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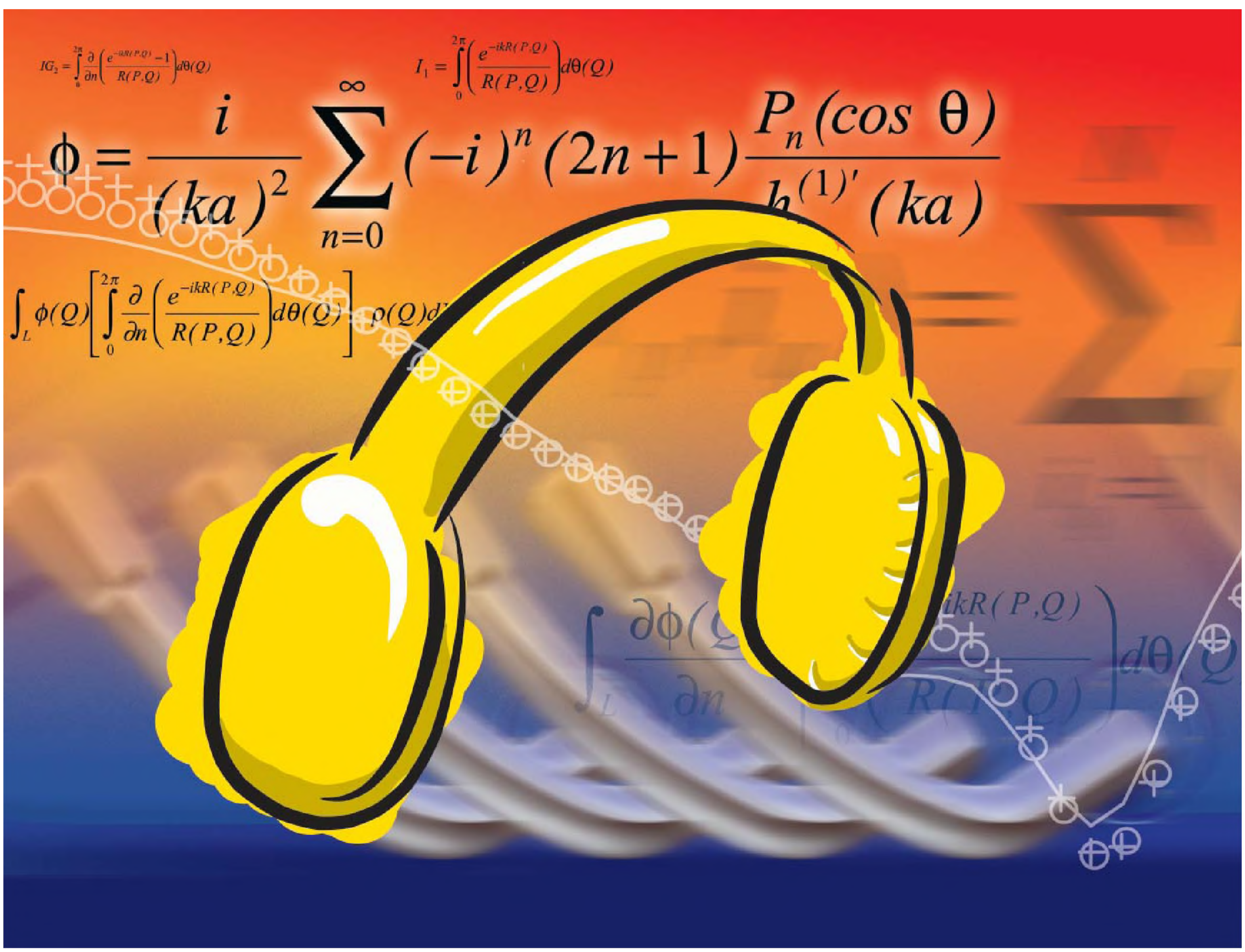

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EDITORIAL / EDITORIAL

Now it is my turn to address our members. Our new president, Prof. Stan Dosso began his tenure (see his Address in the December 2003 issue of the Journal) with an appreciation of the tasks performed by the Executive and the Board. He also wondered how I manage to get the cover art for each issue of the journal. Credit for the cover art goes to our graphic designer, Simon Tuckett of Toronto and thanks also are due to Prof. Murray Hodgson, the former Editor-in-Chief of the journal for discovering Simon. The process for the cover art is as follows. After the line-up of the article is finalized, the abstract of the first article, usually a research paper, is passed on to Simon. I discuss the various acoustic terms and the intent of the paper with him. He produces a draft design and more than 95% of the time, the design is hardly modified by me, thanks to Simon's clear understanding of the paper's intent. So, now you know!!

I'd like to echo the sentiments of Stan Dosso and convey my appreciation to the team that forms the Canadian Acoustical Association. I had the good fortune to work with John Bradley. He was kind and had a subtle way of correcting errors one commits under deadline pressures. He was always the first one to convey his congratulations after the arrival of each issue. He appreciated my commitment to releasing each issue of the journal within a preset time frame. Now that he has become past president, I would like to express my thanks to him for patiently bearing with my many idiosyncrasies.

We have begun two new processes in the journal. It has been a goal to provide pre-prints to authors, but the process was cumbersome and the old printers were demanding high charges. Now that we have found a local printer, the costs are reasonable with the aid of digital printing. So, we have decided to offer this service to our authors. The cost is as follows: Cd$60 for 50 copies and Cd$120 for 100 copies plus postage. The sample postage cost for Los Angeles by Express Post is Cd$25 for 50 copies and Cd$40 for 100 copies. The authors can order pre-prints by contacting the Editor and once the total amount is received, the pre-prints will be shipped to the authors. This service is offered to the articles published in Canadian Acoustics since 1999 (Volume 29 onwards). I'll soon place a pdf version of sample pre-print on our website.

Finally, an exciting development is being attempted with the June 2004 issue of the Journal. If you recall, one of the goals of the journal is to publicize and provide a forum for the acoustical activities in Canada. Recently, in November 2003, DRDC of Halifax conducted a workshop, titled, "2003 Workshop on detection and localization of marine mammals using passive acoustics," under the able chairmanship of our former News Editor, Francine Desharnais. Many of the papers presented at that workshop were reviewed and we decided

Maintenant c'est à moi de s'adresser aux membres. Notre nouveau président, Prof. Stan Dosso a entamé ses fonctions (voir son adresse dans l'édition de décembre 2003 du journal) avec une appréciation des taches exécutées par le bureau exécutif. Il s'est aussi demandé comment je m'organise pour obtenir le graphique de la couverture de chaque édition du journal. Les remerciements devront aller à Simon Tuckett notre designer graphique de Toronto et au Prof. Murray Hodgson, l'ancien rédacteur en chef pour avoir découvert Simon. Le processus du design graphique est comme suit: le résumé du premier article publié dans le journal, qui est généralement un article de recherche scientifique, est envoyé à Simon une fois le formatage de l'article est complété. Par la suite j'entame des discussions avec Simon sur l'objet de la publication ainsi que les termes acoustiques employés pour lui permettre de produire un premier graphique. Dans 95% des cas, le premier jet est quasiment intouchable. Merci à Simon pour sa compréhension et son talent. Maintenant, vous savez ! !

J'aimerais réitérer les sentiments de Stan Dosso et transmettre mes remerciements aux membres de l'Association Canadienne d'Acoustique. J'ai eu la chance de travailler avec John Bradley qui était très aimable et avait sa façon pour corriger les erreurs que quelqu'un commettrait sous les pressions d'échéances. Il a toujours était le premier à envoyer ses félicitations à la publication de chaque édition. Il appréciait mon dévouement à la publication du journal à temps. Maintenant qu'il n'est plus président, j'aimerais le remercier pour sa diligence.

Nous avons commencé deux nouveaux processus dans le journal. Notre objectif était de pouvoir fournir à nos auteurs la possibilité d'acquérir des copies de leurs articles, mais le processus s'était avéré très lourd et les coûts exigés par les anciens éditeurs étaient très élevés. Maintenant que nous avons trouvé un éditeur local, les coûts sont plus raisonnable grâce à l'impression numérique. Nous avons donc décidé d'offrir ce service à nos auteurs. Le coût est de Cd$60 pour 50 copies et Cd$120 pour 100 copies, plus les frais d'envois qui sont de Cd$ 25 et Cd$40 pour courrier express, respectivement pour 50 et 100 copies. Les auteurs peuvent passer commande en contactant directement l'éditeur. Les copies seront envoyées une fois la totalité du payement est effectuée. Ce service est offert à tous les articles publiés dans le CAA journal depuis 1999, i.e. depuis le Volume 29. Je mettrai prochainement un exemple en format pdf sur notre site internet.

Finalement, un développement excitant est en expérimentation dans l'édition de juin 2004 du Journal. Si vous vous rappelez, un des objectifs du journal est de promouvoir l'organisation des forums et conférences sur les activités acoustiques au Canada. Récemment, en novembre 2003, DRDC de Halifax a organisé un forum, intitulé 'Forum

contd. on Page 2. . .

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to publish them in the June 2004 issue. The special June issue will be another proceedings issue where we showcase the efforts of the researchers at DRDC, Halifax, NS. We hope other coordinators of conferences or workshops in Canada will follow the example set by Ms. Desharnais and consider publishing the articles in Canadian Acoustics. I am sure that all of you will agree with me that CAA's profile will be expanded with these initiatives.

2003 sur la détection et localisation des mammifères marins à l'aide de moyens acoustiques passifs' sous la supervision de notre ancienne rédactrice en chef, Francine Desharnais. Après révision des articles présentés, nous avons décidé d'en publier plusieurs dans l'édition de juin 2004 du journal. Cette édition sera spéciale et est entièrement dédiée aux efforts de recherche au DRDC de Halifax, NS. Nous espérons que d'autres coordinateurs de conférences ou forum au Canada suivront l'exemple de Mme Desharnais et envisagent de publier les articles dans notre journal. Je suis sur que vous êtes tous d'accord avec moi que le profil de CAA sera plus étendu avec de telles initiatives.

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A METHOD FOR SELECTING CHIEF POINTS IN ACOUSTIC SCATTERING

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ABSTRACT

In this work, the nonuniqueness problem of solving surface integral equation of acoustic scattering is considered. The solution of the acoustic scattering integral equation is not unique at some frequencies. A unique solution can be obtained by adding some constraints to the problem at some interior points of the scatterer. The primary difficulty is the lack of formalized method for the selection on the interior points to guarantee uniqueness. A simplified method for selecting interior points for CHIEF method is proposed. The new augmented surface integral equation is successful in reducing the needed number of points to solve at the characteristic frequencies of the scattering problem where a unique solution does not exist. The implementation of the method exploits the earlier computations used in selecting the interior points. Numerical results are presented at some characteristic frequencies for an axisymmetric body. A comparitive analysis is also presented to evaluate the potential of the proposed method.

RÉSUMÉ

Ce travail, le problème de non unicité en résolvant l'équation intégrale extérieure de la dispersion acoustique est considérée. La solution de l'équation intégrale de dispersion acoustique n'est pas unique à quelques fréquences. Une solution unique peut être obtenue en ajoutant quelques contraintes au problème à quelques points intérieurs du diffuseur. La difficulté primaire est le manque de méthode formalisée pour le choix sur les points intérieurs pour garantir l'unicité. On propose une méthode simplifiée pour choisir les points intérieurs pour la méthode en CHIEF. La nouvelle équation intégrale extérieure augmentée, prouve le succès en réduisant le nombre nécessaire de points en cas de solution aux fréquences caractéristiques du problème de dispersion, auquel le problème n'a pas une solution unique. L'exécution de la méthode exploite les calculs précédents utilisés en choisissant les points intérieurs. Des résultats numériques sont présentés à quelques fréquences caractéristiques pour un corps axisymmetric. Une analyse de comparaison est également présentée pour évaluer le potentiel de la méthode proposée.

1. INTRODUCTION

Integral equation methods have been used to solve exterior acoustic radiation and scattering problems for many applications. In these problems, the external pressure is represented in terms of a distribution of an acoustic field on the surface of a scatterer or radiator. By forcing this representation to match a specified velocity distribution on the surface, an integral for the unknown source strengths was obtained. Once the source density is obtained, the pressure at any point in the exterior region can be computed. Integral equations of this type do not have a solution at the natural frequencies of an associated interior Dirichlet problem [1].

The surface Helmholtz integral equation is advantageous in that the problem's dimensionality is reduced by one and an infinite domain is transformed to finite boundaries in which the far-field radiation condition is satisfied. The solution of the acoustic (Helmholtz integral) boundary value problem is unique for all frequencies [1, 2]. However, the standard Helmholtz integral equation fails to yield unique solutions at the natural frequencies. For direct formulations, both the Dirichlet and Nuemann problems have the same characteristic frequencies as the eigen frequencies of the interior Dirichlet problem [5]. This problem is one of nonuniqueness rather than nonexistence [1]. Non-uniqueness is a purely mathematical problem arising from the breakdown of boundary integral representation rather than from the nature of the physical problem [5, 3]. For solving this problem, two main approaches have been followed; the first method is called CHIEF, the combined Helmholtz integral equation approach, which is perhaps the most widely used in engineering applications. The second method combines the Helmholtz integral equation with its normal derivative.

The CHIEF method uses the surface Helmholtz integral equation, combined with the corresponding interior Helmholtz integral equation, to form over-determined system of equations which can then be solved. This method may not function properly at the characteristic frequencies when some of the interior points coincide with a nodal surface of the related interior problems. More CHIEF points may be required to yield unique solutions at the characteristic frequencies. Therefore, in this method, the number and location of CHIEF points must be effectively selected, particularly in the high frequency range. The second method is severely limited in that the hyper-singular integral must be evaluated and so numerical difficulties arise. The tangential derivative formulation has been derived to regularize the highly singular kernels [4].

In general, a comparison of these two methods shows that they introduce their own particular complications [4]. Many enhancements have already been introduced to both these methods. Benthien and Schenck, in a recent survey, reviewed various methods for handling the nonexistence and non-uniqueness [1].

The CHIEF method is the most extensively used in engineering applications [5]. A potential problem with this approach is the choice of interior points for the supplemental equations. The selected interior points must not be a nodal point of the corresponding interior eigen mode. It has been demonstrated that it is only needed one non-nodal point (good) to establish a unique solution [2]. A survey on the location of "good" points is, also, found in Reference 2. The primary difficulty is the lack of formalized method for the selection on the interior points to guarantee uniqueness [6]. Despite that there is no systematic way to select the interior points in CHIEF method, the selection of effective interior points is not difficult. Several decades of practical experience has shown that effective interior points are not difficult to choose and that the CHIEF method is very robust [1].

In this work a monitoring method is proposed to test the effectiveness of interior points for CHIEF. The proposed method is simple, needs neither rigorous mathematical formulations nor significant computational burden. The proposed method exploits a non-unique solution at the required characteristic frequency on the scatterer surface. The non-unique solution, is, then used to compute the field at some interior points. The computed field strengths can monitor any nodal points using a simple criterion. The matrix used in the first run is, then, augmented with non-nodal interior points to give the over-determined system of equations which can be solved for the unique surface field.

