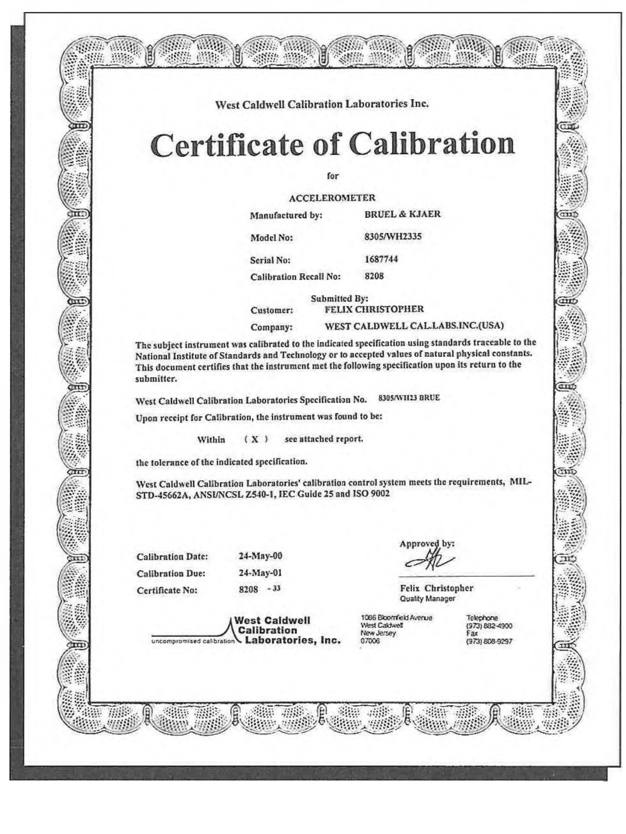
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CASE STUDY OF NOISE ATTENUATION OF LARGE AUTOMOTIVE MANUFACTURING FACILITY

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ABSTRACT

This case study reports on the efforts taken to significantly attenuate the overall noise exposure on several residential receptors from the operations of a large scale automotive manufacturing facility. The exercise began with an extensive noise survey of the facility to identify the significant noise contributors followed by the modeling of the noise propagation from these sources. Noise abatement measures were then modeled to predict attenuation values at the receptors which were then compared to legislated acceptable levels. A noise abatement plan was created along with a schedule to facilitate the attenuation recommendations. While not complete, the measures implemented thus far have exceeded all expectations.

SOMMAIRE

La présente étude de cas rapporte les efforts faits pour diminuer considérablement le bruit parvenant à plusieurs maisons en un quartier résidentiel des opérations d'une grande usine de construction automobile. On a commencé par dresser le plan des bruits créés par celle-ci afin d'identifier leurs sources importantes; ensuite on a modélisé la propagation du bruit émanant de ces sources. Des mesures de réduction du bruit ont ensuite été modélisées pour prédire le taux de diminution perçue dans le quartier, qui a été comparé aux niveaux permis par la loi. On a dressé un plan de réduction du bruit et un calendrier de sa mise en oeuvre. L'effet des mesures déjà exécutées a dépassé les espérances.

1. INTRODUCTION

This case study is the culmination of what originated as a two part effort. The first part was to conduct an extensive survey of existing noise levels impacting on immediate residential receptors from the activities of a large automotive manufacturing facility. The second part was to establish a noise abatement action plan to bring all noise levels within legislated guidelines. Since this facility is in operation 24 hours a day, the target time period for noise attenuation was during the night when acceptable levels are most stringent. This action plan was not to exceed two years in duration. The need to conduct this study was in part due to conditions which precipitated from a previous application to the Ontario Ministry of the Environment for a Certificate of Approval (Air) for a newly installed cooling tower. While the noise emissions from the cooling tower were found to be acceptable, other existing sources of noise were identified to be far in excess of ministry guidelines. Initially, day and nighttime measurements at all sensitive residential receptors were made with and without the plant in operation. In addition, all significant sources of noise emission from the building perimeter, openings and surrounding equipment were identified and quantitatively evaluated. The results of these measurements were then compared to the allowable noise levels as specified by the MOE in Publication NPC-205. From these results, problem areas were identified and noise abatement recommendations, including expected attenuation levels, were made for each of the identified significant noise contributors. This plan included a timetable for the implementation of the recommended mitigating measures.

2. INITIAL INVESTIGATION AND ANALYSIS

2.1 Identification of Residential Receptors and Measurement Results

Five sensitive residential receptors were identified around the property line perimeter of the manufacturing facility from which measurements were conducted. These were done during a shutdown to illustrate ambient conditions as well as with the plant in full production.

For the first location, the sound level meter was located at the property line some 80 metres from the nearest corner of the plant. This site was chosen for its close proximity to the truck entrance of the plant property as well as the loading bay where raw steel rolls are unloaded and stored from flatbed trucks. The second location was chosen for its close proximity to the truck turning area and the crane bay which has a large opening in the wall. The nearest residential receptor at this location is approximately 40 metres away. It should be noted that there is a Canadian National Railway line between this measurement location and the residential receptor. The third site was chosen for its proximity to the longest continuous exterior wall of the manufacturing section of the plant. This wall has several doors and windows as well as various pieces of equipment all of which are potential sources of noise. The fourth location was on the roof top of a storage warehouse building on the west side of the facility property line. This is the site of the nearest residential receptor for the entire plant. These measurements were taken directly in the plane of a second story bedroom window of the residential receptor. No measurements were made at ground level for this receptor since the warehouse building provided significant shielding from the plant operating noise. Location 5 was adjacent to the property line of a residential home which was directly exposed to several noisy pieces of equipment including the cooling tower that initiated the entire exercise. A summary of the Equivalent Sound Levels recorded for each of the measurement locations is given in Table 1.

Inspection of Table 1 shows that the noise levels measured at the chosen receptor locations during plant operation are significantly higher than the ambient noise levels at these same locations measured during the plant shut down. It should be noted that the night time measurements were conducted at the most critical locations when the guidelines are most stringent to illustrate the worst case scenario. Only day measurements were conducted at the first two locations, since truck traffic is not accepted at night. The proposed abatement measures for each of the five locations are discussed in detail in section 3.

Location #	Measurement Period	Plant Operating Condition	Leq (dBA)
1	Day	Not operating	59.1
		Operating	60.4
2	Day	Not Operating	56.2
		Operating	58.5
3	Night	Not Operating	48.5
		Operating	64.7
4	Night	Not Operating	46.1
		Operating	63.9
5	Night	Not Operating	50.2
		Operating	63.6

Table 1: Noise Measurements at Residential Receptors.

2.2 Identification of Significant Noise Contributors

In order to properly attenuate the noise at each of the receptor locations, identification and quantification of the noise contributors for each of the locations were conducted.

For the first two locations, there was very little difference between the sound levels with the plant in full operation and during shutdown. The significant noise contribution was from trucks sitting idle in the parking lot after dropping off their cargo. The increase in sound level at location 2 with the plant operating was the result of CN Rail traffic and not plant operations, therefore, no additional abatement measures were deemed necessary here.

Location 3 had several noise contributors. The identification of these sources along with overall sound level measurements at the source and receiver are given in Table 2.

Assuming a hemispherical radiation of noise, and from the sound level for each source at a one metre distance, the attenuated sound level contribution for each noise source at the receptor can be predicted. This distance attenuation is derived by the equation;

$$LP2 = LP1 - 10 \log (R2/R1)$$

Using the predicted noise contribution from each source at the receptor location, a predicted total sound level was calculated to be 69.7 dBA. This predicted noise level is approximately 5 dB greater than the measured noise level for this same location. This discrepancy is due to the fact that the

Source Item	Sound Level @ 1 m (dBA)	Distance to Residential Receptor (m)	Sound Level at Receiver (dBA)
3 Opened Windows	74.8	50	57.8
Fresh Air Supply Fan	77.3	45	61.2
Air Blower Unit	76.9	45	60.4
Compressor Room Intakes	81.8	59	63.9
Dust Collector Unit	83.3	70	64.8
48" Exhaust Fan	77.9	72	59.3
Total Sound Level at Residential Receptor (dBA)			69.7

Table 2: Predicted Sound Levels at Nearest Residential Receptor for Location 3.

predicted noise assumes that all sources of noise are present continuously, when in reality, many of them including the fans, compressors etc. cycle on and off throughout the day.

The significant contributors to the noise measured at location 4 were the open second story windows facing the residential receptor. These very long opened windows were approximately 50 metres away from the property line of the residential receptor. Measurements were made at a distance of one metre from the windows with the windows both opened and closed. With the windows open the sound level at the source was 85.5 dBA. Modeling these long windows as a line source, the expected level at the receptor would be 68.5 dBA. This is 4.6 dBA greater than the measured value of 63.9 dBA. This difference may be because of the directional characteristics since the line source is not perpendicular to the reception point and covers a viewing area from only 25 degrees to 50 degrees. With the windows closed, a 10 dB reduction was realized. Although significant, the reduction achievable by closing the windows is not enough to meet ministry standards for night time noise.

The significant contributors to the noise measured at location 5 were again the open second story windows facing the residential receptor as well as seven first floor doors and windows, a set of coolers, a water cooling tower and three roof stacks. The seven doors and windows are all in the same area and are assumed to produce equal amounts of noise. The second floor windows were assumed to act as a line source while the rest of the sources were assumed to radiate spherically. The identification of these sources along with overall sound level measurements at the source and receiver are given in Table 3. The modeled sound level of 65.5 dBA is only slightly higher than the measured sound level of 63.6 dBA.

Source Item	Sound Level @ 1 m (dBA)	Distance to Residential Receptor (m)	Sound Level without Attenuation (dBA)
2nd Story Windows	85.5	107	65.2
1st Floor Windows	91.8 (total)	99	51.9
Coolers	81.1	86	42.4
Cooling Tower	76.5	92	37.2
Stack #13	81.7	96	42.1
Stack #14	76.3	97	36.5
Stack #15	78.2	98	38.4
Total Sound Level at Residential Receptor (dBA)			**65.5**

Table 3: Predicted Sound Levels at Nearest Residential Receptor for Location 5.

3. NOISE ABATEMENT PROCEDURE

Once the significant noise contributors were identified and measured, the next step was to establish abatement measures to bring the theoretical noise levels to within ministry standards. Consideration given to the abatement recommendations included capital cost, implementation time frame and most importantly expected attenuation for each of the noise contributors.

3.1 Location 1 and 2

As was mentioned earlier, the only significant noise impact from the plant's activities at this location was identified as idling transport trucks which would frequently park and idle near the residential house for any where from 5 to 30 minutes presumably to fill out log books after being loaded at the docking bays. The proposed solution for this occasional source of noise was to prevent the practice of idling transport trucks near this residential receptor. This was be accomplished by placing large signs in the driveway area indicating that trucks are not permitted to stop and idle their engines in this area. These rules where enforced by plant security staff who where responsible to ensure that the truck drivers were aware and abided by the new rules. Given this, it is felt that no other abatement measures were required in this area.

3.2 Location 3

The ambient noise measurements at location 3 were about 48.5 dBA at night while the measurements conducted during plant operations, also at night, were approximately 64.7 dBA. This represents approximately a 16 dBA difference and a violation of the MOE limits. The representation of this location is amongst the worst case scenario for the entire facility. The significant contributors to the noise measured at this location are identified in Table 4. Table 4 shows the sound levels measured at a distance of one metre for each of these sources, the proposed abatement measure with its corresponding attenuation value, the attenuation due to the distance to the receiver and the expected sound levels at the receiver with and without the implementation of the proposed attenuation.