2. INTEGRAL EQUATION DERIVATION

The governing equation for the propagation of acoustic waves through an unbounded homogenous medium is described by the wave equation

$$\nabla^2\phi(r,t) = \frac{1}{c^2}\frac{\partial^2\phi(r,t)}{\partial t^2} \tag{1}$$

where, ∇^2 denotes the Laplacian operator in three dimensions. ϕ the velocity potential at r and t. c is the speed of sound in the medium at the equilibrium state. The velocity potential ϕ is

$$\mathbf{u} = \nabla\phi \tag{2}$$

It is common practice to express the velocity potential as

$$\phi = \phi^i + \phi^s \tag{3}$$

where $\boldsymbol{\phi}^{\mathbf{i}}$ and $\boldsymbol{\phi}^{\mathbf{s}}$ are the incident and the scattered velocity potentials. The excess pressure can be written as

$$p = -\rho_o \frac{\partial\phi}{\partial t} \tag{4}$$

where ρ_o is the density of the fluid at the equilibrium state. It follows that,

$$p = p^i + p^s \tag{5}$$

where, p^i and p^s are the incident and the scattered pressures. The differential equation for time-harmonic waves with a time factor $e^{i\omega t}$ takes the form

$$(\nabla^2 + k^2)\phi = 0 \tag{6}$$

where, $k=\omega/c$ is the wave number, and ω the angular frequency. Accordingly, equation (4) becomes

$$p = -i\rho_o\,\omega\phi \tag{7}$$

At the surface of a hard scatterer, the normal component **u.n** of the fluid particle velocity **u** is zero; so

$$\frac{\partial\phi}{\partial n} = 0 \tag{8}$$

where **n** is the unit vector normal to the surface of the scatterer body and into the surrounding space, and n is the distance along the external normal vector **n.** At the surface of a soft scatterer, the excess pressure is zero, (i.e.),

$$\phi = 0 \tag{9}$$

Equations 8 and 9 represent the Neumann and Dirichlet boundary conditions of the differential equation, Equation 1, respectively.

The equivalent boundary integral formulation of Equation 6 is valid for an acoustic medium B' exterior to a finite body B with surface S and a unit normal **n,** pointing into B'. The body is submerged into an infinite linear acoustic medium. When a harmonic acoustic wave ϕ^i impinges upon the body B, the resulting integral equation for smooth boundaries has the following form.

$$C(P)\,\phi(P) = \int_S \left(\phi(Q)\,\frac{\partial \psi(P,Q)}{\partial n} - \psi(P,Q)\frac{\partial \phi(Q)}{\partial n} \right) dS_Q + 4\,\pi\,\phi^i(P) \qquad (10)$$

Equation 10 is the surface Helmholtz integral equation, where, $\phi(P) = \phi(r_P)\, e^{i\omega t}$ at point P and Q is a point on the body surface.

The free-space Green's function ψ for the Helmholtz wave is given by

$$\psi(P,Q) = e^{-ikR} / R \qquad (11)$$

where, R is the distance between the field point P and a source point Q, and $\boldsymbol{n}$ is the outward directed normal at Q. The distance is expressed, vectorially, as

$$R = \left| \boldsymbol{r_P} - \boldsymbol{r_Q} \right|, \qquad (12)$$

where, $\mathbf{r_p}$ is the vector to point P from the origin and similarly for Q. The coefficient $C(P)$ is defined at P on S provided that there is a unique tangent to S at such a P, as

$$C(P) = \begin{cases} 0 & for \;\; P \in B' \\ 4\pi & for \;\; P \in B \\ 2\pi & for \;\; P \in S \end{cases} \qquad (13)$$

When P occupies a point on the surface S, there is no unique tangent plane (e.g., when P is on an edge of a sharp corner), $C(P)$, then relates to the solid angle α by [4].

$$\begin{aligned} C(P) &= 1 - \alpha / 4\pi \\ &= 4\pi + \int_S \frac{\partial}{\partial n}\left(\frac{1}{R(P,Q)} \right) dS_Q \end{aligned} \qquad (14)$$

3. DESINGULARIZATION

We consider here only the fully axisymmetric scattering case, (i.e), both the body shape and the acoustic variables are independent of the angle of the revolution of the body. For scattering, this implies that the direction of the incident wave must coincide with the axis of revolution of the body. This simplification is used to test the proposed nonuniqueness solution. The singularity regularization is similar to that used in Seybert and Ranganathan [2]. This formulation is summarized in the next section.

For an axisymmetric body, the integrals in Equation 10 can be rewritten using a cylindrical coordinate system (ρ, θ, z) as

$$\int_L \phi(Q) \left[\int_0^{2\pi} \frac{\partial}{\partial n}\left(\frac{e^{-ikR(P,Q)}}{R(P,Q)} \right) d\theta(Q) \right] \rho(Q) dL(Q) \qquad (15)$$

and

$$\int_L \frac{\partial \phi(Q)}{\partial n} \left[\int_0^{2\pi} \left(\frac{e^{-ikR(P,Q)}}{R(P,Q)} \right) d\theta(Q) \right] \rho(Q) dL(Q) \qquad (16)$$

where, the axisymmetric assumption implies that the field $\phi(P)$ and its derivative are independent of $\theta(P)$ and the differential area element is defined as

$$\boldsymbol{dS(Q) = \rho(Q)\; d\theta(Q)\; dL(Q)} \qquad (17)$$

where, $dL(Q)$ is the differential length of the generator L of the body at a surface point Q, where Q now is interpreted as an arbitrary point on L only.

The evaluation of the integrands in Equations 15 and 16 requires the evaluation of the following integrals

$$I_1 = \int_0^{2\pi} \left(\frac{e^{-ikR(P,Q)}}{R(P,Q)} \right) d\theta(Q) \qquad (18)$$

$$I_2 = \int_0^{2\pi} \frac{\partial}{\partial n}\left(\frac{e^{-ikR(P,Q)}}{R(P,Q)} \right) d\theta(Q) \qquad (19)$$

These integrals are singular and the singularities can be removed by using the following regularization scheme.

$$I_1 = IG_1 + IE_1 \tag{20}$$

where,

$$IG_1 = \int_0^{2\pi} \left(\frac{e^{-ikR(P,Q)} - 1}{R(P,Q)} \right) d\theta(Q) \tag{21}$$

and

$$IE_1 = \int_0^{2\pi} \left(\frac{1}{R(P,Q)} \right) d\theta(Q) \tag{22}$$

The integrand in Equation 21 is nonsingular. However, it can be evaluated numerically using a simple Gaussian quadrature formula. The other integrand defined in Equation 22 can be reduced to elliptic integral formulation and evaluated using standard algorithms. Elliptic integral algorithms are available in the numerical toolboxes on most computers [8].

A similar procedure has been used in Reference 2 to evaluate the integrand in Equation 19.

$$I_2 = IG_2 + IE_2 \tag{23}$$

where,

$$IG_2 = \int_0^{2\pi} \frac{\partial}{\partial n} \left(\frac{e^{-ikR(P,Q)} - 1}{R(P,Q)} \right) d\theta(Q) \tag{24}$$

and

$$IE_2 = \int_0^{2\pi} \frac{\partial}{\partial n} \left(\frac{1}{R(P,Q)} \right) d\theta(Q) \tag{25}$$

4. NUMERICAL FORMULATION

Substituting Equations 21 thru' 25 into Equation 10 at different node points i_p and assuming the index of surface elements i_q, the following discretized form of Equation 10, for N nodes on the surface, can be written as

$$A\,\phi = B \tag{26}$$

where, A is an NxN matrix. ϕ and B are N vectors. An example for the hard scatterer where $\frac{\partial \phi}{\partial n} = 0$ is

$$A(i_p, i_q) = \sum_{i_q=1}^{N} I_1 \rho(i_q) dL(i_q) \quad i_p \neq i_q \tag{27}$$

$$A(i_p, i_p) = \sum_{i_q=1}^{N} I_1 \rho(i_q) dL(i_q) - 2\pi \quad i_p = i_q \tag{28}$$

and

$$B(i_p) = -4\pi\phi^i(i_p) \quad \forall i_p = 1..N \tag{29}$$

where, ϕ is an N -dimension vector representing the field strength on the scatterer surface and ϕ^i is the incident field.

5. NON-UNIQUENESS ISSUES

CHIEF method is used to solve the acoustic scattering problem at the natural frequency. Nonuniqueness manifests itself numerically by producing a nearly singular coefficient matrix A. At these points, the field strength $\phi(i_p)$ is forced to vanish. The interior integral relations are used as constraints that must be satisfied along with the original formulation. Equation 29 is, then, augmented by additional equations at interior points. These equations differ from Equation 29 in that the additional point doesn't lie on the surface. Thus, no singularity is found and the field strength is zero so the 2π in R.H.S of Equation 29 is dropped.

For axisymmetric bodies, it was found that the axis is a good place to put the interior points [1]. The selection of interior points is based on evaluating the field at some interior points on the axis of symmetry and select only those points where the field doesn't vanish, within a preset accuracy [6]. Augmenting the N discretized equations in Equation 26 by M interior equations results in an overdetermined system of equations which can be solved by the least squares method [1]. The nonuniqueness can be overcome using a simple method based on the CHIEF criterion. The proposed method exploits a non-unique solution at the required characteristic frequency on the scatterer surface. The non-unique solution is, then, used to compute the field at some interior points. The computed field strengths can monitor any nodal points using a simple criterion based on that the point of larger field strength is far from nodal points. The matrix used in the first run is, then, augmented with non-nodal interior points to give the over-determined system of equations which can be solved for the unique surface field.

6. NUMERICAL RESULTS

The numerical example, presented here, is the scattering of a plane incident wave by a rigid sphere and is solved at some characteristic frequencies of the problem. The incoming unit plane wave travels toward the scatterer along the positive direction of z-axis in the cylindrical coordinates described as e^{-ikz}. The surface field ϕ is computed using the proposed method. The results will be verified by comparing with the analytical solution. The benefits of the method will be validated by comparing with the results of CHIEF method with multiple points. On the surface of a hard sphere, the analytical solution of Equation 10 for plane incident wave can be expressed as [8].

$$\phi = \frac{i}{(ka)^2} \sum_{n=0}^{\infty} (-i)^n (2n+1) \frac{P_n(\cos\theta)}{h_n^{(1)'}(ka)} \qquad (30)$$

where, ϕ is the total field as defined in (3) and θ is the incidence angle and it has been taken to be *zero* in this application. P_n is the Legender polynomial of order n and h_n is the spherical Hankel function. k is the wave number and a is the radius of the sphere.

Table.1 Field points at different interior points

Normalized interior points z-component	Verified Field
0.90476	0.396
0.80952	0.816
0.71429	1.194
0.61905	1.480
0.52381	1.630
0.42857	1.620
0.33333	1.441
0.23810	1.108
0.14286	0.656
0.04762	0.153
-0.04762	0.433
-0.14286	0.929
-0.23810	1.333
-0.33333	1.602
-0.42857	1.711
-0.52381	1.659
-0.61905	1.460
-0.71429	1.147
-0.80952	0.763
-0.90476	0.351

The performance of each method will be evaluated based on two factors; the computational time and accuracy relative to the analytical solution.

The compared results are taken at the characteristic frequencies; *ka= 4.4934* which is a fictitious frequency of the normal derivative boundary integral equations [3]. The system of equations is solved numerically using least-squares algorithm which is available in [8].

In applying the proposed method, the computations of the non-overdetrmined system of equations are saved by retaining its matrix into memory after monitoring the proposed CHIEF points. The selected point based on the proposed criterion is used to add, just, one matrix row and then the least-squares method is applied to the over-determined system of equations.

Table. 1 shows the verified field at different interior points to select the appropriate one for CHIEF analysis. The field verification is computed through non-overdetermined system of equation for the integral equation for the characteristic frequency *ka=4.4934*. These points are chosen with constant step on the z-axis of symmetry in cylindrical coordinates. The incident field is propagating along the z-direction. According to the proposed method of selection, the field at the candidate point should exceed a certain threshold. Different trials showed that the threshold $\phi/\phi_i > 0.5$ is sufficient.

According to the above discussion, the most appropriate point, which has the most far field value from zero, is *z=0.524.* Figure 1 shows a comparison between the scattered field distribution on the surface computed without any correction and the analytical solution. Figure 2 shows a comparison between the scattered field distribution on the surface computed using the proposed method, CHIEF with multiple points and the analytical solution.

Figures 3-6 show different comparisons between the analytical solution and different selected CHIEF points according to the above criterion. These comparisons show that the selected points serve to adjust the solution in its neighborhood range, since negative points show closer result to the analytical solution around its neighborhood. Applying the method with two points on both sides of the axes of symmetry shows the most accurate result within these comparisons. Adding, more points don't add more accuracy as shown in Figure 3.

CPU time comparison shows that each additional CHIEF point adds about 5% of the whole process time to be processed. The comparison has been conducted under MATLAB 5.2 on PC-233MHz for 10 surface points and 10 additional interior points.

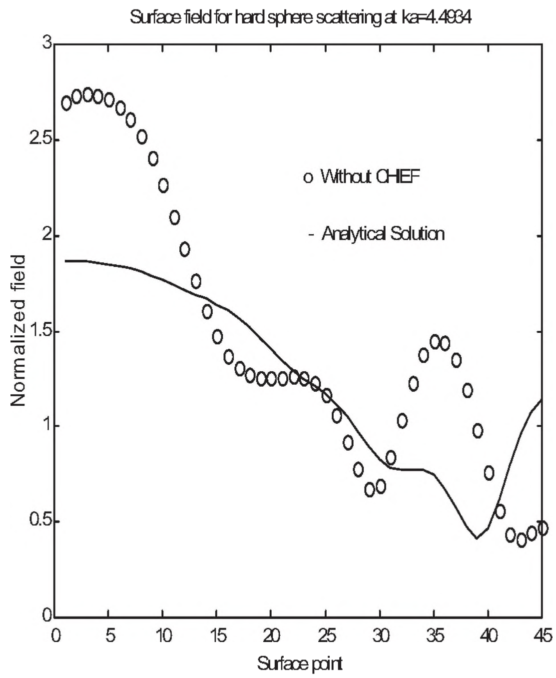

Figure 1. Comparison between the analytical solution at the characteristic frequency *ka=4.4934* and the solution without CHIEF correction.

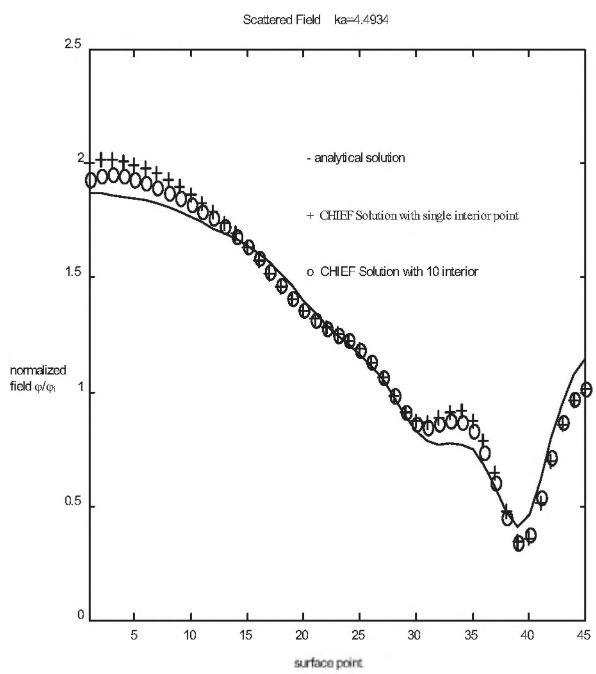

Figure 2. Comparison between different solution methods.

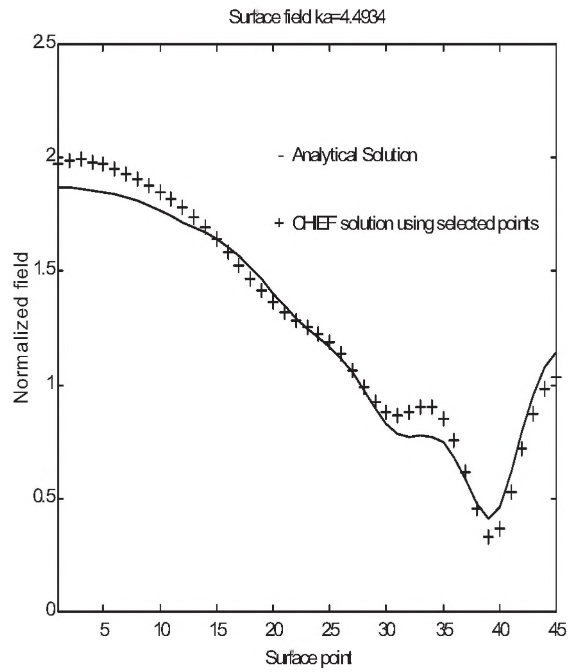

Figure 3. Comparison between analytical solution and a numerical solution at selected point of z=0.52 0.43 and 0.33 respectively.

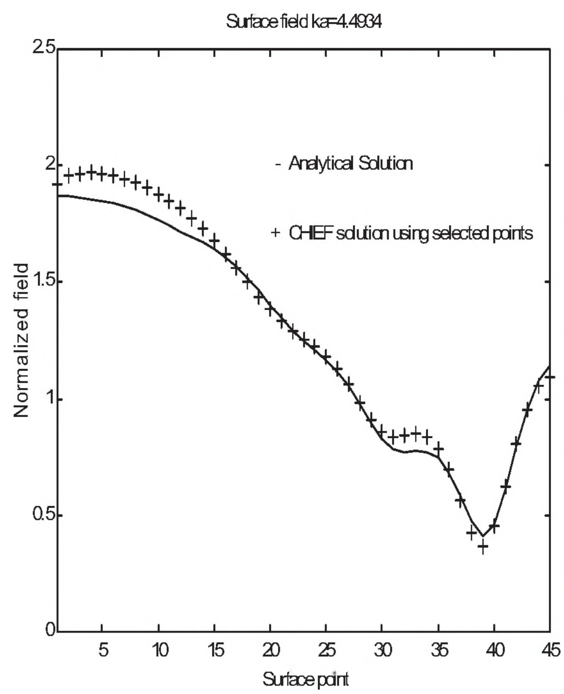

Figure 4. Comparison between analytical solution and a numerical solution at selected point of z=0.62, 0.52, 0.43, -0.62, -0.52 and - 0.43 respectively.

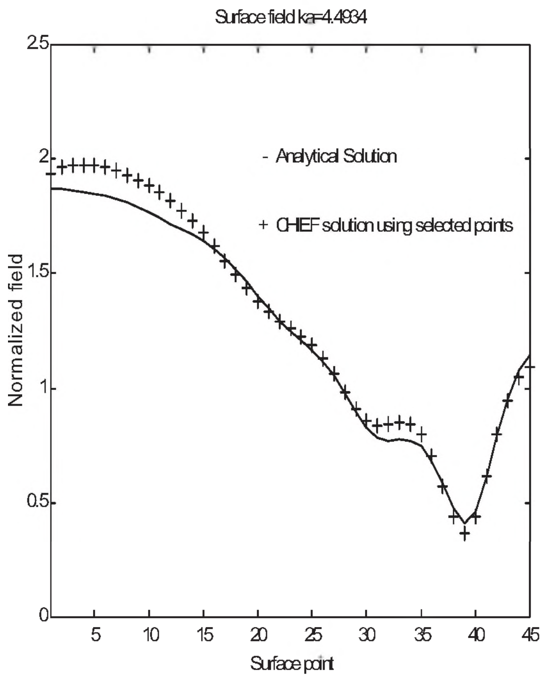

Figure 5. Comparison between analytical solution and a numerical solution at selected point of z= 0.52, 0.43, -0.52 and - 0.43 respectively.

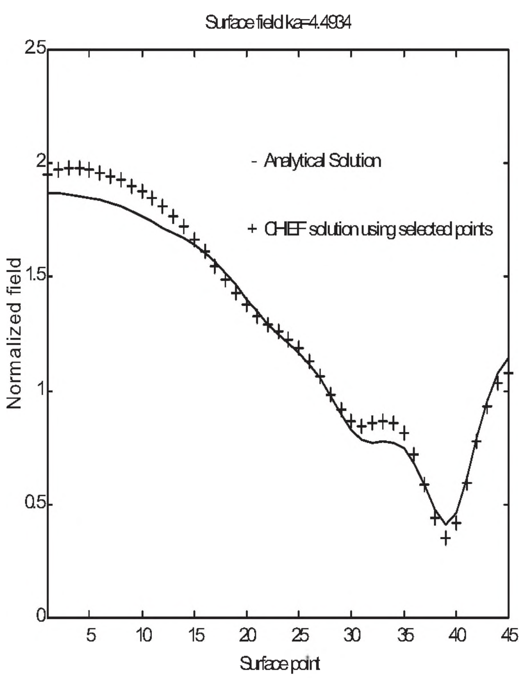

Figure 6. Comparison between analytical solution and a numerical solution at selected point of z= 0.52, 0.43, and -0.52 respectively.

7. Error Analysis

The results obtained in the previous section are then compared based on the error from analytical solution. The error is defined as

$$\boldsymbol{Error} = \boldsymbol{\phi_{num}} - \boldsymbol{\phi_{ana}} \tag{31}$$

where, ϕ_{num} is the resulted solution form Equation 29 and ϕ_{ana} is the analytical solution given in Equation 30.

Figure 7 shows a comparison between the error trends of the numerical solutions using 10 CHIEF-points and one CHIEF-point at $z = 0.52$ as selected by the discussed criterion in section 5 as an extreme choice. Figure 8 shows another comparison between 10 points solution and the solution of 3-points shown in Figure 6.

The energy error is also compared for different CHIEF-point selections in Table 2. The energy error is defined as the ratio between the energy of the error to the energy of the analytical solution as

$$\text{Energy Error ratio} = \frac{\|\phi_{num} - \phi_{ana}\|}{\|\phi_{ana}\|} \tag{32}$$

Table.2 Energy Error Ratio for different Chief-point selections.

# Chief Points	10	6	3	1
Energy Error Ratio	0.0448	0.0456	0.0550	0.0592

Table 2 shows the effect of CHIEF-points on the accuracy of the numerical solution. According to these results, the trade-off remains between less accuracy gain as shown in Table 2, and Figures 1thru' 8 and saving computational cost by reducing the solution matrix as discussed in Section 6.

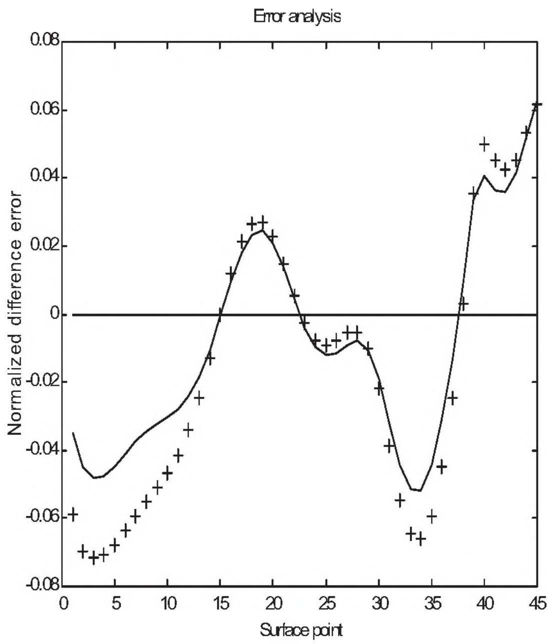

Figure 7. An error comparison between one and 10 CHIEF-point solutions.

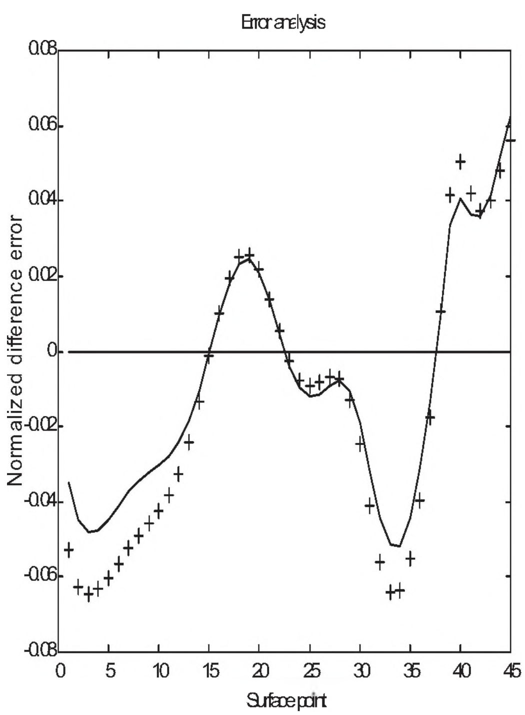

Figure 8. An error comparison between 3 and 10 CHIEF-point solutions.

8. CONCLUSIONS

This work considered the problem of selecting a non-nodal interior point for CHIEF method. This method is essential for solving the acoustic scattering problem at the characteristic frequencies of such problems. The non-uniqueness needed only one interior additional field equation [1]. The selection method, proposed in this work, provided a systematic way which saves both computational time and memory. The results showed convenience with the analytical solution and the traditional solution using multiple interior constraint equations. Error analysis showed that more internal CHIEF-points do not improve accuracy while fewer points reduced the computation time. The proposed method for selecting such points helped in reducing that number. According to the results, the trade-off still remains between less accuracy gain and saving computational cost by reducing the solution matrix.

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Hearing Protectors – Calculations of the Noise Level at the Protected Ear

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ABSTRACT

The risk of noise induced hearing loss, when wearing a hearing protector, can only be calculated with the knowledge of the noise level at the protected ear. Several methods are available for calculating this level, from the hearing protectors' attenuation data. However, most available data are obtained from procedures not usually found in real world situations. Therefore, the results from the calculations are corrupt. This paper reviews some existing methods for the attenuation measurements and noise level calculations. Some ways to overcome the use of non-realistic attenuation data are suggested in the paper.

RÉSUMÉ

Le risque de perdre l'ouïe à cause de bruits excessifs, lorsque qu'un protecteur auditif est utilisé, ne peut être que calculé en connaissant le niveau de bruit à l'oreille protégée. Plusieurs méthodes sont disponibles pour effectuer ce calcul en utilisant les données de l'atténuation du bruit du protecteur auditif. Toutefois, les données disponibles sont obtenues grâce à des méthodes fort éloignées des situations que l'on retrouve dans la réalité de l'environnement de travail. C'est pour cette raison que les résultats sont erronés. Cet article revoit les méthodes existantes pour les calculs des mesures d'atténuation et du niveau du bruit. Il suggère aussi différentes façons pour éviter l'utilisation de données inexactes d'atténuation du bruit.

1. INTRODUCTION

In October 2003, as a part of a Noise Control Seminar organized by the Occupational Hygiene Association of Ontario (OHAO), this author made a presentation on the subject of hearing protectors. Members of the audience suggested that a tutorial on the use of the protectors' attenuation values would be useful for professionals and users alike. This paper is the result of this suggestion.

Hearing protectors are the most frequently used devices for reducing the hazard of hearing loss in the industry as well as in construction. The reasons for this choice are: their relatively low cost; the ease and speed of introducing the protectors in the workplace (even though in many occasions this is not done properly); and the urgent need of showing that something has been done to protect workers from excessive noise.

Hearing protectors are intended to reduce the noise level reaching the inner ear. Therefore there is strong need to know the level of noise reduction provided by the hearing protector, i.e., their attenuation, so that the noise level at the protected ear can be calculated. Noise level at the ear, resulting from donning a protector is the level that is effective when the HPD is worn, i.e. the diffuse field level that would have created that level in the ear canal, minus the attenuation of the device

The objectives of this paper are: to review the different ways of calculating this level; to highlight the pitfalls associated with these calculations; and to suggest ways to overcome the use of non-realistic attenuation data.

2. BACKGROUND

attenuation of hearing protectors is the measurement of the shift of the threshold of hearing of a subject, resulting from donning a protector. The method is known as REAT: Real-Ear Attenuation at Threshold. For this procedure, a person (the subject) "lends" his head to have the protector donned and uses his hearing to detect the minimum sound level that he can perceive (his hearing threshold). Two thresholds are measured: one with and one without the protector in place. The difference between the two thresholds is, by definition, the attenuation of the protector. The signals that the subject is to hear are third-octave band filtered white noise, centered at the audiometric frequencies between 125 and 8000 Hz. The reason for testing at different frequencies is because the attenuation of protectors is frequency-dependant.

When the protector is worn sound reaches the inner ear following three paths:

a) Through the body of the protector,
b) Through cracks between the protector and the skull (in the case of a muff), or between the protector and the ear canal (in the case of a plug), and

c) Through the skull bone (bone conduction).

For all practical purposes, only the first two paths are of importance (unless a protector is very well fitted or unless a combination of an earmuff and earplug is worn), and can reduce considerably the attenuation offered by a protector. The attenuation through the bone is approximately 45 dB, much higher than that of commercial protectors as typically worn [1, 2].

Standards require that between 10 and 20 subjects are used for the measurements to account for the anatomical differences among people. To avoid false results, the attenuation is measured at least twice for the same subject. The results are treated statistically and presented as a table or graph of the mean values and standard deviations of the individual results at each of the above-mentioned frequencies as shown in Figure 1. The convention for the graphs in Figure 1 is that attenuation always appears as negative, while the standard deviation is positive.

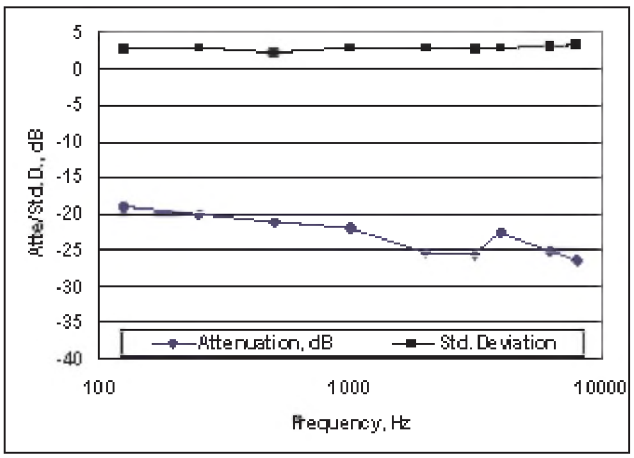

Figure 1. Performance of Hearing Protectors.

Several standards deal with the procedures to be followed when measuring the attenuation, as well as with the instrumentation and environment required for the tests. The most important among them, due to their wide usage, are the ISO 4869-1 and the ANSI S12.6 – 1997 [3, 4]. The ANSI standard contains two measurement procedures. The first of them, Method A, is similar to that in the ISO Standard with small differences, mostly in operational details. Both Standards make use of subjects that are familiar with this kind of tests. The procedure is designed to measure the maximum attenuation that can be achieved using the protector under test. For this purpose, if a test results in an unexpected, low attenuation value, the measurement is repeated. The role of the subject is limited to just fitting the protector under the operator's supervision and to indicate when the signal is perceived.