The three windows, located very close together, are assumed to act as a single noise source with a total sound level measured at one meter of 74.8 dBA. Assuming a hemispherical radiation of noise, the sound level at the receptor would be 57.8 dBA for this source alone. This hemispherical radiation

Source Item	Sound Level @ 1 m (dBA)	Abatement type	Predicted Attenuation @ Source (dB)	Distance to Residential Receptor (m)	Distance Attenuation (dB)	Sound Level without Attenuation (dBA)	Sound Level With Attenuation (dBA)
3 Windows	74.8	Louvre	19.2	50	17	57.8	38.6
Fresh Air Supply Fan	77.3	Louvre / Enclosure	17.1 + 9	45	16.5	61.2	34.7
Air Blower Unit	76.9	Louvre	19.4	45	16.5	60.4	41
Compressor Room Intakes	81.8	Barrier	19.7	59	17.9	63.9	44.2
Dust Collector	83.3	Muffler / Shield	8.4 + 11.3	70	18.5	64.8	45.1
48" Exhaust Fan	77.9	Louvre / Barrier	16.8 + 15.1	72	18.6	59.3	27.4
Total Sound Level at Residential Receptor (dBA)						**69.7**	**49.2**

Table 4: Predicted Sound Levels at Nearest Residential Receptor - Location 3.

assumption was also applied to all other sources along this section of the plant. It was proposed that the installation of louvres over the windows would allow air intake during the summer while the noise produced by this source could be attenuated by 19.2 dB making the total contribution from this source at the receptor 38.6 dBA. Table 5 shows the octave measurements of the windows without the louvres, the louvre attenuation values provided by the manufacturer and the expected attenuated values for the windows with the louvres.

The total sound level measured for the fresh air supply fan was 77.3 dBA at one metre with an expected contribution at the receptor of 61.2 dBA. It was proposed that an enclosure be constructed around the fresh air supply fan unit with a louvred intake. The louvred intake would be the same as that specified previously for the windows and was predicted to provide 17.1 db of attenuation. The enclosure was be constructed of 4 inch thick sound attenuation batt insulation, or rockwool, and capped with 18 gauge metal. This significant surface mass is assumed to provide at least as much attenuation as the 4 inch material capped with 5/8 inch gypsum thus providing an STC rating of at least 60. It was difficult to establish the total attenuation expected by the enclosure alone since it is difficult to determine how much noise was radiating from the fan structure separate from the intake noise due to their close proximately. Therefore, a modest attenuation of 9 dB is assumed thus giving a total contribution from this source at the receptor of 35.1 dBA.

The total sound level measured for the air blower unit was 76.9 dBA at one metre with an expected contribution at the receptor of 60.4 dBA. It was proposed that louvres be installed on the intake of this unit thus providing for an attenuated noise level at the receiver of 41.0 dBA.

The total sound level measured for the compressor room intakes was 81.8 dBA at one metre with an expected contribution at the receptor of 63.9 dBA. It was proposed that along this section of the plant, a 4.6 metre (15 foot) tall barrier be installed along an 80 foot length of the wall. It was recommended that the acoustical wall be located 2.1 metres from the building wall and have 1.5 metre returns at each end. The barrier would block line of sight for the compressor room intakes, a man door, the lower portion of the dust collector unit and the 48 inch prop fan to be discussed later. The barrier was constructed of 4 inch thick sound attenuation

Octave Band Frequency, Hz	Measured Value (dB)	Insertion Loss, dB	A-Weighting	A-Corrected Attenuation
16	78.9		-50.5	28.4
31.5	79.2		-39.4	39.8
63	76.3	-16	-26.2	34.1
125	76.2	-14	-16.1	46.1
250	70	-15	-8.6	46.4
500	73	-19	-3.2	50.8
1000	68.8	-23	0	45.8
2000	65.4	-19	1.2	43.6
4000	67.4	-19	1	49.4
8000	58.8		-1.1	38.7
10000	42.9		-4.3	38.6
Total Attenuated Sound Level (dBA)				**55.6**

Table 5: Window Louvre Attenuation.

1	Octave Band Centre Frequencies	63	125	250	500	1000	2000	4000	8000
2	Measured Octave Sound Levels (dB)	78.2	82	81.9	80.2	76.4	72.4	68.9	59.6
3	A-weighting for band	-26.2	-16.1	-8.6	-3.2	0.0	1.2	1.0	-1.1
4	Measure Weighted Octave Sound Levels (dBA)	52	65.9	73.3	77	76.4	73.6	69.9	58.5
5	Predicted total sound level (dBA)	**81.8**							
	Barrier Calculation								
6	Dsb [7 ft] (metres)	2.1	2.1	2.1	2.1	2.1	2.1	2.1	2.1
7	Spacial Correction to Dsb to account for distributed noise source	3.2	3.2	3.2	3.2	3.2	3.2	3.2	3.2
8	Dbr (metres)	56.9	56.9	56.9	56.9	56.9	56.9	56.9	56.9
9	Bh metres	4.6	4.6	4.6	4.6	4.6	4.6	4.6	4.6
10	Sh (2/3) of dryer base height	1.2	1.2	1.2	1.2	1.2	1.2	1.2	1.2
11	Receiver height [1.5 metres] metres	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5
12	Path length difference (metres)	1.553	1.553	1.553	1.553	1.553	1.553	1.553	1.553
13	Fresnel number	0.570	1.131	2.262	4.524	9.048	18.096	36.193	72.386
14	Barrier attenuation (insertion loss) (dB)	10.9	13.6	16.5	19.5	22.5	23.0	23.0	23.0
15	Predicted Octave Sound Levels (dBA)	41.1	52.3	56.8	57.5	53.9	50.6	46.9	35.5
16	**Predicted Lp from base (dBA)**	**62.1**							
17	**Attenuation Due to Distance (dB)**	**17.9**							
18	**Total Sound Level at Residential Receptor from this Source (dBA)**	**44.2**							

Table 6: Compressor Room Intake Noise Barrier Calculations.

batt insulation, or rockwool, and capped with 18 gauge metal. Like the previously discussed enclosure for the fresh air intake fan, the surface mass of this barrier is assumed to provide an STC rating of at least 60. It was assumed that the transmission loss due to the barrier effects was less than the absorbative characteristics of the wall material, therefore, the barrier effects were used for the assumed attenuation of this source. It was determined that with the barrier attenuation that the noise contribution from this source at the receptor would be 44.2 dBA. The barrier attenuation calculations for this source is shown in Table 6.

The total sound level measured for the dust collector unit was 83.3 dBA at one metre with an expected contribution at the receptor of 64.8 dBA. It was proposed that a shield, or noise barrier, be constructed around the fan unit using the same material previously discussed along with a 6 foot duct silencer for the dust collectors exhaust fan. It was predicted that the barrier and silencer combination would provide 19.7 dB of attenuation thus giving a total sound level at the receptor of 45.1 dBA. The attenuation calculations for the barrier is similar to that shown above. The 6 foot silencer recommended for the exhaust fan was a 6 inch diameter duct with 2 inch acoustical lining. The attenuation calculations for the silencer are given in Table 7.

The total sound level measured for the 48 inch propellor exhaust fan was 77.9 dBA at one metre with an expected contribution at the receptor of 59.3 dBA. It was proposed

Octave Band Frequency, Hz	Measured Value (dB)	Insertion Loss, dB	A-Weighting	A-Corrected Attenuation
16	80.7		-50.5	30.2
31.5	81.6		-39.4	42.2
63	75.9	-16	-26.2	33.7
125	87.8	-14	-16.1	57.7
250	79.5	-15	-8.6	55.9
500	76.1	-19	-3.2	53.9
1000	71	-23	0	48
2000	65.8	-23	1.2	44
4000	60	-19	1	42
8000	55.3	-19	-1.1	35.2
10000	46.2		-4.3	41.9
Total Attenuated Sound Level (dBA)				**61.4**

Table 7: Silencer Attenuation.

that an acoustical cowl and louvre be installed on this fan. This fan is also inside of the noise barrier previously discussed for the compressor room intakes and will benefit from this as well. It was predicted that the louvre and barrier would provide 16.8 and 15.1 dB of attenuation respectively. This would result in an overall noise level at the receptor of 27.4 dB for this source.

Measurements indicated that the highest noise level measured at the representative receptor in this area was 64.7 dBA with the plant in operation. The measurement made at this same location with the plant not operating was 48.5 dBA. This would indicate that during the night time, the plant may not produce levels in access of 48.5 dBA at this residential receptor. In other words, the facility needed to attenuate the sources effecting this receptor by 16.2 dBA. Inspection of table 2 indicates that the theoretical sound level at the receptor location with no abatement was 69.7 dBA. With the installation of the proposed noise abatement measures, the noise is modeled to reduce to 49.2 dBA thus providing an overall attenuation of 20.5 dB. This attenuation is 4 dB more than required according to the worst case scenario represented by the measured numbers. It is assumed that any differences between the theoretical and measured values were due mostly to directivity characteristics and that the relative difference is what is most important.

3.3 Location 4

The ambient noise measurement at location 4 was 46.1 dBA at night while the measurements conducted during plant operations, also at night, was 63.9 dBA. This represents a violation of the MOE limits by 18 dB. The significant contributors to the noise measured at this location were open second story windows facing the residential receptor. These windows were approximately 50 metres away from the property line of the residential receptor. With the windows open the sound level one metre from the source was 85.5 dBA. Modeling this source as a line source, the expected level at the receptor was 68.5 dBA. This is 4.6 dBA greater than the measured values which again may be explain by directional characteristics. With the windows closed, only a 10 dB reduction was realized. It was proposed that these windows be permanently closed and blocked with at least 3 inches of sound attenuation batt insulation with a single layer of 0.5 inch gypsum or better in order to achieve an STC rating of 51. This is expected to provide an overall attenuation of 48.4dB as compared to when the windows are in the open position. This would bring the noise contribution from this source at the represented receptor to well below the night time ambient levels. To allow for additional air intake it was proposed that 5 to 6 inline centrifugal fans be install on the roof in a less sensitive location. These fans would also be specified so that they will not add to the existing ambient noise. No other noise sources significantly impacted this receptor due to the significant barrier attenuation achieved from the adjacent warehouse building.

3.4 Location 5

The ambient noise measurements at location 5 was 50.2 dBA at night while the measurements conducted during plant operations, also at night, was 63.6 dBA. This represents a violation of the MOE limits by 13.4 dB. The significant contributors to the noise measured at this location are the open second story windows facing the residential receptor, seven first floor doors and windows, a set of coolers, a water cool-

Source Item	Sound Level @ 1 m (dBA)	Attenuation type	Predicted Attenuation @ Source (dB)	Distance to Residential Receptor (m)	Distance Attenuation (dB)	Sound Level without Attenuation (dBA)	Sound Level With Attenuation (dBA)
2nd Story Windows	85.5	Close and Block	48.4	107	20.3	65.2	16.8
(7) 1st Floor Windows	91.8 (total)	Barrier	11.1	99	39.9	51.9	40.8
Coolers	81.1	Barrier	8.2	86	38.7	42.4	34.2
Cooling Tower	76.5	Barrier	8.8	92	39.3	37.2	28.4
Stack #13	81.7	N/A		96	39.6	42.1	42.1
Stack #14	76.3	N/A		97	39.7	36.5	36.5
Stack #15	78.2	N/A		98	39.8	38.4	38.4
Total Sound Level at Residential Receptor (dBA)						**65.5**	**46.3**

Table 8: Predicted Sound Levels at Nearest Residential Receptor - Location 5.

ing tower and three roof stacks. The seven doors and windows are all in the same area and were assumed to produce equal amounts of noise. The second floor windows were assumed to act as a line source while the rest of the sources were assumed to radiate spherically.

The second floor windows are the same as those considered for location 4. As in the previous case, it was proposed that these windows be blocked with sound attenuation insulation which would give an attenuation of 48.4 dB. For the seven windows and doors, the coolers and the cooling tower it was proposed that a 3.05 metre (10 foot) high and 19 metre wide barrier be constructed to break line of sight to these noise sources. The barrier was to be attached to an existing quench oil storage building which will provide the rest of the required shielding.

The expected attenuation values for these measures are given in Table 8. It can be seen that a total attenuation of 19.3 db was realized. This meets the required attenuation to bring the sound level down to the acceptable ambient levels for both day and night time.