The second measurement procedure in the ANSI S12.6 – 1997 Standard (Method B) requires the subject to don the protector following the manufacturer's written instructions. The operator is not allowed to intervene in the process, or to provide help or instructions. Also, subjects are naïve: they are not allowed to have previous experiences wearing or testing hearing protectors.

A new ISO Standard, 4869 – 7 is presently in preparation [5]. It follows closely the procedures in the ANSI 12.6 – 1997, Method B. The reason for the ANSI method B and for the new ISO standard will be further discussed in this paper. This reason is based on the fact that the attenuations measured following the Method A or the existing ISO standards are significantly larger than those obtained in realistic work conditions.

4. THE ATTENUATION DATA

There are several procedures for the calculation of the sound level at the protected ear using the results from the attenuation measurements. All of them make use of mean attenuation values and their standard deviations. Following are some of the available calculations.

The ISO 4869 -2 Standard contains three different calculation procedures [6]:

a) The Octave-Band Method: The noise level in the workplace is measured in Octave bands. Then, at each frequency, the attenuation of the protector is subtracted from the noise level. Finally, the difference is corrected according to the percent protection performance, correction that is included in the Standard. The percent protection performance is the percentage of situations for which the A-weighted sound pressure level is equal to or less than the predicted value. The result of the calculation is the sound level of the protected ear in octave bands. Considered as the most accurate calculation, it has not been adopted by users, probably because of the complexity of interpreting the results without converting them to dBA.
b) The HML Method: Relatively popular among the Northern countries in Europe, this method requires the ambient noise to be measured in dBA and dBC. Three coefficients, H, M and L are supplied by the manufacturer. They are calculated using the protector's mean attenuation values and standard deviations. The noise level of the protected ear is calculated in two steps using the above data.
c) The Single Number Rating (SNR): This method requires the measurement of the ambient noise in dBC. A coefficient, SNR, is calculated by the manufacturer using protector's mean attenuations and standard deviations. The noise level at the protected ear is calculated using the ambient noise level, measured in dBC, and the SNR, corrected according to the specific protection performance.

There are other computational procedures that also are used extensively. They are:

a) The NRR. This is the single number rating most often used on this side of the Ocean. It has gained popularity mainly because of the ease of use and also because the USA Environmental Protection Agency EPA requires NRR to be printed on every hearing protector's package [7].

As with the previously described indices, the NRR is calculated from the mean value of the measured attenuations at all audiometric frequencies and the calculated standard deviations of the measurements. Because two standard deviations are subtracted from the mean value of the attenuation, in theory 98% of users will have the calculated noise level of the protected ear or higher. Only 2% of users will have attenuation lower than the calculated. The inherent assumption is that the attenuations are normally distributed. Although this is not a proven fact, it is a working tool used across the hearing protection community.

By definition, the sound level of the protected ear is obtained as follows:

SL = SL(dBC) – NRR, or
= SL(dBA) – NRR + 7 dB,

where SL(dBC) and SL(dBA) are the ambient sound levels measured in dBC and dBA.

b) The NRR(SF) [8]: This is a rating calculated with results from measurements following procedures in the Method B of the ANSI 12.6-1997 Standard [4]. The sound level of the protected ear is calculated in a similar way as with the NRR with some small differences. For the calculation of the NRR(SF), only one standard deviation is subtracted from the mean value. The resulting noise level at the protected ear applies, therefore, to 84% of the wearers.

Results from studies indicate that the calculated sound level at the protected ear using the Octave Band method, the NRR and the HML, is approximately the same [9, 10].

5. PROBLEMS WITH THE CALCULATION RESULTS

Extensive studies have demonstrated, that attenuation values measured in laboratories and reported by manufacturers are significantly higher than those measured in the field [11, 12]. The main reason for the field lower performance appears to be the poorer fitting of the protectors due to lack of training and motivation, poor choice of size in the case of plugs, lack of attention while fitting, etc. Berger quotes the following reasons: comfort, utilization, readjustment, fit, compatibility, deterioration and abuse [13].

Figure 2 is from Reference 14 and shows a comparison between protectors' NRR measured in laboratories and in the field. The field NRRs are calculated using one (1) standard deviation and not two (2) as is the case with the laboratory

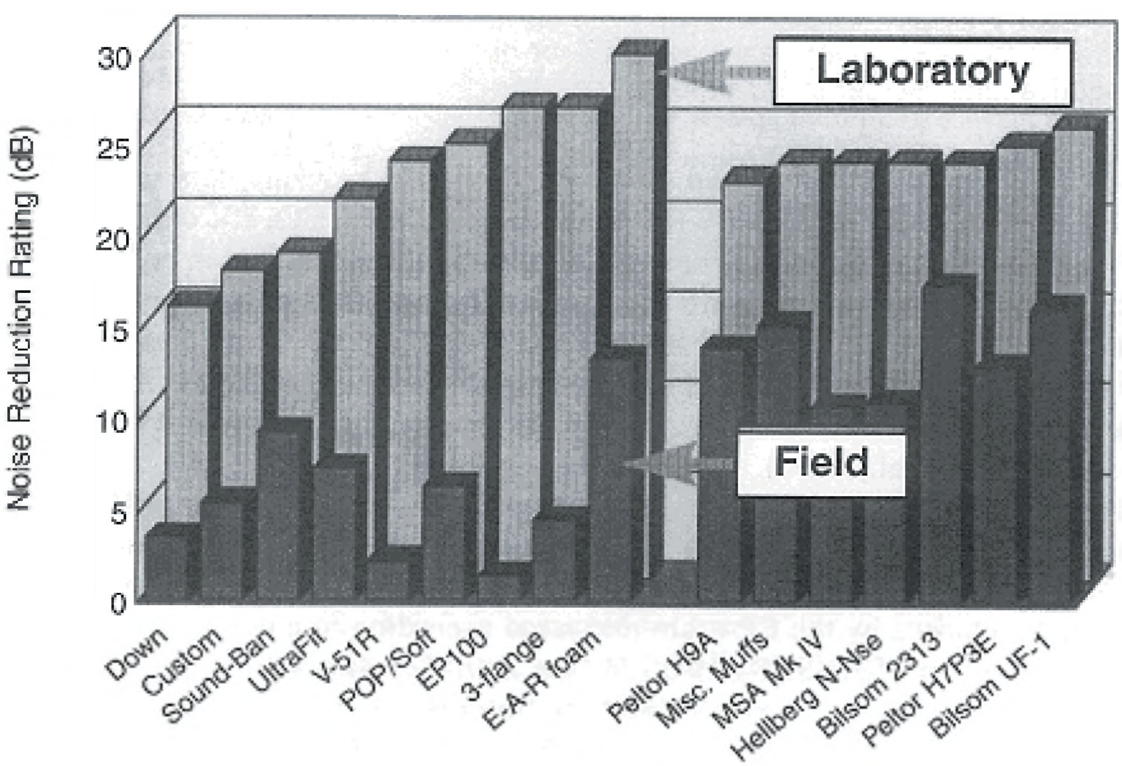

Figure 2: Comparison of NRRs published in North America (labeled values based upon laboratory tests), to real-world "field" attenuation results derived from 22 separate studies.

NRRs. It can be seen that not only the field attenuations are much lower, but also, that there is no relation between both attenuations. For example the laboratory result of the EP100 plug is one of the highest among plugs, while the field one is one of the lowest. The same applies to muffs: see the results for the MSA Mk IV.

There is no need to stress the consequences of using overly optimistic NRR values for the noise exposed population: users would think that they are protected, while, in reality they are not.

However, it must be pointed out that the problem is not inherent to the calculation of the NRR, nor that manufacturers are to be blamed for reporting high NRR values. The problem resides in the data obtained through laboratory measurements that are intended to yield optimum performance values for the protectors. Those are the data used for calculations of the sound level in the protected ear using any of the above-described procedures: the "long" octave band method, the HML, the SNR or the NRR. Using any of them will yield abnormally high attenuation values for the protectors and non-realistic, low values for the sound level at the protected ear.

Therefore, the solution is not in the calculation method, but in the way the attenuation data is obtained. This is why the ANSI 12.6-1997 measuring method B was devised as an alternative to the Method A. This also is the reason for the new ISO 4869-7 (5). As mentioned above, when measuring hearing protectors following the method B, the resulting attenuations are similar to those obtained in field tests in establishments with well-managed hearing conservation programs.

6. DERATING SCHEMES

The disparity between laboratory and field attenuation results has been long recognized. Also, it has been recognized that there is no straight relation between both values. Therefore, there is no mathematical operation that would allow for the calculation of one using the results from the other. In view of the above, several derating schemes have been proposed.

As per OSHA to calculate the noise level of the protected ear, the NRR should be derated by 50% before it is subtracted from the sound level measured in dBC [15]. For example, if the sound level is 100 dBC, and the nominal NRR of a given protector is 30, the noise level of the protected ear will be 100 – 30/2 = 85 dBA.

NIOSH recommends derating NRR by a factor of 75%, 50% and 30% respectively for muffs, slow-recovering plugs and all other plugs [16]. As an example, a muff with a nominal NRR of 30 will be derated to NRR = 20. However, if the protector is a foam plug, it will be reduced to NRR = 15.

Behar recommends a variable scale, similar to that of NIOSH, where 7 will be subtracted from muffs' NRR, 10 from plugs and 13 from cap-mounted muffs [17].

7. POTENTIAL SOLUTIONS

The present situation regarding the attenuation values of hearing protectors can be summarized as follows:

a) The NRR is still the best known and most used rating scheme
b) Very few manufacturers have their products tested as per the ANSI method B, therefore there are few data available for the calculation of NRR(SF).
c) There are no approved standards, other than ANSI12.6-1997 (4) that includes measurement methods that result in attenuation similar to the field attenuation data.

Until the NRR(SF) (or some alternative value) becomes available, the best alternative for users is to derate the nominal NRR using any of the previously described methods. . It will also be extremely useful if users (especially large manufacturing facilities) begin requesting their suppliers and manufacturers for attenuation data obtained using the ANSI method B procedure. Only the users' pressure will force manufacturers to start providing meaningful data for their products.

However, one needs to stress the fact that the simple use of hearing protectors, should not be considered as an alternative to a well-managed hearing conservation program that deals with all issues regarding the use of protectors.

8. ACKNOWLEDGMENT

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Intake Noise Cancellation Using A Manifold Bridging Technique

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ABSTRACT

Automobile manufacturers have expended considerable efforts to attenuate the many noise sources perceived within the passenger compartment with varying degrees of success. Given that these dominant noise sources have been attenuated, induction noise has become more noticeable. The present study investigates the feasibility of using a non-conventional noise cancellation technique. The investigation has attempted to improve the acoustic performance of the induction system by introducing a bridge between the exhaust and intake manifolds. The effectiveness of such a technique is investigated using Ricardo WAVE, a computational engine simulation technique that uses a one-dimensional finite-difference formulation. Graphical results using $1/12^{th}$ octave frequency spectra and three dimensional colour maps of both an unmodified and a bridged engine are presented for both steady state and transient engine cases. A sound quality analysis is also presented using the psychoacoustic metrics of Loudness, Fluctuation Strength and Articulation Index. While a reduction in overall sound level was achieved, an additional benefit of this technique proved to be in the realized sound quality of the induction noise with the implementation of the manifold bridge. This investigation continues with verification of the theoretical model to experimental measurements on a dynamometer.

RÉSUMÉ

Les constructeurs d'automobiles ont fait des efforts considérables pour atténuer les nombreuses sources de bruits perçus dans l'habitacle, avec un succès variable. Ces bruits dominants ayant été atténués, on a fini par percevoir plus clairement le bruit de l'induction. La présente étude examine la possibilité d'annuler ce bruit par une technique innovatrice. On a tenté d'améliorer la performance acoustique du système d'induction en établissant un pont entre les tubulures d'admission et d'échappement. L'efficacité d'une telle technique est testée à l'aide de Ricardo WAVE, technique de simulation virtuelle de moteurs qui emploie une formulation de différence finie à une seule dimension. Des résultats graphiques utilisant des spectres de fréquence de 1/12 octave et des plans en couleur à trois dimensions, d'un moteur non modifié et aussi d'un moteur équipé d'un pont, sont présentés pour des moteurs tournant de manière continue et de manière intermittente. Une analyse de la qualité du son est présentée également, qui utilise les mesures psychoacoustiques de Force, d'Amplitude de fluctuation et d'Index d'articulation. Une réduction du niveau global de bruit a effectivement été accompli, et cette technique a aussi donné un avantage supplémentaire: la qualité du bruit d'induction s'est améliorée, grâce à l'installation du pont. On poursuit la recherche en vérifiant le modèle théorique au moyen de mesures expérimentales faites sur un dynamomètre.

1. INTRODUCTION

There are many sources of noise in the modern day automobile which include the combustion process of the engine, exterior wind noise, tire noise as well as exhaust and intake noise. Manufacturers have responded with stiffer and better acoustically insulated bodies, better aerodynamics, improved tire technology and quieter mufflers. This has resulted in an overall reduction of sound pressure levels in the passenger cabin, which has also resulted in a more acute awareness of other sources of noise, specifically induction noise. Given this, greater emphasis is being given to the study of induction noise and what can be done to lessen its impact on the consumer.

Traditional acoustic attenuators of intake noise include simple Helmholtz resonators or perhaps adaptive passive systems which allow the acoustic resonator volume to react according to engine RPM. However, due to the growing popularity of smaller vehicles with increasing limits on under hood space, it is becoming more difficult to facilitate the installation of these traditional methods of intake noise attenuation.

As a result, new design concepts for induction noise attenuation are being investigated by automotive engineers. One such approach is active noise cancellation where preliminary work to date has shown promising results.

The objective of this work was to investigate the feasibility of attenuating automotive induction noise using active noise cancellation through the implementation of tuned exhaust noise feedback through the intake system. This process typically utilizes a computer controlled speaker as the negating noise source. However here, the reduction of acoustical energy in the intake system would be realized by using exhaust noise as the effective dynamic noise source instead of a speaker. This is accomplished with the introduction of an open physical bridge inserted between the exhaust and intake manifold of the engine.

2. ACOUSTICS OF AIR INDUCTION SYSTEMS

The noise emitted from an automotive induction system is the result of a combination of two processes. The first process is the propagation of pressure pulses generated when the intake valve opens to the cylinder which has a pressure greater than atmospheric. A second pulse occurs when the intake valve closes [1]. The repetition of these pulses result in the oscillation of intake air at the natural frequency of the inlet passage column. The frequency of these pulses is further reduced to approximately 80 to 150 Hz in the engine firing range due to the influence of the intake system ducting, silencers and air cleaner package. Figure 1 illustrates these oscillations with respect to the timing of the inlet valve.

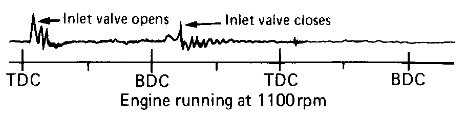

Figure 1: Inlet Noise Oscillogram[1].

The second process is flow generated, or gas flow noise, which is the result of turbulence of the mean flow traveling across the valve seat. This high velocity flow generates a high frequency broad spectrum noise above 1000 Hz. This noise, however, is effectively attenuated by the air cleaner and transmission path between the engine and passenger compartment. Consequently, this flow generated noise does not warrant concern [2]

For the purpose of this study, the propagation of noise in the intake system is assumed to be a one-dimensional wave. It has been shown in the past that the consideration of this noise as plane acoustic propagation has been able to provide reliable prediction of automotive intake noise [3].

2.1 The One-Dimensional Wave Equation for Pulse Noise

A brief derivation of the one-dimensional wave equation is presented I this section. The wave equation illustrates the propagation of acoustic pressure fluctuations. The acoustic variables of interest are pressure p, density ρ, and particle velocity u.

Consider a fixed volume of a duct, as shown in Figure 2, with a cross section S and an arbitrary length dx in the x direction.

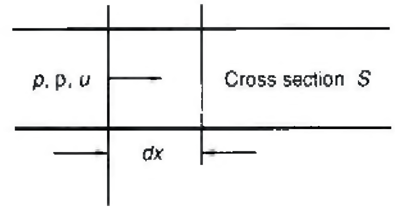

Figure 2. Control Volume with Wave Propagation through a Duct [4]

The total density of the fluid in the control volume in the duct is given as:

$$\rho_{tot} = \rho_o + \rho$$

where, ρ_0 is the initial uniform fluid density and ρ is the change in the fluid density caused by the acoustic wave. Applying the continuity equation to the control volume, the rate of mass inflow to the control volume is given as:

$$\rho_{tot}uS - \left[\rho_{tot}u + \frac{\partial(\rho_{tot}u)}{\partial x}dx\right]S = -\frac{\partial(\rho_{tot}u)}{\partial x}dxS$$

Any increase of mass in the control volume must be balanced by the net inflow of mass given above. After simplification, we have:

$$\frac{\partial(\rho_{tot})}{\partial t} + \frac{\partial(\rho_{tot}u)}{\partial x} = 0$$

Linearizing the above equation and considering only the first order terms, the mass continuity reduces to,

$$\frac{\partial \rho}{\partial t} + \rho_0 \frac{\partial u}{\partial x} = 0$$

One can also apply the principal of momentum conservation to the same fluid element and using the argument above, express the net pressure in the fluid as:

$$p_{tot} = p_o + p$$

where, p represents the acoustic pressure. The net force acting on the element in the x direction is then:

$$p_{tot}S - (p_{tot} + \frac{\partial p_{tot}}{\partial x}dx)S = -\frac{\partial p_{tot}}{\partial x}dxS$$

Linearizing the above equation and considering only the first order terms, the momentum continuity reduces to,

$$\rho_0 \frac{\partial u}{\partial t} + \frac{\partial p}{\partial x} = 0$$

Differentiating the linearized equation of mass conservation with respect to time and the linearized equation of momentum conservation with respect to position we get:

$$\frac{\partial^2 p}{\partial x^2} - \frac{\partial^2 \rho}{\partial t^2} = 0$$

The acoustic wave propagation is assumed to be adiabatic where a change in pressure is directly related to a corresponding change in density by a proportionality constant of the square of the speed of sound 'c'. Substitution of this into the previous equation gives the one-dimensional wave equation for the propagation of acoustic pressure fluctuations in the manifold duct.

$$\frac{\partial^2 p}{\partial x^2} - \frac{1}{c_0^2}\frac{\partial^2 p}{\partial t^2} = 0$$

This relationship illustrates the way that acoustic pressure fluctuations act with respect to the co-ordinate distance x and with respect to time.

3. CONVENTIONAL CONTROL METHODS

Before presenting the non-conventional noise cancellation technique, review of some of the traditional methods of intake noise attenuation and active noise control (ANC) are discussed below.

3.1 Passive Attenuation

Automotive intake noise is traditionally attenuated through the application of passive control techniques. These techniques are usually the simplest and least expensive form of attenuation but do not always yield the best results. Two primary categories of passive noise control are: 1) path redirection of the acoustic energy; and 2) reduction of the acoustic energy flow, usually through either absorption with acoustic insulation or by changing the acoustic impedance of the power output, perhaps through the use of a sudden cross section change.

The Helmholtz resonator is one of the most common passive noise control technique used in automotive induction systems. When acoustic energy travels down a tube or pipe, a specifically chosen attached volume can be used to attenuate the traveling noise. This technique is particularly effective

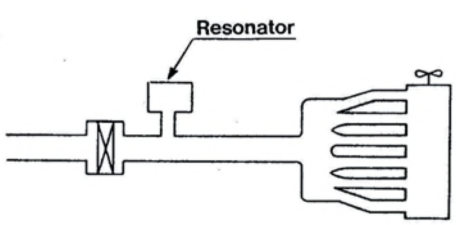

Figure 3. Resonator in Automotive Intake System [5].

when the unwanted noise consists of a narrow frequency band and the volume, or resonator, is tuned to the target frequency. A schematic of a typical resonator in an automotive induction system is given in Figure 3.

Another common passive noise control technique is the use of an expansion chamber. These may or may not include absorptive elements. An expansion chamber without the presence of absorbing material is called a reactive muffler. Here, the performance of the muffler is dependant entirely on the geometrical shape of the expansion chamber. A cutaway of a multi-chambered muffler is illustrated in Figure 4. If sound absorbing material is integral to the attenuating abilities of the muffler, it is referred to as a dissipative muffler. Such designs are generally best suited for controlling frequencies higher than 500 Hz.

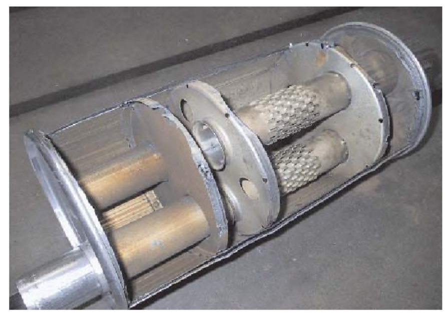

Figure 4. Cutaway of Multi-Chambered Muffler.

3.2 Active Attenuation

Active noise control (ANC) attenuates unwanted noise by canceling the unwanted noise waves through the introduction of a second set of noise waves which are equal in amplitude but opposite in phase to the undesired acoustic signal. The traditional method of ANC most often utilizes loudspeakers to generate a sound field to cancel the existing sound field.

The most commonly found types of active noise control systems are the adaptive feedforward, the adaptive feedback and the wave synthesis system. A typical adaptive feedforward active noise control system is shown in Figure 5.

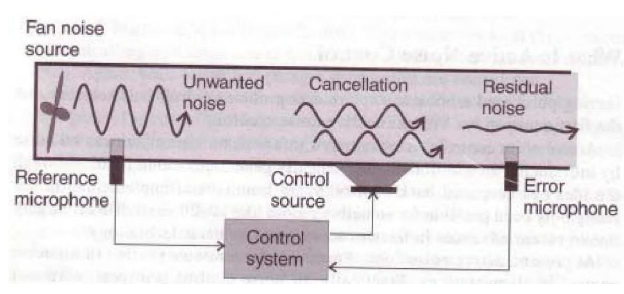

Figure 5. Schematic of Adaptive Feedforward Active Noise Control System [6].

Noise propagation through an automotive intake system can be considered synonymous to the propagation of a sound field through a duct. Here, a duct is simply considered to be an enclosure where one of the dimensions of the enclosure is very long. This enclosure, most often, terminates into open space where in the case of an automotive induction system, the termination would be the air intake opening.

One application of ANC in an automotive intake system (McLean paper 2001-01-1613) used a source coupling technique to control the automotive intake noise. This was accomplished by placing a conventional loudspeaker co-axially inside the air intake. Here, the speaker diaphragm was co-planar with the termination of the air intake duct. A sketch of the speaker system inside the intake duct is shown in Figure 6.

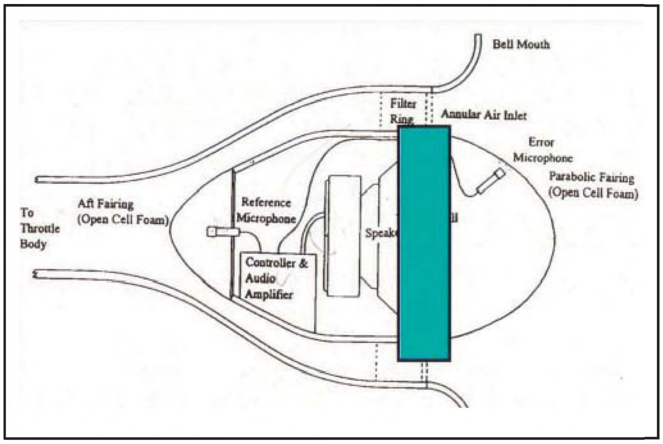

Figure 6. Sketch of Air Induction ANC via Source Coupling [7]

It was found that placing the speaker in the co-axial configuration provided much more attenuation when compared to configurations where the speaker was aligned with the pipe axis, with the speaker pointed into the pipe inlet or where the speaker and pipe axes were orthogonal.

4. THEORETICAL MODEL

The results of this work are the culmination of a theoretical modeling investigation and hence, a model of the original unmodified engine first had to be created.

4.1 Modelling Software

The modelling software program used for this investigation is called Ricardo WAVE. "WAVE is a computer-aided engineering code developed by Ricardo to analyze the dynamics of pressure waves, mass flows and energy losses in ducts plenums and the intake and exhaust manifolds of various systems" [8]. This is accomplished by applying a one-dimensional finite difference approach of the theoretical thermo-fluid equations of the working fluids of the defined system. First, a representation of the subject components must first be synthesized to a rendering of the subject engine. In order to facilitate this, specific information about the engine must be obtained. This information is comprised of three categories: i) geometric data; ii) engine data; and iii) the operating conditions.

The geometric data required to completely model the engine includes the physical dimensions of the intake and exhaust manifold duct lengths, port sizes and air box volumes. These lengths and volumes play an important role in the determining the performance characteristic of the engine. Given this, the geometric dimensions used in the model must be as accurate as possible to ensure the most accurate numerical results. Even the component materials and surface finishes will influence the effects of wall friction. Without these considerations, actual induced flow losses may not be realized in the modeled results.

The engine data is the quantitative information associated with the engine block and cylinder head. This would include valve diameters, timing and lift profiles, and complete port flow coefficients. The bore, stroke, connecting rod length and pin offset, along with the compression ratio, firing order and frictional details are required information of the engine block.

Information of the operating conditions of the engine is also required for the simulation model to reach steady state condition. The better the initial operating information is, the more capable the simulation will be able to quickly and accurately reach its final results. Some of the operating conditions required are the inlet and exhaust wall temperature, operating speed, head, piston and cylinder temperatures. Further requirements include the ambient conditions and combustion information.

4.2 Model Design

In order to analyze an engine with WAVE, it must first be created with the preprocessor WAVEBUILD. This canvas provides the ability to create and synthesize all of the building blocks representing the various ducts, volumes and other engine component. WAVEBUILD also allows for the input of the required physical data and operating conditions of the engine.

Figure 7 provides an illustration of the unmodified model of the engine used in this investigation. By unmodified,

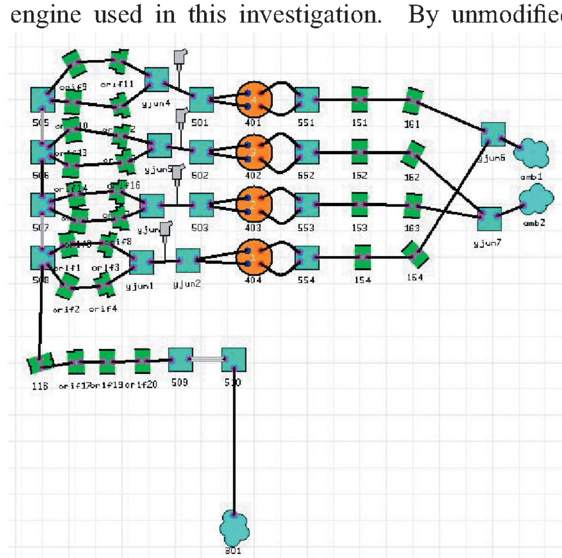

Figure 7. Unmodified WAVEBUILD Engine model.

it is meant that this engine is the original design prior to the implementation of any manifold bridge. The engine modeled is based on a Toyota 4A-GE used in the North American MR2 Mark I and Corolla GTS applications. The engine configuration is a 16 valve inline 4 cylinders with a displacement of 1587 cc and a compression ratio of 10:1. This model provided the reference acoustical performance information of the motored engine that was used to evaluate the results of the implementation of the manifold bridge.

Figure 8 is a schematic of the front end of the intake system illustrated with a corresponding WAVE model for the shown components. The inlet snorkel, which is opened to the ambient conditions, is attached to the airbox where the air filter is housed. The airbox is modeled as two cylindrical volumes joined at the air filter element which is represented by the hatched line in the figure. The air filter is modeled as a zero length duct, with a perforated obstruction. The zip tube exits the airbox and is connected to the throttle body. The diameter of the throttle is set as a variable and can be adjusted to represent different engine loading conditions.

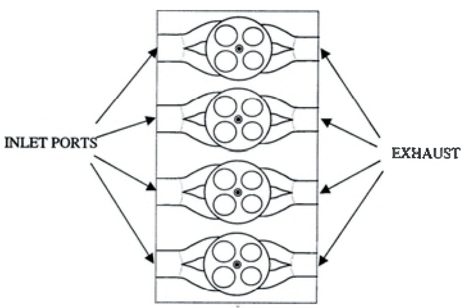

Figure 8. Sub-system of Intake Front End [9].

Figure 9 is a representation of a simplified unmodified intake manifold. The actual intake manifold used in this study is more complex in that each cylinder has duel runners, however, this representation provides a good understanding of the modelling procedure used. This simplified manifold consists of a plenum with four runners. The inlet to the plenum is the throttle body discussed above. The runners are modeled by several individual ducts so as to accurately represent the changes in cross sectional area and bends.

The next sub-section of the engine model created

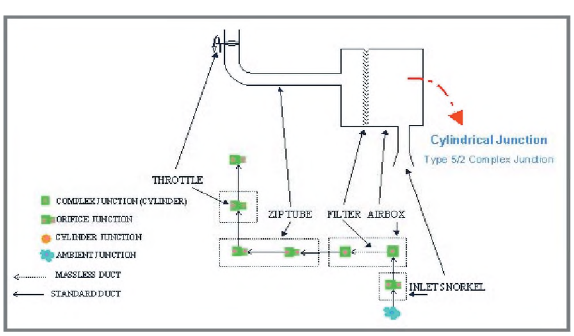

Figure 9. Intake Manifold System [9].

is the cylinder head as represented in Figure 10. As already stated, the engine is a 16 valve, four-cylinder engine with four valves per cylinder. The diameters and lift information of the intake and exhaust valves is input into the model. All losses related to the ports are taken into account through the specification of flow coefficients.

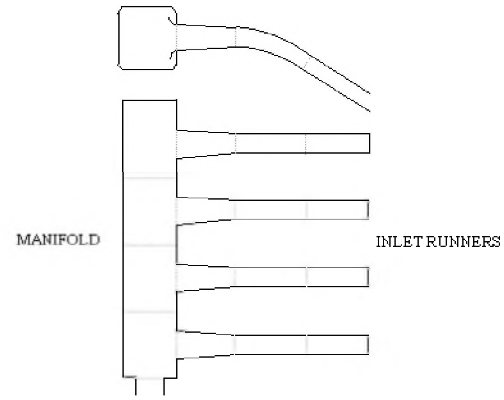

Figure 10. 16 Valve Cylinder Head [9].