4. ABATEMENT SCHEDULE

In order to complete the above abatement measures, a two year plan was established which was to commence immediately after the plan had been approved by the MOE. During the first year, the following item were proposed to be completed. The signs in the area of residential receptor 1 disallowing the parking and idling of transport trucks were to be erected and enforced by plant security personnel. For location 3, it was proposed that the louvres be installed on the three windows at the east end of the building, the enclosure and louvre were to be installed on the Fresh Air Supply Fan and the louvre was to be installed on the air blower unit. Also, the shield and duct silencer was to be installed on the Dust Collector unit and the louvre and cowl was to be installed on the 48 inch propellor fan. During the second year, it was proposed that the south facing second story windows near receptor 4 be closed and blocked with sound attenuating material and that the 3.05 metre tall barrier wall be constructed in order to protect residential receptor 5 from the coolers, water cooling tower and the open first floor windows and door. Also during this period, it was proposed that the 4.57 metre tall barrier wall be installed along the north wall to protect the residential receptors from any noise from the compressor room air intakes.

5. INTERIM RESULTS

While a final investigation of the results of the abatement measures is still outstanding to date, an interim investigation was conducted with approximately half of the abatement recommendations addressed.
The signs at location 1 indicating that trucks are not permitted to idle on the property have been erected and followup investigations with the nearby residents have confirmed that these new procedures have been successful in lessening the noise impact from the truck traffic.

Along location 3, a noise enclosure has been installed on the 48" Exhaust Fan which resulted in a noise reduction of 21 dB. This is about 4 dB greater than predicted. The dust collector unit was relocated inside the building and all exterior ducting was enclosed with noise attenuating material. At a distance of one metre away, this piece of equipment was now all but inaudible. The Fresh Air Supply fan and Air Blower unit were also enclosed with noise attenuating material and louvres were installed on the intakes. Preliminary measurements indicated that the realized attenuation values are also greater than predicted. The windows along this wall have been permanently filled with solid acoustical panels. And it was observed that no noise was detectable at the openings. The largest noise contributor in this location was from the intake louvres for the air compressor room. Instead of shielding this source as originally suggested, the entire compressor room was relocated in a less noise sensitive area of the plant with new, quieter, compressor units. The noise emissions measured at a distance of one metre from the newly acoustically treated openings was 59 dBA. This is approximately a 23 dB noise reduction. While night time measurements were not made at the residential receptor for this location, it is felt that the improvements made thus far have greatly lessened the impact on these houses.

At the time of the visit, implementation of the remaining abatement measures were still outstanding.

6. CONCLUSIONS

Large scale noise investigations are extremely dynamic in nature with a great deal of variables to consider. While not entirely complete, this case study is on the road to a successful end. The abatement measures implemented thus far have exceeded all expectations.
This case study illustrates how a complex noise problem can be broken down into several small parts and then synthesized back into a whole result. That being an overall reduction in noise to an acceptable level.

This case study also illustrated that while actual measurements and predicted results may differ, they do provide important relative values that can be used as effective tools for noise attenuation prediction and implementation.

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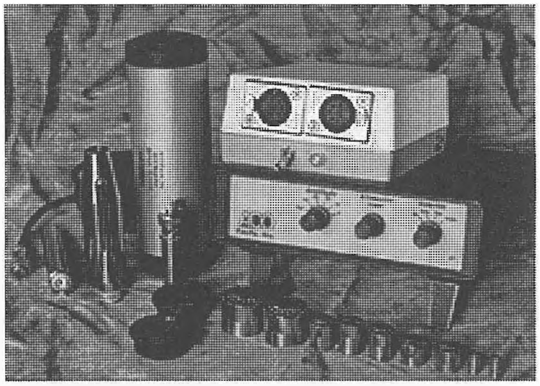

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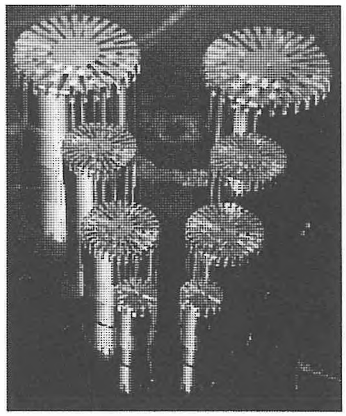

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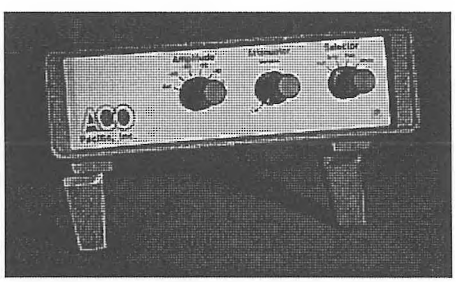

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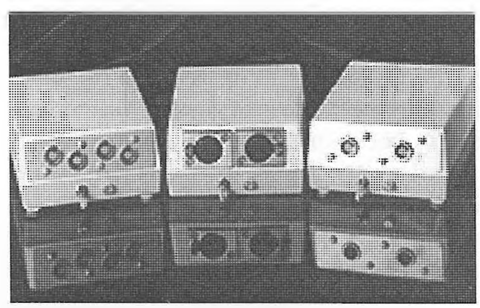

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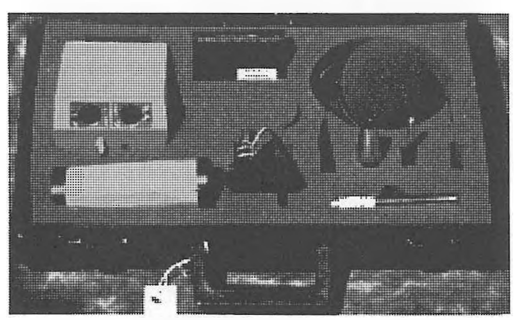

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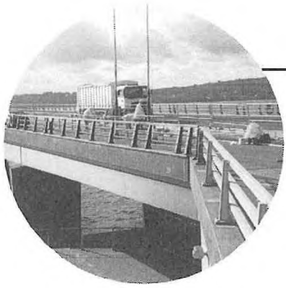

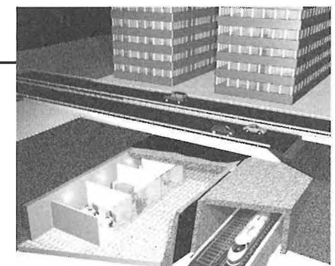

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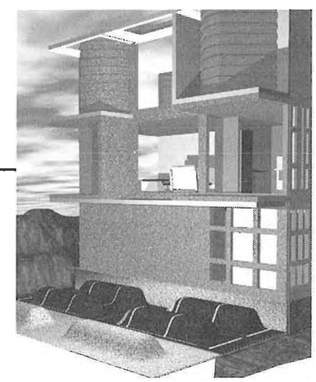

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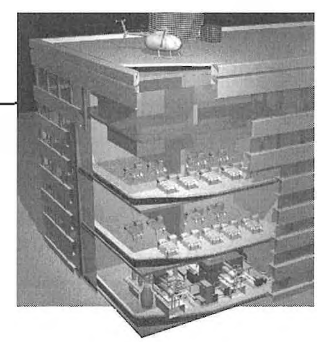

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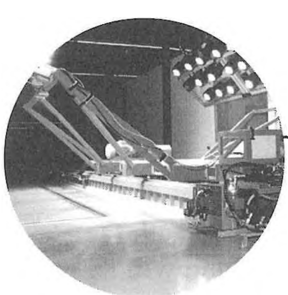

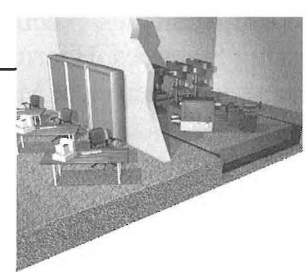

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Invitation to Attend the Annual Meeting of the Canadian Acoustical Association, 2002

Acoustics: Bridge to the Future. Inspired by the unique locale of the Province of Prince Edward Island (PEI) and its recently completed Confederation Bridge, the theme of Acoustics Week in Canada 2002 emphasizes links in the various disciplines of acoustics. For example, close to the site of this next Annual Meeting, models of vibrational modes contributed to the final design of the 13 km bridge from PEI to New Brunswick. In a research station housed inside the bridge, acoustical work continues to monitor vibrations in relation to weather conditions. In the same way that knowledge of acoustics enabled the safe construction and maintenance of the Confederation Bridge, knowledge of acoustics helps forge new paths in domains as diverse as human communication, mapping the seabed, and the aerospace industry. The Annual Meeting of the Canadian Acoustical Association (CAA) builds and reinforces bridges between sub-disciplines of acoustics, over geographical boundaries, and across acoustical and non-acoustical fields.

Contributions from *all* fields of the science of sound are welcome for the CAA meeting, including but not limited to: Architectural Acoustics, Engineering Acoustics/Noise Control, Physical Acoustics/Ultrasound, Musical Acoustics/Electroacoustics, Psychological Acoustics, Physiological Acoustics, Shock/Vibration, Hearing Sciences, Hearing Conservation, Speech Sciences, Underwater Acoustics, Signal Processing/Numerical Methods, and Education in Acoustics.

The short abstract should be prepared and sent (for receipt by May 31, 2002) in accordance with the instructions printed in this issue of *Canadian Acoustics*. Abstracts will be peer reviewed and will be printed and posted. Direct on-line submission will also be available through the conference web-site. For those without access to e-mail, digital files on diskette or paper copy should be mailed to either technical program co-coordinator. The voluntary 2-page proceedings paper is due August 14. This deadline will be strictly enforced to meet the publication schedule of the proceedings issue of the journal *Canadian Acoustics.*

Proposals for Special Sessions on a particular topic in acoustics are welcome. Contact Annabel Cohen or David Stredulinsky soon if you wish to organize a special session.

Student participation in Acoustics Week in Canada is strongly encouraged. **Awards** are available to students whose presentations at the conference are judged to be particularly noteworthy. To qualify, students must apply by enclosing an Annual Student Presentation Award form with their abstract. Students presenting papers may also apply for a travel subsidy to attend the meeting if they live at least 100 km from Charlottetown. To apply for this subsidy, students must submit an Application for Student Travel Subsidy. Forms are also available on the web-site.

Accommodation. The Delta Hotel located in downtown Charlottetown PE will provide accommodation and meeting space (http://www.deltaprinceedward.pe.ca). The special (double/single occupancy) room rate for delegates is $109.00 per night. This rate applies to several days prior and after the conference (including the following Thanksgiving weekend). To reserve accommodation, please contact the hotel directly by telephone (1-800-268-1133; Fax 1-902-566-1745) and mention the CAA meeting. You may also contact Jason Clark directly (jclark@deltahotels.com, 902-894-1237, fax 902-566-1746). The reservation cut-off date is August 27, 2002. After these dates, the special rates are subject to availability. Several other hotels for every budget are located within walking distances from the conference site. For details, check the PEI tourist web site: http://www.gov.pe.ca/visitorsguide/

Workshops/Seminars. As a tradition, several half-day and full-day workshops may be offered Oct. 8, the day before the technical program and exhibition begins, giving opportunity for continuing education in acoustics. If you are interested in giving a workshop, please contact the convener. An IRC/NRC full day seminar "Containing Fire and Sound: Challenges and Solutions" is now scheduled. It focuses on design tradeoffs to deal effectively with both fire resistance and sound insulation between units in multi-family buildings. (Contact Dave.Quirt@nrc.ca, tel: 613-954-1495).

Exhibits. An exhibition of the latest technologies in acoustics and vibration equipment, materials and software will take place Wed. and Thurs. Oct. 9-10. Exhibitors will be well integrated into the conference setting and featured in a special session of the conference program. Sponsorship by exhibitors of nutrition breaks and/or lunches is also welcome. (Contact the conference convener or Teresa Drew).

Canadian Standards Acoustics. Canadian Standards Association Committee Z107 in Acoustics and Noise Control will hold a meeting (organizer: Cameron Sherry, Cwsherry@aol.com). All welcome.

Hospitality. In the tradition of past CAA meetings, a full program is planned for receptions, meals, banquet, award ceremony, and a sample of the best Prince Edward Island and Maritime hospitality and entertainment.