The configuration of the exhaust manifold for this engine is a four to two arrangement. In other words, there are four exhaust runners which join to become two runners which subsequently meet to become a single outlet. A more simplified four to one is shown in Figure 11. Care must be taken in the modelling process to ensure that the angles and dimensions of the ducts are carefully represented in the model as they play an essential role in determining the dynamic behaviour of the exhaust system.

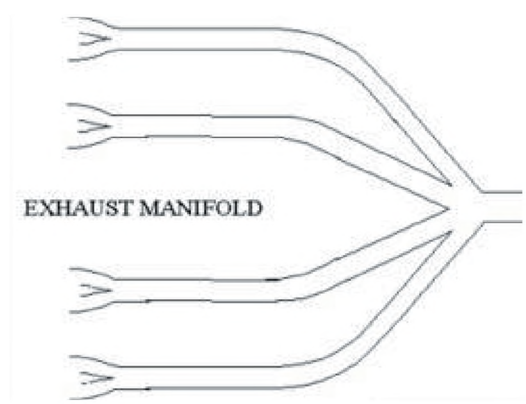

Figure 11. Exhaust Manifold System [9].

4.3 Model Outputs

The purpose of this work is to validate the feasibility of using a manifold bridge to improve the acoustical performance of an automotive intake system. The success of this investigation is determined by both the realized attenuation due to the implementation of the bridge as well as any improvement in the measured sound quality. This is accomplished through application of several psychoacoustic metrics.

The traditional acoustical parameters reported in this paper to measure the attenuation are all measured at a position 0.1 meters from the intake opening of the engine. This is an industry standard. Here, the overall linear and A-weighted sound pressure level and frequency spectra are measured for various steady rpm's of the engine's operating range. Colour maps are also determined for transient runs over the rpm range of the engine.

In addition to the traditional acoustical parameter, several psychoacoustic metrics were employed to measure the effectiveness of the manifold bridge. The metrics used included loudness, sharpness, fluctuation strength and roughness and are described below.

Zwicker Loudness is a standardized metric that describes the human perception of loudness instead of simply a reported sound pressure level. This value takes into account the temporal processing of sounds as well as audiological masking effects [10]. The unit of loudness is sones and is given across Bark, or critical, bands, as opposed to frequency bands.

Sharpness, which has units of acum, describes the high frequency annoyance of noise by applying a weighting factor on sounds above 2 kHz. This overall measurement is useful for such sounds as broadband sources, wind or rushing air noise and gear meshing or grinding sounds. Given that a high frequency component of intake noise is created by the intake air traveling across the valve seat at a high velocity, sharpness is an appropriate metric for the evaluation of the merits of the manifold bridge.

Fluctuation strength and roughness are both metrics used to describe the annoyance of modulating sounds depending on the frequency of the modulation. The fluctuation strength focuses on sounds which modulate at frequencies between 0.5 Hz and 20 Hz, with 4 Hz being the most annoying fluctuation. The unit of amplitude for fluctuation strength is the vacil. Roughness focuses on noise which is modulating at frequencies between 20 Hz and 300 Hz, with the most annoying modulation being 70 Hz. The unit of amplitude for roughness is the asper. When sounds modulate faster than 300 Hz, the human ear is not be able to distinguish this from a normal pure tone. Examples of modulating sources include beating sounds, sirens and fan blades.

4.4 Optimization of the Model

In order to achieve both the greatest attenuation and improvement in sound quality as a result of the insertion of the proposed manifold bridge, the physical parameters of the bridge needed to be determined. In other words, the configuration of the bridging runners as well as their lengths and diameters needed to be calculated.

Several bridging configurations were investigated and evaluated with respect to overall noise attenuation. The first included a single bridge from the exhaust manifold output to the intake manifold plenum. Secondly, bridging ducts running from exhaust manifold runners to their corresponding intake manifold runners were looked at. In other words, a bridging duct attached to the exhaust runner associated with cylinder number one would be similarly attached to the intake runner, also for cylinder number one. The third configuration looked at the outcome of attaching the bridging runners to each of

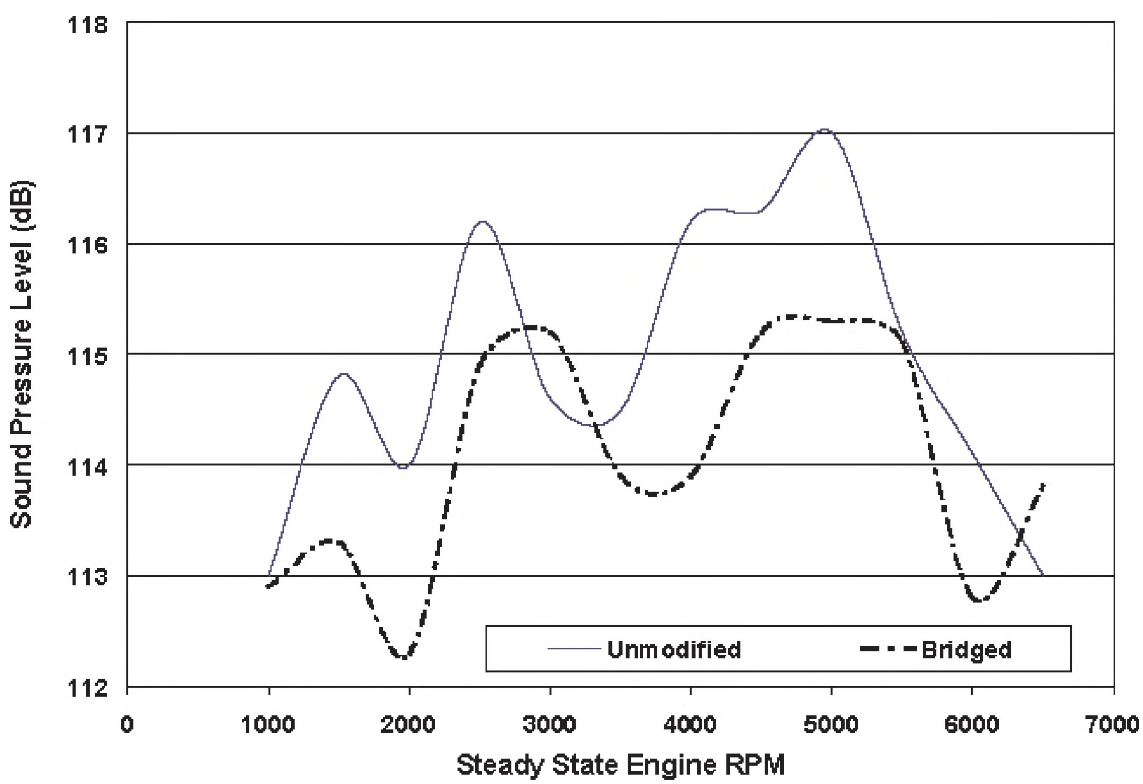

Figure 12. Steady State Intake Noise of Unmodified and Bridged Engines.

the four exhaust manifold runners and routing them to the corresponding intake runners which were associated with cylinders that were 180 degrees out of phase with respect to the firing order of the engine.

Using the results from the three alternatives described above, it was determined that the second approach provided the best attenuation at the measurement position outside the air induction inlet. That is, the configuration where the bridging ducts were linked from the exhaust manifold runners to their corresponding intake manifold runners achieved the greatest noise reduction. Using this configuration, the physical parameters were further optimized. The length of the duct was varied from the minimum physical length possible to a maximum of approximately double. While it was found that additional length for some runners proved to have additional attenuating characteristics, the investigation was limited to double the original length for practical purposes. The diameter of the bridging ducts were also varied from 18 to 32 mm in order to find an optimal cross sectional area. This range was again restricted to practical sizes.

5. DISCUSSION OF RESULTS

Once the feasibility range of the physical characteristics was determined, more detailed acoustic analyses were carried out. Specifically, the noise attenuation between the original and bridged engines was determined for steady state conditions for engine speeds from 1000 to 6500 rpm. Similar analyses were performed for transient runs between the same rpm range. Sound quality analyses were also carried out on the steady state and transient cases. Figure 12 is an illustration of the modeled sound pressure levels for the steady state conditions for engine speeds ranging from 1000 to 6500 rpm. The two shown curves represent the sound pressure levels for the case of the unmodified engine along with the case of engine modified with the manifold bridge. It can be seen that for the entire operating range of the engine, with the exception of between approximately 2800 to 3400 rpm, the bridged engine is quieter then the original unmodified engine. It was found that an attenuation of up to approximately 2.5 dB at around 4000 rpm was realized with the implementation of the bridge over the unmodified engine. Only at a speed of approximately 3000 rpm was the bridge found detrimental to the acoustic performance of the engine with an increase in sound level of approximately 0.8 dB.

Figures 13 and 14 further emphasize the apparent attenuation for steady state conditions between the original and modified engines. These figures show the frequency spectra of the two engines at engine speeds of 2000 rpm and 4000 rpm respectively. Again, it can be seen that the curves representing the bridged engine demonstrate lower amplitudes for much of the frequency spectrum.

In addition to the steady state simulations, acoustic tests were also carried out on transient simulation runs of the two engine models. Like the steady state simulations, the transient runs were from 1000 to 6500 rpm. It was found that an overall sound reduction of 5.6 dB was achieved with the implementation of the manifold bridge. Figures 15 and 16 are colour map representations of the induction noise during these transient simulations. Figure 15 shows the frequency of the intake noise for the rpm range of the unmodified engine. Here, the various colours represent the amplitude of the predicted sound pressure level.

Similarly, Figure 16 illustrates the same for the engine modified with the manifold bridge. The acoustic shortcomings of the unmodified engine are incontrovertibly obvious. The yellow and orange streaks representing the fundamental and subsequent harmonic frequencies are more apparent with more red showing on the map of the unmodified engine. This shows higher amplitudes of sound at the fundamental frequencies which are obviously associated with the speed of the engine. Also, the bridged engine simulation has less of the higher sound pressure level represented by the green colour. Similarly, it has more of the lower sound pressure represented by the mid and dark blue shades.

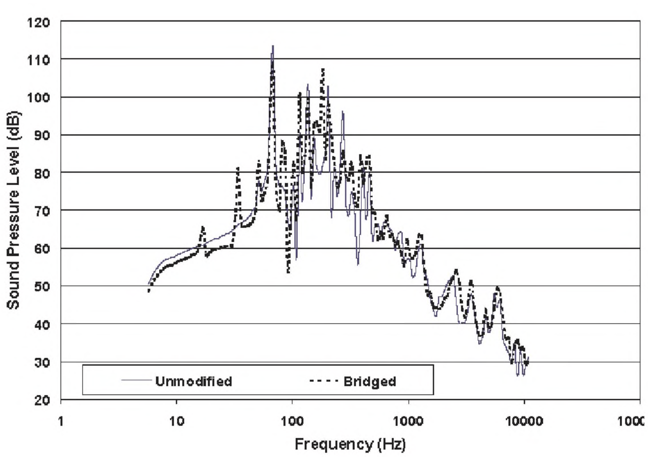

Figure 13. FFT of Both Modeled Engines at 2000 rpm.

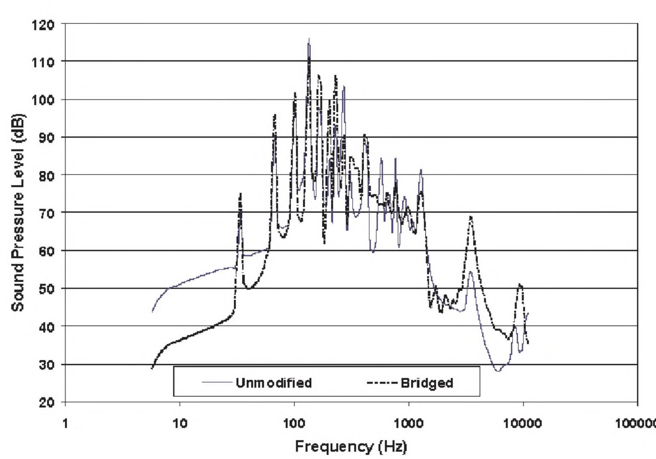

Figure 14. FFT of Both Modeled Engines at 4000 rpm.

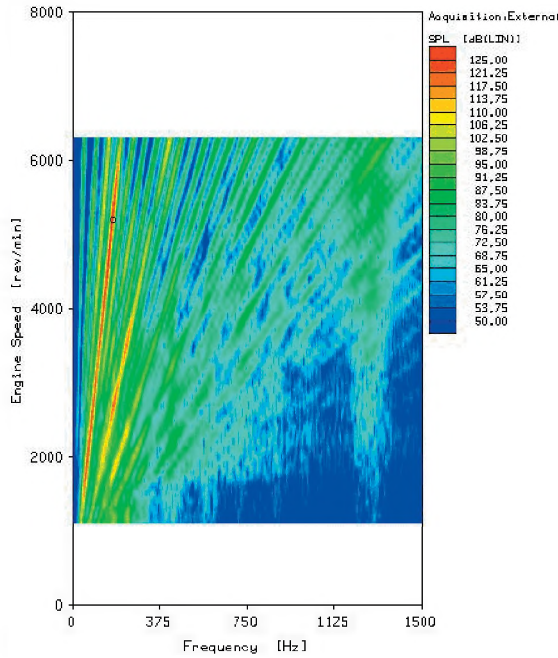

Figure 15. Colour Map of Intake Noise of Unmodified Engine.

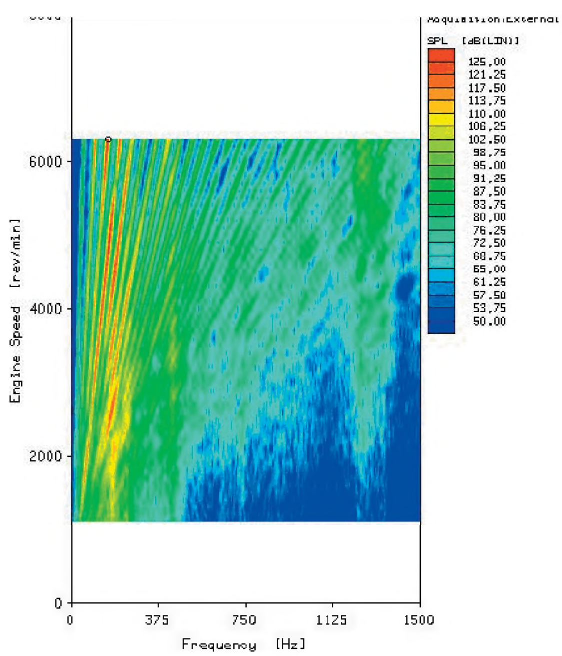

Figure 16. Colour Map of Intake Noise of Bridged Engine.

In addition to the traditional methods of acoustic analysis, psychoacoustic metrics were applied to quantify the differences in sound quality of the two engines. The results of this sound quality analysis are illustrated in Table 1.

Sound Quality Results		
	Unmodified	Bridged
Loudness (sones)	155.07	120.30
Sharpness (acum)	0.53	0.65
Fluctuation Strength (vacil)	1.80	1.44
Roughness (asper)	2.51	3.49

Table 1. Sound Quality Comparison of Unmodified and Bridged Engine.

Perhaps the largest difference between the two engines is shown by the loudness reduction of approximately 35 sones. While this metric is spectrally influenced in that it loosely follows the attenuation realized by the A-weighting curve over the frequency range of interest, one advantage of the overall loudness is that it also includes the influence of masking and temporal effects. Given this, the advantages of the bridged engine are more apparent.

An increase in sharpness is shown with the bridged engine which indicates that an increase in the frequency content of the signal is present in the range above 2000 Hz.

The decrease in fluctuation strength from 1.8 to 1.44 vacil shows that a drop in low frequency modulation is present. This concept is reinforced by the fact that there is an increase in roughness with the bridge engine results. This suggests that the loss of low frequency modulation was just simply a shift to a higher frequency, somewhere above 20 Hz.

6. CONCLUSIONS

For the conditions investigated, it has been shown that the implementation of the manifold bridge has a positive influence on both the amplitude and the sound quality of induction noise. While this investigation was limited to the presentation of the theoretical modelling results, experimental verification is also being pursued. The focus of this presentation was the realized acoustical results of the bridge implementation and did not report on some of the other engine performance criteria. It should be realized that the addition of the manifold bridge will affect some of these other criteria, particularly the influence of the additional exhaust gas recirculation. However, if this was found to be detrimental to the engine performance, the exhaust and intake systems could be isolated from each other through the addition of a membrane or a dual walled bladder system. This investigation, however, does demonstrate the merits in pursuing further refinements of this unique noise control approach.

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Architectural Acoustics, Principles and Practice
Edited by W. J. Cavanaugh and Joseph A. Wilkes, Pages 332; John Wiley and Sons, Inc., 1999 - ISBN 0-471-30682-7, US$99.00

As someone who has taught a 13-week long architectural acoustics course to undergraduate architecture students for the past two decades (the course is now shortened to 6 weeks due to the proverbial budget cuts), I always look to new publications that aid the instructor to teach a complex, mathematical subject to audiences without advanced mathematical background. Prof. David Egan's book, "Architectural Acoustics," has sufficed so far. Now, I can add the above book, so ably edited by Cavanaugh and Wilkes to my list. Both of these editors have excellent credentials to undertake the difficult process of producing an architectural acoustics book. They have assembled a team of eight experts to write the six chapters that comprise the bulk of the book.

The six chapters of the book are: 1) Introduction to Architectural Acoustics and Basic Principles (William J. Cavanaugh); 2) Acoustical Materials and Methods (Rein Pirn); 3) Building Noise Control Applications (Gregory C. Tocci); 4) Acoustical Design: Places for Listening (L. Gerald Marshall and David L. Klepper); 5) Sound Reinforcement Systems (J. Jacek Figwer); 6) Recent Innovations in Acoustical Design and Research (Gary W. Siebein and Bertram Y. Kinsey, Jr.). Each chapter ends with at least one case study which highlights the material presented in the chapter.

Chapter 1 is a true introductory chapter. Through brief, but well-written short paragraphs, the reader quickly traverses a wide canvas of acoustical terminology such as decibel scale, frequency bands, and weighting networks and subjects such as sound propagation, sound absorption, sound transmission and sound control. The various criteria to be used in different contexts are highlighted next with a glossary of applicable standards. The chapter ends with an important and interesting study of Fogg Art Museum Lecture Hall at Harvard and its acoustical problems. It is interesting to note that Wallace Clement Sabine was the acoustician in charge of the above project.

Chapter 2 deals with acoustical materials. Materials used for sound attenuation such as silencers and barriers as well as those used for sound absorption are described in detail in this chapter. After describing the general properties of the acoustical materials, the chapter focuses on material through three different categories: a) common building materials; b) acoustical materials; and c) special devices like silencers, springs, elastomers, door seals and spring hangars. The chapter ends with the requisite performance tables of absorption coefficients and transmission loss values as well as a case study. Chapter 3 is a natural progression from Chapter 2 as it deals with the building noise control applications that use the materials described in Chapter 2. Various control techniques as well as applicable noise criteria for buildings are described. There is also a section on HVAC noise control for buildings.

Chapter 4 describes the acoustical design of places used for listening. A brief introduction of both outdoors as well as indoors sound propagation is presented first. The chapter is then divided into three main areas: concert halls, opera houses and general places. The acoustical measures, both objective measures such as reverberation as well as measures which describe subjective attributes (loudness, clarity, intimacy, warmth, etc.) are highlighted for each of the three major areas. The required specification values, both quantitatively as well as qualitatively to describe the acoustics of the listening spaces, are briefly explained for different spaces with varying shapes and sizes. The requirements, even for spaces such as sports facilities and home listening rooms are presented. Three case studies are presented to highlight the acoustical design features described in this chapter.

Chapter 5 discusses sound reinforcement systems. Even though the chapter presents a good overview of system components such as microphones, speakers and amplifiers and their different arrangements, this is the weakest chapter in the book. The main reason for this is that the chapter assumes that sound system is a must and does not adequately provide the system's need in its relationship to existing room acoustical character. There is no proper description of the room acoustics and its conflict with sound systems. The sound system requirements for different user spaces are described briefly, but once again not in the context of basic acoustic design of the rooms. The description in the case studies, similarly, suffers from lack of reasons for the said design details. In that sense, the description in the case studies becomes perfunctory.

Chapter 6 is the main reason for buying this book. Gary Siebein and Bertram Kinsey Jr. have provided a clear and concise description of the recent innovations in design and research in architectural acoustics. The chapter is divided into three main areas: room acoustical qualities; modeling and aural simulations; and new directions in building acoustics. The material covered in each of the three sections is thorough. The thoroughness can only be proved by enumerating the topics covered under the third section - New Directions. They are: sound diffusing materials; active noise control; sound isolation testing; plumbing system noise control; impact noise; and the impact of noise control methods on health and energy issues. Four case studies are included to showcase the materials presented in this chapter.

Finally, the book includes as one of its appendices, the guidelines for the selection of an acoustical consultant. Further, the case studies like the Fogg Art Museum are the salient features of this book.

Ramani Ramakrishnan, Ph. D., P. Eng.
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ClassTalk

Classroom Sound Level Prediction Software - 2003 - Cd$500 from University of British Columbia

A classroom is a mixture of many acoustical conditions such as moving and stationary sound sources, absorbing and/or rigid room envelopes (such as floors, walls and hanging panels), and intruding ambient sound levels. In noisy classroom environments, the teachers and principals would always be interested in the noise character of their classrooms. In addition, the noise control engineer would also be interested in knowing the distribution of sound levels around the rooms. Speech intelligibility becomes the most important acoustical descriptor.

ClassTalk, a sound level prediction software written by University of British Columbia professor Murray Hodgson, provides a valuable tool for the above-stated purposes. The intent of the software, atleast as per the understanding of this reviewer, is to provide a novel hardware and software for predicting, visualizing and auralizing classroom noise as one (teacher, principal and/or students) walks through the room floor. A strong acoustical background is not a prerequisite. The software, in the event, does provide adequate opportunities to realize the simulated walk-through. Its stated aim is to provide a simple, fast, accurate and user-friendly tool for evaluating the acoustical quality for speech, and sound-control measures, in typical classrooms, both objectively and subjectively.

The software is based on empirical models and data from actual simulations where image-source models of sound prediction were used. The software was easily usable with simple (for the most part) input requirements, such as room dimensions, the envelope acoustical details and source sound power levels. A special sound card (that has sound fonts) is needed to auralize the sound levels as one undertakes the simulated walk-through.

The rooms must be rectangular in shape with data limited to: length between 4.4 m to 24 m; room floor area between 36 and 485 sq.m.; height between 2.2 to 8.0 m; and average surface absorption coefficient between 0.06 and 0.29. The software predicts, for a given source sound power, five quantities: A-weighted background noise level, A-weighted speech level, A-weighted signal-to-noise level difference, Speech Transmission Index (STI), and speech intelligibility. *ClassTalk* also displays qualitative descriptor from Excellent (E) to Bad (B).

ClassTalk is very valuable, except for the following shortcomings. The software limits the sound absorption values to 0.29 or less. Also it deals with a two-dimensional representation of the room floor. A three-dimensional representation of the classroom as one walks through it would be visually more appealing. In addition, only rectangular room shapes are allowed by *ClassTalk*. A user would have liked to use different room shape options. At least, some adjustment procedures could have been provided to include models other than rectangular shaped rooms. *ClassTalk*, however, is a simple tool that can be valuably used by noise control engineers and school boards to visually and aurally present the effect of control measures. The cost is very reasonable.

Prof. Ramani Ramakrishnan, Ph. D., P. Eng.
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Ryerson University, Toronto, ON
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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 stevenb@aciacoustical.com

2004

17-19 March: Spring Meeting of the Acoustical Society of Japan, Atsugi, Japan. Fax: +81 3 5256 1022; Web: wwwsoc.nii.ca.jp/asj/index-e.html

22-25 March: Joint Congress of the French and German Acoustical Societies (SFA-DEGA), Strasbourg, France. Fax: +33 1 48 88 90 60; Web: www.sfa.asso.fr/cfa-daga2004

23-26 March: International Conference: Speech Prosody 2004, Nara Japan (K. Hirose, School of Frontier Sciences, University of Tokyo, 7-3-1 Hongo, Bunkyo-ku, Tokyo 113-0033, Japan; Fax +81 3 5841 6648; Web: www.gavo.t.u-tokyo.ac.jp/sp2004)

31 March – 3 April: International Symposium on Musical Acoustics (ISMA2004), Nara, Japan. Fax: +81 774 95 2647; Web: www2.crl.go.jp/jt/al32/isma2004

4-9 April: 18th International Congress on Acoustics (ICA2004), Kyoto, Japan. Web: ica2004.or.jp

11-13 April: International Symposium on Room Acoustics (ICA2004 Satellite Meeting), Hyogo, Japan. Fax: +81 78 803 6043; Web: rad04.iis.u-tokyo.ac.jp

8-10 May: 116th AES Convention, Berlin, Germany. (Web: aes.org/events/116)

10-12 May: 10th AIAA/CEAS AeroAcoustics Conference, Manchester, UK. Web: www.aiaa.org

17-21 May: International Conference on Acoustics, Speech, and Signal Processing (ICASSP 2004), Montreal, Canada. Web: www.icassp2004.com

24-28 May: 75th Anniversary Meeting (147th Meeting) of the Acoustical Society of America, New York, NY. 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

6-9 June: 13th International Conference on Noise Control, Gdynia, Poland. Web: www.ciop.pl/noise_04

8-10 June: Joint Baltic-Nordic Acoustical Meeting, Mariehamm, Åland, Finland. Contact: Acoustical Society of Finland, Helsinki University of Technology, Laboratory of Acoustics and Signal Processing, P.O. Box 3000, 0215 TKK, Finland; Fax: +358 09 460 224; e-mail: asf@acoustics.hut.fi

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 courriel à stevenb@aciacoustical.com

2004

17-19 mars: Rencontre de printemps de la Société japonaise d'acoustique, Atsugi, Japon. Fax: +81 3 5256 1022; Web: wwwsoc.nii.ca.jp/asj/index-e.html

22-25 mars: Congrès combiné des Sociétés française et allemande d'acoustique (SFA-DEGA), Strasbourg, France. Fax: +33 1 48 88 90 60; Web: www.sfa.asso.fr/cfa-daga2004

23-26 mars: Conférence internationale : la parole prosodie 2004, Nara Japan (K. Hirose, School of Frontier Sciences, University of Tokyo, 7-3-1 Hongo, Bunkyo-ku, Tokyo 113-0033, Japan; Fax +81 3 5841 6648; Web: www.gavo.t.u-tokyo.ac.jp/sp2004)

31 mars – 3 avril: Symposium international sur l'acoustique musicale (ISMA2004), Nara, Japon. Fax: +81 774 95 2647; Web: www2.crl.go.jp/jt/al32/isma2004

4-9 avril: 18e Congrès international sur l'acoustique (ICA2004), Kyoto, Japon. Web: ica2004.or.jp

11-13 avril: Symposium international sur l'acoustique des salles (Rencontre satellite de ICA2004), Hyogo, Japon. Fax: +81 78 803 6043; Web: rad04.iis.u-tokyo.ac.jp

8-10 mai: 116e Convention sur AES, Berlin, Germany. (Web: aes.org/events/116)

10-12 May: 10e Conférence d'AIAA/CEAS AeroAcoustics, Manchester, UK. Web: www.aiaa.org

17-21 mai: Conférence internationale sur l'acoustique, la parole, et le traitement de signal (ICASSP 2004), Montréal, Canada. Web: www.icassp2004.com

24-28 mai: 75e rencontre anniversaire (147e rencontre) de l'Acoustical Society of America, New York, NY. 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

6-9 June: 13e Conférence internationale sur Noise Control, Gdynia, Poland. Web: www.ciop.pl/noise_04

8-10 juin: Rencontre acoustique jointe Baltique-Nordique, Mariehamm, Åland, Finlande. Info: Acoustical Society of Finland, Helsinki University of Technology, Laboratory of Acoustics and Signal Processing, P.O. Box 3000, 0215 TKK, Finland; Fax: +358 09 460 224; courriel: asf@acoustics.hut.fi

5-8 July: 7th European Conference on Underwater Acoustics ECUA 2004, Delft, The Netherlands. Contact: Debbie Middendorp, Secretariat of the 7th European Conference on Underwater Acoustics ECUA 2004, D'Launch Communications, Forellendaal 141, 2553 JE The Hague, The Netherlands; Tel.: +31 70 3229900; Fax: +31 70 3229901; E-mail: middendorp@dlaunch.nl

5-8 July: 11th International Congress on Sound and Vibration, St. Petersburg, Russia. Web: www.iiav.org

11-16 July: 12th International Symposium on Acoustic Remote Sensing (ISARS), Cambridge, UK. Contact: S. Bradley, School of Acoustics and Electronic Engineering, Brindley Building, Room 301, University of Salford, Salford M5 4WT, UK; Fax: +44 161 295 3815; Web: www.isars.org.uk

3-7 August: 8th International Conference of Music Perception and Cognition, Evanston, IL. Contact: School of Music, Northwestern Univ., Evanston, IL 60201; Web: www.icmpc.org/conferences.html

22-25 August: Inter-noise 2004, Prague, Czech Republic. Web: www.internoise2004.cz

23-27 August: 2004 IEEE International Ultrasonics, Ferroelectrics, and Frequency Control 50th Anniversary Conference, Montreal, Canada. Contact: R. Garvey, Datum, 34 Tozer Road, Beverly, MA 01915-5510; Fax: +1 978 927 4099; Web: www.ieee-uffc.org/index2-asp

30 August – 1 September: Low Frequency 2004, Maastricht, The Netherlands, Contact: G. Leventhall, 150 Craddlocks Avenue, Ashtead, Surrey KT 21 1NL, UK; Web" www.lowfrequency2004.org.uk.