Important Dates 2002	
Fri., May 31	Deadline for receipt of abstracts
Fri., June 28	Notice of acceptance of abstracts
Wed., August 14	Deadline for receipt of summary paper and early registration
Tues., October 8	Acoustics Week in Canada begins: Workshops/Seminars
Wed.-Fri. October 9-11	Acoustics Week in Canada: Technical Program and Exhibition

Contact Persons & Information	
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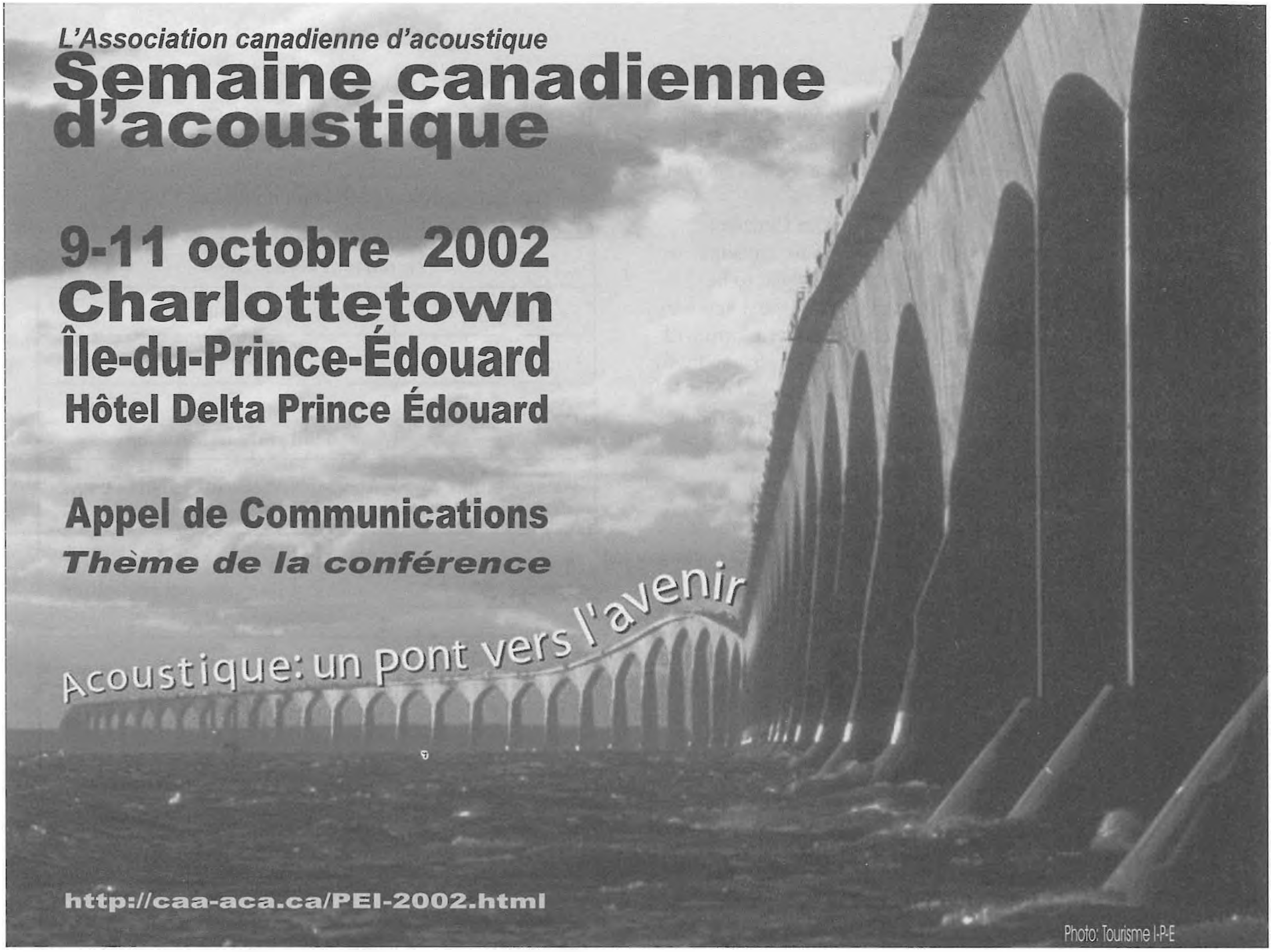
Audio Visual **Robert Drew**	rdrew@upei.ca Dept. of Psychology (902) 628-4331 FAX: (902) 628-4359
Web-site address	http://caa-aca.ca/PEI-2002.html



Invitation à participer

La réunion annuelle de l'Association canadienne de l'acoustique, Charlottetown 2002

Acoustique: Un pont vers l'avenir. Inspiré par l'unique situation de la province de l'Île-du-Prince-Édouard (ÎPÉ) et le Pont de la Confédération qui fut récemment complété, le thème de la semaine d'acoustique au Canada 2002 souligne les liens entre les disciplines variées de l'acoustique. Par exemple, situé non loin du site de cette prochaine réunion annuelle de l'ACA, des modèles de modes de vibrations ont contribué au plan final du pont de 13 km qui rejoint l'IPÉ au Nouveau-Brunswick. Le travail en acoustique se poursuivent dans la station de recherches située dans l'intérieur du pont où les vibrations sont analysées en correspondance avec les conditions climatiques. De la même façon que nos connaissances en acoustique ont put faciliter la prudente construction et entretien du pont de la Confédération, nos connaissances en acoustique facilitent la création de nouvelles directions dans des domaines aussi divers que la communication humaine, la caractérisation du fond de la mer, et l'industrie aérospatiale. La réunion annuelle de l'Association canadienne d'acoustique crée et renforce les liens entre les sous-disciplines de l'acoustique à travers les frontières géographiques et entre les domaines acoustiques et non-acoustiques.

Les contributions de toutes les disciplines de la science du son seront les bienvenues pour la réunion de l'ACA, incluant mais non limitées à: l'acoustique architecturale, le génie acoustique et contrôle du bruit, l'acoustique physique et l'ultrason, l'acoustique musicale et l'électro-acoustique, la psycho-acoustique, l'acoustique physiologique, les chocs et vibrations, l'audiologie, la science du langage, l'acoustique sous-marine, le traitement du signal et les méthodes numériques, et finalement l'éducation en acoustique.

Les résumés seront préparés et envoyés (pour être reçus avant le 31 mai 2002) suivant les instructions incluses dans ce numéro d'Acoustique canadienne. Les résumés seront examinés par un pair et publiés. Les soumissions par courrier électronique seront disponibles sur le site web. Pour ceux qui n'ont pas accès au courrier électronique, les documents digitaux sur disquette ou papier devront être envoyés à l'un des co-présidents du programme technique. Les sommaires optionnels de 2 pages devront Ltre soumis avant le 14 août 2002. Cette échéance sera strictement respectée afin de pouvoir publier le programme dans les actes d'Acoustique canadienne.

Les propositions pour des sessions spéciales sur un sujet particulier en acoustique sont les bienvenues. Contactez Annabel Cohen ou David Stredulinsky si vous désirez organiser une session spéciale durant la conférence.

La participation des étudiants à la conférence de l'ACA 2002 est fortement encouragée. Des **prix** seront accordés aux étudiants dont la présentation à la conférence aura été jugée particulièrement remarquable. Afin d'être éligibles à ces prix, les étudiants doivent remplir le formulaire du Prix Annuel de Présentation Étudiante. Ce formulaire devrait être envoyé avec le résumé. Les étudiants qui habitent dans une région suffisamment éloignée de Charlottetown (plus de 100 km) et qui désirent présenter leur article à la conférence, peuvent également faire application pour une subvention de voyage, afin de défrayer leurs frais de déplacement. Les formulaires sont disponibles sur le site web.

Logement. L'Hôtel Delta Prince Édouard (http://www.deltaprinceedward.pe.ca) situé à Charlottetown ÎPÉ fournira l'hébergement et les salles de réunion. Le prix spécial de chambre (double ou simple) pour les délégués est de 109$ par nuit. Ce prix s'applique aussi pour plusieurs jours avant et après la réunion, y compris la fin de semaine de l'action de grâce. Pour réserver une chambre, s'il-vous-plaît contactez l'hôtel directement (1-800-268-1133; Fax 1-902-566-1745) et mentionnez la réunion de l'ACA. Vous pouvez aussi contacter Jason Clark directement (jclark@deltahotels.com, 902-894-1237; fax: 902-566-1746). Les réservations doivent être faites avant le 27 août 2002. Après cette date, le tarif préférentiel sera sujet à la disponibilité des chambres. Pour d'autres hôtels et auberges, près du site de la conférence, visitez le site web de la ville de Charlottetown (http://www.gov.pe.ca/visitorsguide/).

Ateliers. Suivant la tradition, plusieurs ateliers (demi- ou pleine-journée) pourront être offerts le jour précédent le début des programmes scientifiques et techniques. Si vous êtes intéressé à présenter un atelier, S.V.P. contactez la présidente de la conférence. Un atelier pleine-journée sera offert par IRC/CNRC intitulé "Containing Fire and Sound: Challenges and Solutions". L'atelier porte sur les compromis apportés au design pour améliorer la résistance au feu et l'insonorisation entre les unités de domiciles multi-familiaux. (Personne contact: Dave.Quirt@nrc.ca, tel: 613-954-1495)

Exposants. Une exposition portant sur les dernières technologies entourant l'équipement, l'instrumentation, les matériaux, et les logiciels reliés aux domaines de l'acoustique et des vibrations, sera ouverte mercredi-jeudi (9-10 oct.). Les exposants seront intégrés à la conférence et seront mis en vedette durant une session spéciale du programme. La commandite des périodes de pauses alimentaires et/ou de déjeuners est également invitée. (Contactez Annabel Cohen ou Teresa Drew).

Normes canadiennes en acoustique. Une rencontre du normes canadiennes en acoustique Z107 est organisée par Cameron Sherry (cwsherry@aol.com). Tous sont invités.

L'hospitalité. Suivant la tradition des conférences passées, un programme social sera organisé avec des réceptions, des repas, un banquet, une cérémonie de remise de prix et des exemples d'hospitalité et de divertissement de l'Île-du-Prince-Édouard et des Maritimes.

Dates à retenir	
vendredi, 31 mai	Date limite pour la soumission des résumés
vendredi, 28 juin	Avis pour les résumés approuvés
mercredi, 14 août	Date limite pour la soumission des articles-sommaires et l'inscription à l'avance
mardi, 8 octobre	Semaine canadienne d'acoustique 2002 - Ateliers
mercredi-vendredi 9-11 octobre	Semaine canadienne d'acoustique 2002 programme technique et exposition (9-10)

Personnes contacts	
Présidente de la conférence **Annabel Cohen**	acohen@upei.ca Dept. de Psychologie U. de l'Île-du-Prince-Édouard Charlottetown, PE C1A 4P3 (902) 628-4325 (902) 628-4359 (fax)
Co-président du programme technique **David Stredulinsky**	stredulinsky@drea.dnd.ca DREA 9 Grove St. Dartmouth, NS B2Y 3Z7 (902) 426-3100 x352 (902) 426-9654 (fax)
Secrétaire du programme **Reina Lamothe**	rlamothe@upei.ca Dept. de Psychologie, UPEI (902) 628-4331 (902) 628-4359 (fax)
Exposition **Teresa Drew**	tdrew@jacqueswhitford.com Jacques, Whitford & Assoc. Ltd. (902) 468-7777 (902) 468-9009 (fax)
Audio-visuel **Robert Drew**	rdrew@upei.ca (902) 628-4331 (902) 628-4359 (fax)
Site web	**http://caa-aca.ca/PEI-2002.html**

Instructions for the Preparation of Abstracts for Papers to be Presented at the 2002 Meeting of the Canadian Acoustical Association (see web for electronic submission)

1. One copy of an abstract is required for each meeting paper. Send it to either Technical Program Co-coordinators, Annabel Cohen or David Stredulinsky, to be received by Friday, May 31, 2002.

2. Limit the abstract to 250 words, including title and first author's name and address; names and addresses of coauthors are not counted. Display formulas set apart from the text are counted as 40 words. Do not use the forms "I" or "we"; use passive instead.

3. Use the sample format shown on the second page of these instructions. Title of abstract and names and addresses of authors should be set apart from the abstract as shown. Text of abstract should be one single, indented paragraph. The entire abstract should be typed on one side of 8 ½ x 11 or A4 paper.

4. Be sure that the mailing address of the author to receive the acceptance notice is complete on the abstract, to insure timely deliveries.

5. Do not use footnotes. Use square brackets to cite references or acknowledgements. Give references as shown in the example.

6. At the bottom of an abstract give the following information:

(a) If the paper is part of a special session, indicate the session. If invited state, "invited".

(b) Name the area of acoustics most appropriate to the subject matter: Architectural Acoustics, Engineering Acoustics, Noise Control, Physical Acoustics/ Ultra-sound, Musical Acoustics/ Electroacoustics, Psychological Acoustics, Physiological Acoustics, Shock/Vibration, Hearing Sciences, Hearing Conservation, Speech Sciences, Underwater Acoustics, Signal Processing/ Numerical Methods, Education in Acoustics or other.

(c) Telephone number, including area code, of the author to be contacted for information. Non-Canadian authors should include country.

(d) If more than one author, name the one to receive the acceptance notice.

(e) Overhead projectors will be available. Describe any other special equipment that will be needed.

(f) Indicate your preference to present a lecture or a poster.

[Adapted from Acoustical Society of America Guidelines]

Instructions pour la préparation des résumés des communications qui seront présentées au Congrès de l'ACA 2002. (Consultez le site web pour les soumissions par courrier électronique)

1. Un résumé est exigé pour chaque communication. Une copie de chaque résumé doit être expédiée à l'un des co-présidents du programme technique, Annabel Cohen ou David Stredulinsky, à temps pour parvenir avant le 31 mai 2002.

2. Limitez le résumé à 250 mots, y compris le titre et le nom du premier auteur et son adresse; les noms et adresses des co-auteurs ne sont pas comptés. Les formules qui sont séparées du texte sont comptées comme 40 mots. N'utilisez pas les formes personnelles "je" ou "nous" mais plutôt la forme passive.

3. Utilisez l'exemple présenté à la suite de ces instructions. Le titre du résumé, les noms et adresses des auteurs ne doivent constituer qu'un seul paragraphe et la première ligne doit commencer en retrait. Le résumé tout entier doit être tapé au recto d'une page de format 8 ½ par 11 pouces ou A4.

4. Pour éviter tout retard dans l'acheminement du courrier, assurez-vous que le résumé comporte l'adresse complète de l'auteur qui doit recevoir la notification d'acceptation.

5. N'utilisez aucune note en bas de page. Pour citer les références ou les remerciements, utilisez des crochets [.] (voir l'exemple sur l'autre page).

6. Au bas du résumé, ajoutez les informations suivantes:

(a) Si la communication fait partie d'une session spéciale, précisez la session. Si c'est une communication invitée, écrire "invité".

(b) Spécifiez le domaine de l'acoustique qui correspond le mieux au sujet traité: acoustique architecturale, génie acoustique et contrôle du bruit, acoustique physique et ultrasons, acoustique musicale et électro-acoustique, psycho-acoustique, acoustique physiologique, chocs et vibrations, audiologie, science du langage, acoustique sous-marine, traitement du signal et méthodes numériques, éducation en acoustique ou autre.

(c) Indiquez le numéro de téléphone y compris le code régional de l'auteur qui doit être contacté pour information. Les auteurs non-canadiens doivent indiquer le pays.

(d) S'il y a plus d'un auteur, précisez celui qui doit recevoir la notification d'acceptation.

(e) Des rétro-projecteurs seront disponibles à toutes les sessions. Décrivez tout équipement spécial qui serait nécessaire.

(f) Indiquez votre préférence: présentation d'une conférence ou d'un poster.

[réalisé d'après les directives de la Société Américaine d'Acoustique]

Sample Format for Abstract (250 words maximum)

A survey of new facilities for measurement of x. John S. Doe & Jane S. Bruin (X Ltd., 90 X Ave., Xcity, Province Postal Code).
The measurement of x and the changes in the shape of y during z has been a persistent problem for workers in m industries. In recent years a number of new techniques, described in the present work, have become available for measuring x and y. Included will be a description of the M. Institute of N Super microbeam project .... etc. This concurs with an earlier report of J. S. Smith and J. R. Bourque [Can. J. of N, 14, 31-35, 1999]. It is suggested that because of a, b is the most practical approach to x in most situations [work supported by ZZZ Foundation]

Technical Area (s): Speech communication
This presentation is part of a special session on XYZ organized by ______
Method of Presentation: Prefer lecture but willing to give as poster
Telephone number: (418) 555-7897 Ext. 481 (J. S. Doe)
Fax number: (418) 555-7890
e-mail: jsdoe@xpl-ace.ca
Send acceptance or rejection notice to J. S. Doe
Audio visual requirements: slide projector

Exemple de résumé (250 mots maximum)

Une enquête sur de nouveaux dispositifs de mesure.
Jacques S. Doe & Jeannette S. Bruin (X Ltd., 90 X Ave., Xcity, Province, Code Postal).

La mesure de x et des changements de forme de y pose un problème persistant pour les scientifiques depuis z années. Récemment, de nouvelles techniques sont apparues pour mesurer x et y et cette communication va brièvement en évaluer quelques-unes. Le projet du super rayon microscopique N de l'Institut M sera décrit.......................etc..................................Ceci est en accord avec les conclusions de J. S. Smith and J. R. Bourque dans un rapport paru précédemment [J. Can. de X, 14, 31-35, 1999]. A cause de a, l'approche b est conseillée car c'est la plus pratique pour x dans la plupart des situations. [Recherche financée par l'organisme ZZ].

Cette présentation fait partie d'une session spéciale sur XYZ organisée par ______
Domaine technique: Psychoacoustique.
Numéro de téléphone: (418) 555-7897 Ext. 481 (J. S. Doe)
Fax: (418) 555-7890
e-mail: jsdoe@xpl-ace.ca
Envoyer la notification d'acceptation ou de rejet à: J. S. Doe.
Équipement spécial: projecteur de diapositives 35 mm, etc…
Méthode de présentation: Préfère une conférence mais acceptera de présenter un poster.

Sound Reinforcement Engineering – Fundamentals and Practice by Wolfgang Ahnert and Frank Steffen
Routledge, 412 pages, 1999
ISBN 0-415-23870-6

When I first heard of this book I actually jumped for joy - for two reasons. First, there was a new book on sound system design (this is rare) and what's more, it had the words engineering in the title (distinguishing it from so many other non-technical treatments). I was also excited when I discovered it was co-written by Wolfgang Ahnert, well known in electro-acoustics for his development of one of the most useful computer aided design tools in the sound reinforcement industry - the EASE software. It accurately predicts loudspeaker coverage and has become the standard for electro-acoustic design.

For the purpose of their book, Ahnert and Steffen have divided the subject of sound reinforcement into: Functions of a sound reinforcement system; Room acoustics and auditory psychophysiology; Components for sound reinforcement engineering; Calculations; Systems layout; Calibration and testing; and Case histories. These, and the Introduction, comprise the 8 chapters of the book.

The book was first published in German in 1993 and then in English in 1999, the translation being done by Hans-Joachim Kaminski, an associate of the authors for 15 years. He has ensured that the German and English acoustic terms are matched. The translation was then transformed into modern technical English by Peter Mapp and then copy-edited. Despite all this, the result is a book that still reads like it was written in a far away land and I think this is just great! In the *Introduction* the authors express their views that there are many ways electro-acoustic advancement have allowed a variety of new applications to be provided. The greatest advancement being in the field of sound reinforcement for cultural events.

In *Functions of a sound reinforcement system,* the broad and generalized requirements of sound reinforcement are documented and this sets the basis for the rest of the book. I found the information presented to be interesting and it represents a different emphasis from that found in North America. For example, there was little mention of surround and effects loudspeakers.

Room acoustics and auditory psychophysiology presents an excellent and mathematical treatment of room acoustics specialized to the subject of electro acoustics. Discussions include: reverb time and radius; energy-time curves; intelligibility and clarity; ALcons; RASTI and more. Psychophysiology topics include pitch, loudness, masking, timbre, time behavior, localization, distortion and more. The chapter finishes with a section on the implications of all of these on sound engineering.

Components for sound reinforcement engineering attempts, in 80 pages, to provide relevant information about loudspeakers, microphones, and amplifiers, etc. This is a tall task, and this section is far from complete. Regardless, a great deal of useful information is presented and in some cases in a new light.

Calculations provides the reader with the mathematics and the models to analyze and predict important sound characteristics. These include acoustical parameters such as sound level, critical distance, equivalent acoustic distance and the sound level in a room. Subsections of this chapter present the sound level and dynamic range of the radiated signal, achievable sound level and audio power, acoustic gain and the required frequency transmission range and timbre. This section of the book will be interesting to North American readers due to its unique treatment. The mathematics are thorough.

Systems Layout discusses typical loudspeaker system designs. The sections headings are: Information systems (read "paging over large areas"); Sound reinforcement systems with and without playback reproduction; Sound reinforcement systems for improving room-acoustical parameters; Sound reinforcement systems for ensuring acoustics localization of sound sources in the performance; and Sound reinforcement systems serving as a means of artistic expression.

These topics are further broken down in venue types, such as sports and theatre, and approach such as, central and decentralized (read "distributed"). The ideas are interesting, well presented, and in several cases, great food for thought. There are numerous installation photos.

The section *Calibration and testing* provides a brief overview of the methodology and techniques that are in practice. While there is a lot of useful information provided here, I felt this chapter did not properly address the subject – leaving far too much for the reader to learn the hard way. This chapter is not a complete reference on this subject and is dated as well.

The final chapter is *Case Histories* and discusses a wide variety of installations or all types such as sports venues, performance threatres, concert halls, lecture theatres and more. Some very interesting approaches and techniques are presented along with photos and diagrams yielding a very interesting and worthwhile read.

In conclusion, Ahnert and Steffen have provided a detailed and well thought out discussion of sound reinforcement that will benefit even the most knowledgeable reader. It is particularly enlightening and interesting for North American readers due to its new and unusually rigorous treatment of many topics.

Reviewd by Philip Giddings, P.Eng.
Engineering Harmonics, Toronto, ON; Tel:416-465-3378
e-mail: pgiddings@engineeringharmoics.com

CAA 2001 CONFERENCE SUMMARY / Compte-rendu de la Conférence de l'ACA 2001

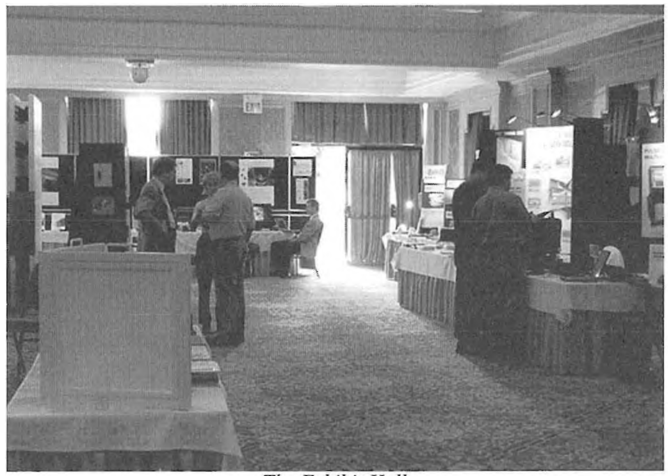
Dalila Giusti, Conference Chair/ Présidente de la Conférence
Jade Acoustics
203-545 N. Rivermede Road, Concord, ON L4K 4H1
jade_acoustics@compuserve.com

By now the CAA 2001 Conference is a distance memory for most people. We hope that it is a pleasant memory for all who attended. For me, the organizing and execution of the Conference was both challenging and rewarding.

The Conference was held at the beautiful Nottawasaga Inn, in Alliston, Ontario. The format chosen for this Conference was different from other conferences in that daily registration was offered in addition to the full 3 day conference registration. This was done to encourage the local CAA members to attend. This proved to be successful. A large number of members and student members attended and participated in the Conference. In addition to attending the Conference many of the students were active in acoustics throughout the year as is evidenced by the fact that most of the available CAA prizes were awarded this year. We wish to congratulate all prize recipients.

The 2001 Annual General Meeting
Executive - (l to r) Trevor Nightingale, John Bradley and Dalila Guisti

The 2001 Conference was interesting and diverse with over 80 papers and discussion sessions. The plenary speakers, Dr. Blake Papsin and Mr. Bill McMurray presented discussions that were both riveting and informative.



The Exhibit Hall

Ms. Atarah Ben-Tovim provided the entertainment at our banquet. She and Mr. Bill Gastmeier (accompanist) had the CAA delegates dancing, singing, playing musical instruments and laughing. A special thank you to our surprise entertainer, "Elvis" (better know as Mr. John Swallow) for his beautiful singing. The banquet was truly a night to remember.

We were fortunate to have many sponsors for the coffee breaks, wine for the banquet and tote bags. In addition we had 23 exhibitors. All this was accomplished through the hard work of the organizing committee.

I have enjoyed meeting the challenges presented to us in the course of my duties as the Conference Chair. By all accounts the 2001 CAA Conference was a success. Thank you to all the people who were involved in putting this Conference together, to all the people who assisted at the Conference and especially to those who attended and participated in the Conference. We look forward to the 2002 CAA Conference in Charlottetown, PEI!

La Conférence de l'ACA 2001 est à présent un souvenir lointain pour la plupart d'entre vous. Nous espérons qu'il est agréable à tous ceux qui y ont assisté. Pour ma part, l'organisation et le bon déroulement de la conférence ont représenté un défi ainsi qu'une expérience gratifiante.

La conférence s'est tenue dans le cadre magnifique de l'auberge Nottawasaga en Alliston, Ontario. Cette année, en plus de l'inscription complète, nous proposions une inscription à la journée afin d'encourager la venue des membres de l'ACA de la région de Toronto. Ce fut un véritable succès. Un nombre important de membres et d'étudiants ont assisté et participé. De plus, la plupart des prix de l'ACA ont été attribués cette année, ce qui témoigne du dynamisme des étudiants dans le domaine de l'acoustique. Nous tenons ici à féliciter tous les lauréats.

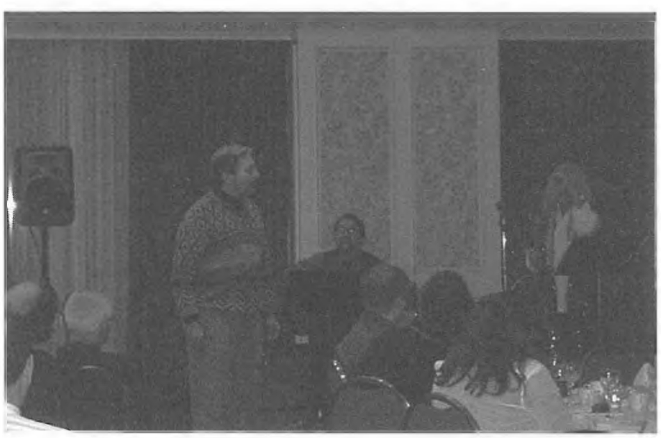
La Conférence 2001 a été intéressante et variée avec plus de 80 contributions et discussions. Les lectures plénières présentées par le Dr Blake Papsin et M. Bill Mc Murray ont été attrayantes et instructives.

Mlle Atarah Ben-Tovim nous a amusé pendant le banquet. Accompagnée par M. Bill Gastmesiter, elle a fait danser, chanter, rire et jouer sur des instruments de musique tous les convives. Nos remerciements vont particulièrement à notre humoriste surprise, "Elvis" (plus connu sous le nom de John Swallow) pour sa merveilleuse maitrise du chant. La soirée du banquet a vraiment été un moment inoubliable.

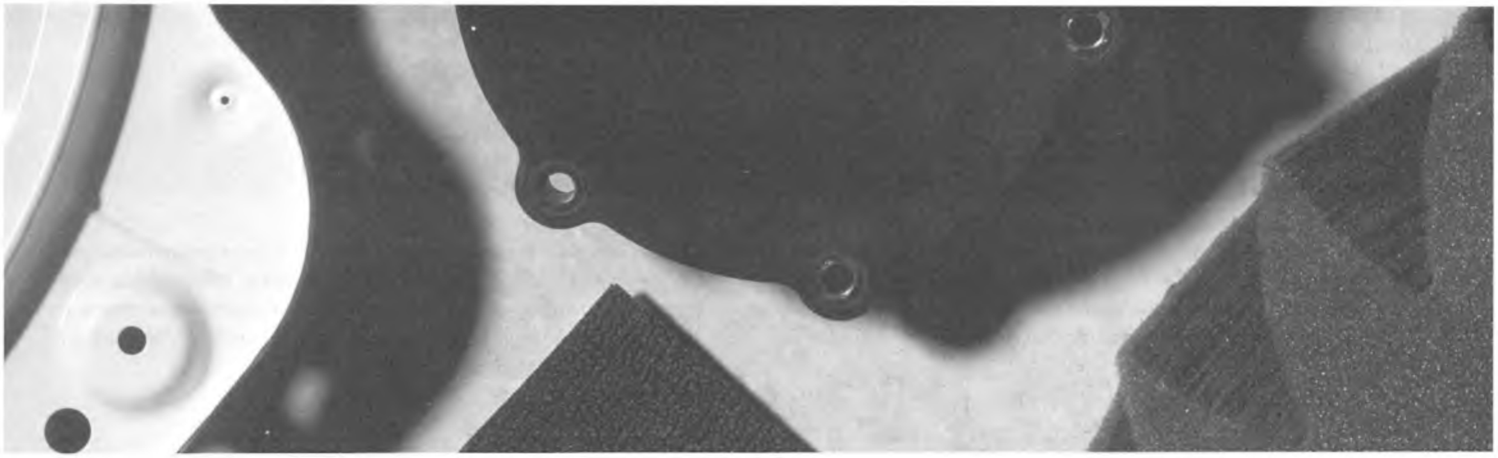
Technical Session

Nous avons eu la chance que de nombreux commanditaires se manifestent pour les pauses-café, le vin du banquet et les fourre-tout. Il y'a eu de plus 23 exposants. Ces résultats ont été possible grâce à la persévérance du comité organisateur.

Au cours de mes fonctions de Présidente de la Conférence, c'est avec plaisir que j'ai relevé les défis qui se sont présentés à nous. A tout point de vue, l'édition 2001 de la Semaine d'Acoustique Canadienne a été un succès. Merci à tous ceux qui se sont investis dans l'organisation et surtout, à tous ceux qui ont assisté et participé. Nous espérons vous revoir à la Conférence de l'ACA 2002, à Charlottetown à l'Ile du Prince Edouard.



Banquet Evening: The artistes are:
(l to r) John Swallow, Bill Gastmeier and Atarah Ben-Tovim



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NEWS / INFORMATIONS

CONFERENCES

The following list of conferences was mainly provided by the Acoustical Society of America. If you have any news to share with us, send them by mail or fax to the News Editor (see address on the inside cover), or via electronic mail to desharnais@drea.dnd.ca

2002

4-8 March: German Acoustical Society Meeting (DAGA 2002), Bochum, Germany. Contact: J. Blauert, Institute of Communication Acoustics, Ruhr-Universität Bochum, 44780 Bochum, Germany; Fax: +49 234 321 4165; Web: www.ika.ruhr-uni-bochum.de

10-13 March: Annual Meeting of American Institute of Ultrasound in Medicine, Nashville, TN. Contact: American Institute of Ultrasound in Medicine, 14750 Sweitzer Lane, Suite 100, Laurel, MD 20707-5906; Tel.: 301-498-4100 or 800-638-5352; Fax: 301-498-4450; E-mail: conv_edu@aium.org; Web: www.aium.org

18-20 March: Spring Meeting of the Acoustical Society of Japan, Kanagawa, Japan. Contact: Acoustical Society of Japan, Nakaura 5th-Bldg., 2-18-20 Sotokanda, Chiyoda-ku, Tokyo 101-0021, Japan; Fax: +81 3 5256 1022; Web: wwwsoc.nacsis.ac.jp/asj/

25-27 March: Institute of Acoustics Spring Conference, Manchester, UK. Contact: IoA, 77A St Peter's Street, St. Albans, Herts AL1 3BN, UK; Fax: +44 1727 850553; Web: www.ioa.org.uk

8-11 April: 6th Congress of the French Acoustical Society, joint with the Belgian Acoustical Society, Lille, France. Contact: Société française d'acoustique, 23 av. Brunetière, 75017 Paris, France. Web: www.isen.fr/cfa2002

22-24 April: International Meeting on Acoustic Pollution in Cities, Madrid, Spain. Contact: Viajes, Princesa 47-4a Planta, 28008 Madrid, Spain; Fax: +34 1 559 74 11; E-mail: dccimad8@viajeseci.es

27-30 May: Joint Meeting: Russian Acoustical Society and Conference on Ocean Acoustics, Moscow, Russia. Contact: Yu. A. Chepurin, P.P. Shirshov Institute of Oceanology, Russian Academy of Sciences, Nakhimovsky Prospekt 36, 117851 Moscow, Russia; Fax: +7 095 124 5983; Web: rav.sio.rssi.ru/Ixconf.html

30 May – 1 June: 2nd International Conference on Newborn Hearing Screening, Diagnosis, and Intervention, Como, Italy. Contact: D. Hayes, The Children's Hospital, 1056 East 19th Ave. B030, Denver, CO 80218 USA; Fax: +1 303 764 8220; Web: www.biomed.polimi.it/nhs2002

3-7 June: 143rd Meeting of the Acoustical Society of America, Pittsburg, PA. Contact: Acoustical Society of America, Suite 1NO1, 2 Huntington Quadrangle, Melville, NY 11747-4502; Tel: 516-576-2360; Fax: 516-576-2377; E-mail: asa@aip.org; Web: asa.aip.org

CONFÉRENCES

La liste de conférences ci-jointe a été offerte en majeure partie par l'Acoustical Society of America. Si vous avez des nouvelles à nous communiquer, envoyez-les par courrier ou fax (coordonnées incluses à l'envers de la page couverture), ou par courrier électronique à desharnais@drea.dnd.ca

2002

4-8 mars: Rencontre de la Société allemande d'acoustique (DAGA 2002), Bochum, Allemagne. Info: J. Blauert, Institute of Communication Acoustics, Ruhr-Universität Bochum, 44780 Bochum, Germany; Fax: +49 234 321 4165; Web: www.ika.ruhr-uni-bochum.de

10-13 mars: Rencontre annuelle de l'Institut américain des ultrasons en médecine, Nashville, TN. Info: American Institute of Ultrasound in Medicine, 14750 Sweitzer Lane, Suite 100, Laurel, MD 20707-5906; Tél.: 301-498-4100 ou 800-638-5352; Fax: 301-498-4450; Courriel: conv_edu@aium.org; Web: www.aium.org

18-20 mars: Rencontre de printemps de la Société japonaise d'acoustique, Kanagawa, Japon. Info: Acoustical Society of Japan, Nakaura 5th-Bldg., 2-18-20 Sotokanda, Chiyoda-ku, Tokyo 101-0021, Japan; Fax: +81 3 5256 1022; Web: wwwsoc.nacsis.ac.jp/asj/

25-27 mars: Conférence de printemps de l'Institut d'acoustique, Manchester, Royaume-Uni. Info: IoA, 77A St Peter's Street, St. Albans, Herts AL1 3BN, UK; Fax: +44 1727 850553; Web: www.ioa.org.uk

8-11 avril: 6e Congrès combiné de la Société française d'acoustique et de la Société belge d'acoustique, Lille, France. Info: Société française d'acoustique, 23 av. Brunetière, 75017 Paris, France. Web: www.isen.fr/cfa2002

22-24 avril: Rencontre internationale sur la pollution acoustique dans les villes, Madrid, Espagne. Info: Viajes, Princesa 47-4a Planta, 28008 Madrid, Spain; Fax: +34 1 559 74 11; Courriel: dccimad8@viajeseci.es

27-30 mai: Rencontre combinée: Société russe d'acoustique, et Conférence sur l'acoustique océanique, Moscou, Russie. Info: Yu. A. Chepurin, P.P. Shirshov Institute of Oceanology, Russian Academy of Sciences, Nakhimovsky Prospekt 36, 117851 Moscow, Russia; Fax: +7 095 124 5983; Web: rav.sio.rssi.ru/Ixconf.html

30 mai – 1 juin: 2e conférence internationale sur le dépistage de l'audition des nouveaux-nés, diagnostique et intervention, Como, Italie. Info: D. Hayes, The Children's Hospital, 1056 East 19th Ave. B030, Denver, CO 80218 USA; Fax: +1 303 764 8220; Web: www.biomed.polimi.it/nhs2002

3-7 juin: 143e rencontre de l'Acoustical Society of America, Pittsburg, PA. Info: Acoustical Society of America, Suite 1NO1, 2 Huntington Quadrangle, Melville, NY 11747-4502; Tél: 516-576-2360; Fax: 516-576-2377; Courriel: asa@aip.org; Web: asa.aip.org

4-6 June: 6th International Symposium on Transport Noise and Vibration, St. Petersburg, Russia. Contact: East-European Acoustical Association, Moskovskoe Shosse 44, St. Petersburg 196158, Russia; Fax: +7 812 127 9323; E-mail: noise@mail.rcom.ru

10-14 June: Acoustics in Fisheries and Aquatic Ecology, Montpellier, France. Contact: D.V. Holliday, BAE SYSTEMS, 4669 Murphy Canyon Road, Suite 102, San Diego, CA 92123-4333, USA; Web: www.ices.dk/symposia/

24-27 June: 6th European Conference on Underwater Acoustics, Gdansk, Poland. Fax: +48 58 347 1535; Web: www.ecua2002.gda.pl/

24-28 June: 11th Symposium of the International Society for Acoustic Remote Sensing, Rome, Italy. Contact: +39 06 20660291; Web: ISARS2002.ifa.rm.cnr.it/

3-6 July: International Conference on Mechatronics (MECH2K2) jointly with the 3rd International Symposium on Acoustics, Noise and Vibrations (ICANOV3), Linz, Austria. Contact: WBJ Zimmerman, University of Sheffield, W.Zimmerman@shef.ac.uk; Web: eyrie.shef.ac.uk/mech2k2

15-17 July: ACTIVE 2002 — 2002 International Symposium on Active Control of Sound and Vibration, Southampton, UK. Contact: Stephen J. Elliott, Institute of Sound and Vibration Research, Southampton University, University Road, Highfield, Southampton SO17 1BJ, United Kingdom; Tel.: +44 23 8059 2384; Fax: +44 23 8059 3190; E-mail: sje@isvr.soton.ac.uk; Web: www.isvr.soton.ac.uk/ACTIVE2002

19-21 August: Inter-Noise 2001 - 31st International Congress and Exposition on Noise Control Engineering, Dearborn, MI. Contact: Inter-Noise 2002 Congress Secretariat, Dept. Mechanical Engineering, Ohio State University, 206 West 18th Avenue, Columbus, OH 43210-1107, USA. E-mail: peersen.1@osu.edu; Web: www.internoise2002.org

19-23 August: 16th International Symposium on Nonlinear Acoustics (ISNA16), Moscow, Russia. Contact: O. Rudenko, Physics Department, Moscow State University, 119899 Moscow, Russia; E-mail: isna@acs366b.phys.msu.su

26-28 August: 2nd Biot Conference on Poromechanics, Grenoble, France. Contact: J.-L. Auriault, Laboratoire 3S, Domaine Universitaire, BP53, 38041 Grenoble, France. Fax: +33 4 76 82 70 43; Web: geo.hmg.inpg.fr/biot2002

26-28 August: Joint Baltic-Nordic Acoustical Meeting 2002, Lyngby, Denmark. Fax: +45 45 88 05 77; Web: www.dat.dtu.dk

11-13 September: 10th International Meeting on Low Frequency Noise and Vibration, York, UK. Contact: W. Tempest, Multi-Science Co. Ltd., 5 Wates Way, Brentwood, Essex CM15 9TB, UK; Fax: +44 1277 223 453; Web: www.lowfrequency2002.org.uk

4-6 juin: 6e Symposium international sur le bruit et vibrations des transports, Saint-Pétersbourg, Russie. Info: East-European Acoustical Association, Moskovskoe Shosse 44, St. Petersburg 196158, Russia; Fax: +7 812 127 9323; Courriel: noise@mail.rcom.ru

10-14 juin: Acoustique des pêches et écologie aquatique, Montpellier, France. Info: D.V. Holliday, BAE SYSTEMS, 4669 Murphy Canyon Road, Suite 102, San Diego, CA 92123-4333, USA; Web: www.ices.dk/symposia/

24-27 juin: 6e conférence européenne sur l'acoustique sous-marine, Gdansk, Pologne. Fax: +48 58 347 1535; Web: www.ecua2002.gda.pl/

24-28 juin: 11e symposium sur la Société internationale de la télédétection acoustique, Rome, Italie. Info: +39 06 20660291; Web: ISARS2002.ifa.rm.cnr.it/

3-6 juillet: Conférence internationale sur la mécatronique (MECH2K2) combinée avec le 3e Symposium international sur l'acoustique, le bruit et les vibrations (ICANOV3), Linz, Autriche. Info: WBJ Zimmerman, University of Sheffield, W.Zimmerman@shef.ac.uk; Web: eyrie.shef.ac.uk/mech2k2

15-17 juillet: ACTIVE 2002 — Symposium international 2002 sur le contrôle actif du bruit et des vibrations, Southampton, Royaume-Uni. Info: Stephen J. Elliott, Institute of Sound and Vibration Research, Southampton University, University Road, Highfield, Southampton SO17 1BJ, United Kingdom; Tél.: +44 23 8059 2384; Fax: +44 23 8059 3190; Courriel: sje@isvr.soton.ac.uk; Web: www.isvr.soton.ac.uk/ACTIVE2002

19-21 août: Inter-Noise 2001 — 31e Congrès international et exposition sur le génie du contrôle du bruit, Dearborn, MI. Info: Inter-Noise 2002 Congress Secretariat, Dept. Mechanical Engineering, Ohio State University, 206 West 18th Avenue, Columbus, OH 43210-1107, USA. Courriel: peersen.1@osu.edu; Web: www.internoise2002.org

19-23 août: 16e Symposium international sur l'acoustique non-linéaire (ISNA16), Moscou, Russie. Info: O. Rudenko, Physics Department, Moscow State University, 119899 Moscow, Russia; Courriel: isna@acs366b.phys.msu.su

26-28 août: 2e Conférence de Biot sur la Poro-mécanique, Grenoble, France. Info: J.-L. Auriault, Laboratoire 3S, Domaine Universitaire, BP53, 38041 Grenoble, France. Fax: +33 4 76 82 70 43; Web: geo.hmg.inpg.fr/biot2002

26-28 août: Rencontre acoustique baltique-nordique combinée 2002, Lyngby, Danemark. Fax: +45 45 88 05 77; Web: www.dat.dtu.dk

11-13 septembre: 10e rencontre internationale sur le bruit et les vibrations à basse fréquence, York, Royaume-Uni. Info: W. Tempest, Multi-Science Co. Ltd., 5 Wates Way, Brentwood, Essex CM15 9TB, UK; Fax: +44 1277 223 453; Web: www.lowfrequency2002.org.uk

16-21 September: Forum Acusticum 2002 (Joint EAA-SEA-ASJ Meeting), Sevilla. Fax: +34 91 411 7651; Web: www.cica.es/aliens/forum2002

26-28 September: Autumn Meeting of the Acoustical Society of Japan, Akita, Japan. Contact: Acoustical Society of Japan, Nakaura 5th-Bldg., 2-18-20 Sotokanda, Chiyoda-ku, Tokyo 101-0021, Japan; Fax: +81 3 5256 1022; Web: wwwsoc.nacsis.ac.jp/asj/

2-6 December: Joint Meeting: 9th Mexican Congress on Acoustics, 144th Meeting of the Acoustical Society of America, and 3rd Iberoamerican Congress on Acoustics, Cancun, Mexico. Contact: Acoustical Society of America, Suite 1NO1, 2 Huntington Quadrangle, Melville, NY 11747-4502; Tel: 516-576-2360; Fax: 516-576-2377; E-mail: asa@aip.org; Web: asa.aip.org/cancun.html

2003

7-9 April: WESPAC8, Melbourne, Australia. Web: www.wespac8.com

28 April – 2 May: 145th Meeting of the Acoustical Society of America, Nashville, TN. Contact: Acoustical Society of America, Suite 1NO1, 2 Huntington Quadrangle, Melville, NY 11747-4502; Tel: 516-576-2360; Fax: 516-576-2377; E-mail: asa@aip.org; Web: asa.aip.org

8-13 June: XVIII International Evoked Response Audiometry Study Group Symposium, Puerto de la Cruz, Tenerife, Canary Islands, Spain. Fax: +34 922 27 03 64; Web: www.ierasg-2003.org

7-10 July: 10th International Congress on Sound and Vibration, Stockholm, Sweden. Contact: Congress Secretariat, Congrex Sweden AB; Tel: +46 8 459 66 00; Fax: +46 8 8 661 91 25; E-mail: icsv10@congrex.se; Web: www.congex.com/icsv10

14-16 July: 8th International Conference on Recent Advances in Structural Dynamics, Southampton, UK. Web: www.isvr.soton.ac.uk/sd2003

1-4 September: Eurospeech 2003, Geneva, Switzerland. Contact: SYMPORG SA, Avenue Krieg 7, 1208 Geneva, Switzerland; Fax: +41 22 839 8485; Web: www.symporg.ch/eurospeech2003

10-24 November: 146th Meeting of the Acoustical Society of America, Austin, TX. Contact: Acoustical Society of America, Suite 1NO1, 2 Huntington Quadrangle, Melville, NY 11747-4502; Tel: 516-576-2360; Fax: 516-576-2377; E-mail: asa@aip.org; Web: asa.aip.org

2004

5-9 April: 18th International Congress on Acoustics (ICA2004), Kyoto, Japan. Web: ica2004.or.jp

16-21 septembre: Forum Acusticum 2002 (Rencontre conjointe EAA-SEA-ASJ), Séville. Fax: +34 91 411 7651; Web: www.cica.es/aliens/forum2002

26-28 septembre: Rencontre d'automne de la Société japonaise d'acoustique, Akita, Japon. Info: Acoustical Society of Japan, Nakaura 5th-Bldg., 2-18-20 Sotokanda, Chiyoda-ku, Tokyo 101-0021, Japan; Fax: +81 3 5256 1022; Web: wwwsoc.nacsis.ac.jp/asj/

2-6 décembre: Rencontres combinées: 9e Congrès mexicain d'acoustique, 144e rencontre de l'Acoustical Society of America, et 3e Congrès ibéro-américain d'acoustique, Cancun, Mexique. Info: Acoustical Society of America, Suite 1NO1, 2 Huntington Quadrangle, Melville, NY 11747-4502; Tél: 516-576-2360; Fax: 516-576-2377; Courriel: asa@aip.org; Web: asa.aip.org/cancun.html

2003

7-9 avril: WESPAC8, Melbourne, Australie. Web: www.wespac8.com

28 avril – 2 mai: 145e rencontre de l'Acoustical Society of America, Nashville, TN. Info: Acoustical Society of America, Suite 1NO1, 2 Huntington Quadrangle, Melville, NY 11747-4502; Tél.: 516-576-2360; Fax: 516-576-2377; Courriel: asa@aip.org; Web: asa.aip.org

8-13 juin: XVII Symposium international du Groupe expérimental sur l'audiométrie des potentiels évoqués, Puerto de la Cruz, Tenerife, Iles Canaries, Espagne. Fax: +34 922 27 03 64; Web: www.ierasg-2003.org

7-10 juillet: 10e Congrès international sur le bruit et les vibrations, Stockholm, Suède. Info: Congress Secretariat, Congrex Sweden AB; Tél.: +46 8 459 66 00; Fax: +46 8 8 661 91 25; Courriel: icsv10@congrex.se; Web: www.congex.com/icsv10

14-16 juillet: 8e Conférence internationale sur les développements récents en dynamique structurelle, Southampton, Royaume-Uni. Web: www.isvr.soton.ac.uk/sd2003

1-4 septembre: Eurospeech 2003, Genève, Suisse. Info: SYMPORG SA, Avenue Krieg 7, 1208 Geneva, Switzerland; Fax: +41 22 839 8485; Web: www.symporg.ch/eurospeech2003

10-24 novembre: 146e rencontre de l'Acoustical Society of America, Austin, TX. Info: Acoustical Society of America, Suite 1NO1, 2 Huntington Quadrangle, Melville, NY 11747-4502; Tél.: 516-576-2360; Fax: 516-576-2377; Courriel: asa@aip.org; Web: asa.aip.org

2004

5-9 avril: 18e Congrès international sur l'acoustique (ICA2004), Kyoto, Japon. Web: ica2004.or.jp

Museum Seeks Nominations for Canadian Science and Engineering Hall of Fame

The Canada Science and Technology Museum invites nominations for this year's inductees to the Canadian Science and Engineering Hall of Fame. Inductees will be chosen by a Selection Committee representing a diversity of science and engineering institutions, and will be honoured at a special induction ceremony this November in Ottawa.

The Canadian Science and Engineering Hall of Fame seeks to promote Canadian achievements and careers in science and engineering. Nominees are required to have contributed in an exceptional way to the advancement of science and engineering in Canada; their work must have brought great benefits to society and their communities; and they must possess leadership qualities that can serve as an inspiration to young Canadians to pursue science, engineering or technology careers.

Nominations should state, in 500 words or less, the supporting arguments for an individual's induction into the Hall of Fame. Nominations can be submitted by individuals or on behalf of an organization, and individuals nominated in previous years can be nominated again. The deadline for this year's nominations is April 1, 2002.

The two newest members of the Canadian Science and Engineering Hall of Fame, inducted in November 2001, are pioneering ecologist Pierre Dansereau and accomplished medical geneticist, Charles Scriver. With these two new members, there are now 28 Canadian scientists and innovators honoured in the Hall of Fame, including Maude Abbot, Sir Frederick Banting, Joseph-Armand Bombardier and Elizabeth MacGill. The Hall of Fame is located at the Canada Science and Technology Museum in Ottawa, and in a virtual format at the museum's website, www.nmstc.ca

For more detailed information about submitting a nomination, visit the Hall of Fame pages of the Canada Science and Technology Museum's website at www.nmstc.ca, e-mail: cts@nmstc.ca or phone (613) 991-3044.

Appel de mises en candidature pour le Panthéon de la science et de l'ingénierie canadiennes

Le Musée des sciences et de la technologie du Canada invite la présentation de candidatures pour les nouveaux membres du Panthéon de la science et de l'ingénierie canadiennes. Les nouveaux members seront choisis par un comité de sélection représentant diverses institutions du domaine des sciences et de l'ingénierie, et seront honorés dans une cérémonie qui prendra place à Ottawa en Novembre.

Le Panthéon de la science et de l'ingénierie canadiennes cherche à promouvoir les réalisations canadiennes et les carrières en sciences et en ingénierie. Les candidats doivent avoir contribué de façon exceptionnelle à l'avancement des sciences et du génie au Canada; leur travail doit avoir grandement bénéficié à la société en général et à l'ensemble de leur collectivité; ils doivent posséder des qualités de chef, susceptibles d'inspirer aux jeunes du Canada le goût de poursuivre des carrières en science, en génie ou en technologie.

Les nominations doivent présenter, en 500 mots ou moins, un argument indiquant pourquoi le candidat est apte à devenir membre du Panthéon. Des nominations peuvent être proposer de la part des individus ou des organisations, et des individus qui ont été nominé dans les années précédentes peuvent être nominés encore. Le date d'échéance pour recevoir les nominations pour cette année est le 1er avril 2002.

Les deux dernièrs membres du Panthéon de la science et de l'ingénierie canadiennes, qui ont été intronisés en Novembre 2001, sont Pierre Dansereau, pionnier en science de l'écologie, et Charles Scriver, médecin-généticien accompli, Charles Scriver. Avec ces deux nouveaux membres, le Panthéon compte maintenant 28 scientifiques et innovateurs canadiens parmi ses membres, dont Maude Abbot, Sir Frederick Banting, Joseph-Armand Bombardier et Elizabeth MacGill. Le Panthéon se trouve au Musée des sciences et de la technologie du Canada à Ottawa, et dans un format virtuel au site web du musée à www.sciences-tech.smnst.ca

Pour plus de détails sur le processus de la présentation des candidatures, visitez les pages du Panthéon de la science et de l'ingénierie canadiennes sur la site web du Musée des sciences et de la technologie du Canada à www.sciences-tech.smnst.ca, ou contactez le Musée par courrier électronique à cts@nmstc.ca ou téléphonez au (613) 991-3044.

INCE/USA Publishes the NOISE-CON 01 Proceedings on CD-ROM

NOISE-CON 01, 2001 National Conference on Noise Control Engineering, was held in Portland, Maine, USA on October 29-31, 2001. The proceedings of the conference have been published on a CD-ROM together with the proceedings of five previous conferences, several other technical reports, the tables of content of all previous NOISE-CON proceedings, and a sample of audio signals useful for analysis or demonstration.

The CD-ROM includes the NOISE-CON proceedings for 1996, 1997, 1998, 2000 and 2001. The proceedings of SQS 98, the 1998 Sound Quality Symposium, are also included. More than 600 papers on noise, all in Portable Document Format (PDF), are included on the CD-ROM. The CD-ROM is searchable by key word or by a string of words, and is used with the freely-available Adobe Acrobat reader on Windows, Macintosh, or Unix platforms.

Three technical reports prepared by the International Institute of Noise Control Engineering have been published in the magazine *Noise/News International*, and have been included on the CD-ROM. The titles of these reports are *Noise Emissions of Road Vehicles — Effect of Regulations* (2001), *Technical Assessment of the Effectiveness of Noise Walls* (1999), and *Technical Assessment of Upper Limits on Noise in the Workplace* (1997). The reports, prepared by international teams, review the state of technology in these three areas, and contain many references.

The CD-ROM, stock number CD-NC01, is available for 70 US$, shipped by first class mail in the United States and by air mail to other countries, and may be ordered by postal mail, telephone (USA and Canada only), by FAX, or by e-mail. Major credit cards are accepted. Postal mail: Bookmasters, Inc., Distribution Services Division, 30 Amberwood Parkway, Ashland, OH 44805, USA. Telephone: 1800 247 6553; FAX: 1 419 281 6883; e-mail: order@bookmaster.com.

INSTRUCTIONS TO AUTHORS FOR THE PREPARATION OF MANUSCRIPTS

Submissions: The original manuscript and two copies should be sent to the Editor-in-Chief.

General Presentation: Papers should be submitted in camera-ready format. Paper size 8.5" x 11". If you have access to a word processor, copy as closely as possible the format of the articles in Canadian Acoustics 18(4) 1990. All text in Times-Roman 10 pt font, with single (12 pt) spacing. Main body of text in two columns separated by 0.25". One line space between paragraphs.

Margins: Top - title page: 1.25"; other pages, 0.75"; bottom, 1" minimum; sides, 0.75".

Title: Bold, 14 pt with 14 pt spacing, upper case, centered.

Authors/addresses: Names and full mailing addresses, 10 pt with single (12 pt) spacing, upper and lower case, centered. Names in bold text.

Abstracts: English and French versions. Headings, 12 pt bold, upper case, centered. Indent text 0.5" on both sides.

Headings: Headings to be in 12 pt bold, Times-Roman font. Number at the left margin and indent text 0.5". Main headings, numbered as 1, 2, 3, ... to be in upper case. Sub-headings numbered as 1.1, 1.2, 1.3, ... in upper and lower case. Sub-sub-headings not numbered, in upper and lower case, underlined.

Equations: Minimize. Place in text if short. Numbered.

Figures/Tables: Keep small. Insert in text at top or bottom of page. Name as "Figure 1, 2, ..." Caption in 9 pt with single (12 pt) spacing. Leave 0.5" between text.

Line Widths: Line widths in techincal drawings, figures and tables should be a minimum of 0.5 pt.

Photographs: Submit original glossy, black and white photograph.

Scans: Should be between 225 dpi and 300 dpi. Scan: Line art as bitmap tiffs; Black and white as grayscale tiffs and colour as CMYK tiffs;

References: Cite in text and list at end in any consistent format, 9 pt with single (12 pt) spacing.

Page numbers: In light pencil at the bottom of each page.

Reprints: Can be ordered at time of acceptance of paper.

DIRECTIVES A L'INTENTION DES AUTEURS PREPARATION DES MANUSCRITS

Soumissions: Le manuscrit original ainsi que deux copies doivent être soumis au rédacteur-en-chef.

Présentation générale: Le manuscript doit comprendre le collage. Dimensions des pages, 8.5" x 11". Si vous avez accès à un système de traitement de texte, dans la mesure du possible, suivre le format des articles dans l'Acoustique Canadienne 18(4) 1990. Tout le texte doit être en caractères Times-Roman, 10 pt et à simple (12 pt) interligne. Le texte principal doit être en deux colonnes séparées d'un espace de 0.25". Les paragraphes sont séparés d'un espace d'une ligne.

Marges: Dans le haut - page titre, 1.25"; autres pages, 0.75"; dans le bas, 1" minimum; latérales, 0.75".

Titre du manuscrit: 14 pt à 14 pt interligne, lettres majuscules, caractères gras. Centré.

Auteurs/adresses: Noms et adresses postales. Lettres majuscules et minuscules, 10 pt à simple (12 pt) interligne. Centré. Les noms doivent être en caractères gras.

Sommaire: En versions anglaise et française. Titre en 12 pt, lettres majuscules, caractères gras, centré. Paragraphe 0.5" en alinéa de la marge, des 2 cotés.

Titres des sections: Tous en caractères gras, 12 pt, Times-Roman. Premiers titres: numéroter 1, 2, 3, ..., en lettres majuscules; sous-titres: numéroter 1.1, 1.2, 1.3, ..., en lettres majuscules et minuscules; sous-sous-titres: ne pas numéroter, en lettres majuscules et minuscules et soulignés.

Equations: Les minimiser. Les insérer dans le texte si elles sont courtes. Les numéroter.

Figures/Tableaux: De petites tailles. Les insérer dans le texte dans le haut ou dans le bas de la page. Les nommer "Figure 1, 2, 3,..." Légende en 9 pt à simple (12 pt) interligne. Laisser un espace de 0.5" entre le texte.

Largeur Des Traits: La largeur des traits sur les schémas technique doivent être au minimum de 0.5 pt pour permettre une bonne reproduction.

Photographies: Soumettre la photographie originale sur papier glacé, noir et blanc.

Figures Scanées: Doivent être au minimum de 225 dpi et au maximum de 300 dpi. Les schémas doivent être scannés en bitmaps tif format. Les photos noir et blanc doivent être scannées en échelle de gris tifs et toutes les phoots couleurs doivent être scannées en CMYK tifs.

Références: Les citer dans le texte et en faire la liste à la fin du document, en format uniforme, 9 pt à simple (12 pt) interligne.

Pagination: Au crayon pâle, au bas de chaque page.

Tirés-à-part: Ils peuvent être commandés au moment de l'acceptation du manuscrit.

The Canadian Acoustical Association — **l'Association Canadienne d'Acoustique**

SUBSCRIPTION INVOICE

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