13-16 September: Subjective and Objective Assessment of Sound, Poznan, Poland. Contact: Institute of Acoustics, Adan Mankiewicz University, Poznan, Poland. Web: www.ia.amu.edu.pl/index.html

13-17 September: 4th Iberoamerican Congress on Acoustics, 4th Iberian Congress on Acoustics, 35th Spanish Congress on Acoustics, Guimarães, Portugal. Contact: Sociedade Portuguesa de Acústica, Laboratório Nacional de Engenharia Civil, Avenida do Brasil 101, 1700-066 Lisboa, Portugal; Fax: +351 21 844 3028; E-mail: dsilva@lnec.pt

15-17 September: 26th European Conference on Acoustic Emission Testing. Berlin, Germany. Contact: DGZIP, Max-Planck-Str. 25, 12489 Berlin, Germany. Web: www.ewgae2004.de

20-22 September: International Conference on Noise and Vibration Engineering. Leuven, Belgium. Contact: Fax +32 16 32 29 87. Web:
www.isma-isaac.be/fut_conf/default_en.phtml

28-30 September: Autumn Meeting of the Acoustical Society of Japan. Naha, Japan. Contact: Fax +81 3 5256 1022. Web: www.soc.nii.ac.jp/asj/index-e.html

4-9 October 8th Conference on Spoken Language Processing (INTERSPEECH). Jeju Island, Korea. Web: www.icslp2004.org

6-8 October: Acoustics Week in Canada, Conference of the Canadian Acoustical Association. Ottawa, Ontario. Contact: John Bradley, National Research Council of Canada, Ottawa Ontario. Phone (613) 993-9747, Fax (613-954-1495, web: www.caa-aca.ca

13-16 October: 7th annual Canadian Academy of Audiology Conference, Quebec City. Contact: 250 Consumers Road, Suite 301, Toronto, Ontario M2J 4V6. Phone 1-800-264-5106, Fax (416) 495-8723. web: www.canadianaudiology.ca

4-5 November: Autumn Meeting of the Swiss Acoustical Society, Rapperswil, Switzerland. Contact: SGA-SSA, c/o Akustik, Suva, P.O. Box 4358, 6002 Luzern, Switzerland; Fax: +41 419 62 13; Web: www.sga-ssa.ch

5-8 juillet: 7e Conférence européenne sur l'acoustique sous-marine ECUA 2004, Delft, Pays-Bas. Info: Debbie Middendorp, Secretariat of the 7th European Conference on Underwater Acoustics ECUA 2004, D'Launch Communications, Forellendaal 141, 2553 JE The Hague, The Netherlands; Tél.: +31 70 3229900; Fax: +31 70 3229901; Courriel: middendorp@dlaunch.nl

5-8 July: 11th Congrès internationale sur Sound and Vibration, St. Petersburg, Russia. Web: www.iiav.org

11-16 juillet: 12e Symposium international sur la télédetection acoustique (ISARS), Cambridge, Royaume-Uni. Info: S. Bradley, School of Acoustics and Electronic Engineering, Brindley Building, Room 301, University of Salford, Salford M5 4WT, UK; Fax: +44 161 295 3815; Web: www.isars.org.uk

3-7 août: 8e Conférence internationale sur la perception et la cognition de la musique, Evanston, IL. Info: School of Music, Northwestern Univ., Evanston, IL 60201; Web: www.icmpc.org/conferences.html

22-25 août: Inter-noise 2004, Prague, République tchèque. Web: www.internoise2004.cz

23-27 août: 50e Conférence anniversaire internationale IEEE 2004 sur les ultra-sons, la ferroélectricité et la régulation par la fréquence, Montréal, Canada. Info: R. Garvey, Datum, 34 Tozer Road, Beverly, MA 01915-5510; Fax: +1 978 927 4099; Web: www.ieee-uffc.org/index2-asp

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13-17 septembre: 4e Congrès ibéro-américain d'acoustique, 4e Congrès ibérien d'acoustique, 35e Congrès espagnol d'acoustique, Guimarães, Portugal. Info: Sociedade Portuguesa de Acústica, Laboratório Nacional de Engenharia Civil, Avenida do Brasil 101, 1700-066 Lisboa, Portugal; Fax: +351 21 844 3028; Courriel: dsilva@lnec.pt

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www.isma-isaac.be/fut_conf/default_en.phtml

28-30 septembre: Rencontre d'automne de l'Acoustical Society of Japan. Naha, Japan. Contact: Fax +81 3 5256 1022. Web: www.soc.nii.ac.jp/asj/index-e.html

4-9 octobre 8e Conférence sur Spoken Language Processing (INTERSPEECH). Jeju Island, Korea. Web: www.icslp2004.org

6-8 octobre: Congrès annual de l'Association canadienne d'acoustique. Ottawa, Ontario. Contact: John Bradley, National Research Council of Canada, Ottawa Ontario. Phone (613) 993-9747, Fax (613-954-1495, web: www.caa-aca.ca

13-16 octobre: 7e Congrès annual de l'Academy canadian de audiology, Quebec City. Contact: 250 Consumers Road, Suite 301, Toronto, Ontario M2J 4V6. Phone 1-800-264-5106, Fax (416) 495-8723. web: www.canadianaudiology.ca

4-5 novembre: Rencontre d'automne de la Société suisse d'acoustique, Rapperswil, Suisse. Info: SGA-SSA, c/o Akustik, Suva, P.O. Box 4358, 6002 Luzern, Switzerland; Fax: +41 419 62 13; Web: www.sga-ssa.ch

15-19 November: 148th Meeting of the Acoustical Society of America, San Diego, CA. 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

2005

16-19 May: SAE Noise and Vibration Conference, Grand Traverse Resort, Traverse City Michigan. Contact: Mrs. Patti Kreh, SAE International, 755 W Big Beaver Rd, Ste 1600, Troy, Michigan, 48084. Tel: (248) 273-2474, E-mail: pkreh@sae.org

16-20 May: 149th Meeting of the Acoustical Society of America, Vancouver, BC, Canada. 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

6-10 August: Inter-Noise, Rio de Janeiro, Brazil. Details to be announced later.

28 August – 2 September: Forum Acusticum Budapest 2005, Budapest, Hungary. Fax: +36 1 202 0452; Web: www.fa2005.org; E-mail: sea@fresno.csic.es

4-8 September: 9th Eurospeech Conference, Lisbon, Portugal. Contact: Fax: +351 213145843. Web: www.interspeech2005.org

5-9 September: Boundary Influences in High Frequency, Shallow Water Acoustics. Bath, UK (Details to be announced later)

17-21 October: 150th Meeting of the Acoustical Society of America, Minneapolis, Minnesota. 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

2006

26-28 June: 9th Western Pacific Acoustics Conference. Seoul, Korea. Web: www.wespac8.com/WespacIX.html

June: 151st Meeting of the Acoustical Society of America, Providence, Rhode Island. 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

28 November – 2 December: 152nd meeting, 4th Joint Meeting of the Acoustical Society of America and the Acoustical Society of Japan, Honolulu, Hawaii. 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

2007

2-7 September 19th International Congress on Acoustics (ICA2007), Madrid Spain. (SEA, Serrano 144, 28006 Madrid, Spain; Web: www.ia.csic/sea/index.html)

2008

23-27 June: Joint Meeting of European Acoustical Association, Acoustical Society of America, and Acoustical Society of France. Paris, France (Details to be announced later)

15-19 novembre: 148e rencontre de l'Acoustical Society of America, San Diego, CA. 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

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16-19 mai: SAE Noise and Vibration Conference, Grand Traverse Resort, Traverse City Michigan. Contact: Mrs. Patti Kreh, SAE International, 755 W Big Beaver Rd, Ste 1600, Troy, Michigan, 48084. Tel: (248) 273-2474, E-mail: pkreh@sae.org

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6-10 août: Inter-Noise, Rio de Janeiro, Brésil. Information à suivre.

28 août – 2 septembre: Forum Acusticum Budapest 2005, Budapest, Hongrie. Fax: +36 1 202 0452; Web: www.fa2005.org; E-mail: sea@fresno.csic.es

4-8 septembre: 9e Conférence d'Eurospeech, Lisbon, Portugal. Contact: Fax: +351 213145843. Web: www.interspeech2005.org

5-9 septembre: Boundary Influences in High Frequency, Shallow Water Acoustics. Bath, UK (Details to be announced later)

17-21 octobre: 150e rencontre de l'Acoustical Society of America, Minneapolis, Minnesota. 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

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26-28 juin: 9e Conférence Western Pacific Acoustics. Seoul, Korea. Web: www.wespac8.com/WespacIX.html

juin: 151e rencontre de l'Acoustical Society of America, Providence, Rhode Island. 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

28 November – 2 December: 152nd meeting, 4th Joint Meeting of the Acoustical Society of America and the Acoustical Society of Japan, Honolulu, Hawaii. 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

2007

2-7 September 19th International Congress on Acoustics (ICA2007), Madrid Spain. (SEA, Serrano 144, 28006 Madrid, Spain; Web: www.ia.csic/sea/index.html)

2008

23-27 June: Joint Meeting of European Acoustical Association, Acoustical Society of America, and Acoustical Society of France. Paris, France (Details to be announced later)

MEDIA RELEASE

January 28, 2004, Center for Health and Environment Research, University of British Columbia. ClassTalk software for design and assessment of classroom acoustics is available for use. A free demonstration version of ClassTalk can be downloaded at www.flintbox.ca. Contact: Prof. Murray Hodgson, Tel: (604) 822-3073, E-mail: hodgson@interchange.ubc.ca

APPOINTMENTS

Mr. Ralph K. Hillquist of Benzonia, Michigan has been elected to the prestigious status of "SAE Fellow" by the Society of Automotive Engineers International. Mr. Hillquist is Vice President with RKH Consultants and is the founder of the SAE Noise and Vibration Conference (started in 1985).

EXCERPTS FROM "*WE HEAR THAT*", IN *ECHOS*, ASA

- **Leo L. Beranek** is one of eight of the Nation's leading scientists and engineers to receive the 2002 National Medal of Science. The presidential medal is the Nation's highest honor for researchers who have made major impacts in fields of science and engineering through career-long, ground-breaking achievements.

- **David Feit** received the Per Bruel Gold Medal for Noise Control and Acoustics at the 2003 ASME meeting in Washington, DC, 16-21 November.

- A special edition of *Acoustics Research Letters Online (ARLO)* is planned to commemorate the life and work of **Robert E. Apfel**, former ASA president and founder of ARLO. Contributions to this commemorative issue are solicited on acoustics topics in which he was involved, including acoustics education, physical acoustics, cavitation, drop dynamics, acoustic levitation, radiation detection, mixture laws, and the acoustic nonlinear parameter. Former colleagues of Robert Apfel are particularly urged to contribute. Publication is expected during summer 2004.

- **Richard H. Lyon** received the Gold Medal award from the Acoustical Foundation of India. The Foundation presents this award to an internationally renowned acoustician whose work has developed acoustics in the world community. The award was presented at a meeting of the Automotive Research Association of India on October 31, 2003.

- **Kenneth N. Stevens** and **Gunnar Fant** will receive the 2004 James L. Flanagan Speech and Audio Processing Award from IEEE.

AWARDS TO CAA MEMBERS

Dr Stan Dosso has been awarded the Acoustical Society of America 2004 Medwin Prize in Acoustical Oceanography, "For the development of non-linear methods for geoacoustic inversion and acoustic localization." The Medwin prize is awarded in recognition of effective use of sound in the discovery and understanding of physical and biological parameters and processes in the sea. Stan is an Associate Professor from the University of Victoria and a fellow of the Acoustical Society of America. His research interests centre on ocean acoustic/seismic inverse problems, including the inversion of ocean acoustic fields for geoacoustic properties, seismo-acoustic and marine seismic surveys, acoustic localization and positioning, seismic tomography, and sea ice acoustics. He is just returning from a productive sabbatical at the NATO Undersea Research Centre in La Spezia, Italy.

DRDC Atlantic defence scientist Nicole Collison has won the 2003 Nova Scotia Discovery Centre Science and Technology Award Emerging Professional Award. This award goes to the science student/professional who demonstrates ingenious and innovative thinking in the development of unique ideas, theories and processes in their field while also maintaining diverse interests outside of science. Nicole has been with DRDC Atlantic since 1999 where she is part of the Sonar Signal Processing group. Her work includes the use of sonar to detect everything from submarines in coastal waters to Right Whales in the Bay of Fundy. In her short time at DRDC, she has become the leading authority on array element localization, underwater acoustic experimentation, and the optimization of large-dimensional problems. In recognition of her innovative scientific investigations, she received the 1999 CAA Fessenden Student Prize in underwater acoustics and the 2001 CAA Director's Graduate Student Author Award. In addition, in May 2003, the Acoustical Society of America awarded her the Technical Committee on Signal Processing Award for Outstanding Paper by a Young Presenter. She has also been invited to review papers for Canadian Acoustics and to work internationally with NATO groups and with the Technical Cooperation Program, which includes Canada, the US, the UK, Australia and New Zealand.

CAA Annual Conference in Ottawa

October 6-8, 2004

www.caa-aca.ca/ottawa-2004.html

Second Announcement

The 2004 annual conference of the Canadian Acoustical Association will be held in Ottawa October 6-8, 2004. With a location in Ottawa, and the theme 'Acoustics: A National Issue', it should be one of our more significant conferences. You can participate in three days of three parallel sessions of papers on all areas of acoustics. In addition to various associated meetings, there will be tours of local acoustical laboratories. Mark your calendars and plan now to participate!

Special Sessions

We are planning many special sessions including invited and contributed papers. To date, the planned special sessions and organisers include those listed below. Please contact the organiser to participate in one of these or the conference Technical Chair to add other special sessions.

"Outdoor Sound Propagation",
contact: Cameron Sherry, CWSherry@aol.com
"Instrumentation and Measurements",
contact: George Wong, George.Wong@nrc.ca
"Acoustical Materials: Simulation and Characterisation",
contact: Raymond Panneton, Raymond.Panneton@USherbrooke.ca
"Acoustics of Educational Facilities",
contact: John Bradley, John.Bradley@nrc.ca
"Audio Systems and Signal Processing",
contact: Scott Norcross, Scott.Norcross@crc.ca
"Acoustic and Non-Acoustic Influences on Speech Understanding",
contact: Kathy Pichora-Fuller, kpfuller@utm.utoronto.ca
"Hearing Aids",
contact: Vijay Parsa, parsa@nca.uwo.ca
"Hearing and the Workplace",
contact: Christian Giguère, cgiguere@uottawa.ca
"Musical and Ecological Psychoacoustics",
contact: Frank Russo, frusso@utm.utoronto.ca
"Musical Acoustics",
contact: David Gerhard, david.gerhard@uregina.ca
"Underwater Sound",
contact: Nicole Collison, Nicole.collison@drdc-rddc.gc.ca
"Signal Processing Applications",
contact: Dave Havelock, Dave.Havelock@nrc.ca
"Guidelines for Environmental Noise",
contact: Stephen Bly, Stephen_Bly@hc-sc.gc.ca
"Noise Emission Declaration for Machinery Noise",
contact: Stephen Keith, Stephen_Keith@hc-sc.gc.ca

Deadlines

Abstracts	20 June 2004
Two-page papers	15 August July 2004

Plenary Speakers

Two distinguished Canadian acousticians will give plenary lectures coordinated with two of the special sessions above. Final details are being worked out and will be included in our next announcement. Stay tuned for the details to follow.

Associated Events

"What's New in Building Acoustics at IRC-NRC?"
organiser: Dave Quirt (Friday AM, open to all CAA conference attendees)
CSA Z107 Committee Acoustics and Noise Control meeting (Wednesday evening),
contact: Cameron Sherry, CWSherry@aol.com
Lab tours various NRC labs, Health Canada, (Friday PM)

Exhibits

There will be a one and a half day exhibit of measurement equipment and other acoustical products. The exhibit will run all day Thursday and Friday morning (October 7-8). As usual the exhibit area will also be the central coffee break area. Please contact our exhibit coordinator for exhibitor information and sponsorship of various aspects of this meeting.

Student Participation

CAA has a very strong emphasis on encouraging students. Student members of CAA who make presentations can apply for travel support and can win one of a number of student presentation awards. See our website for details.

Submissions and Important Dates

Submissions on all aspects of acoustics are welcomed. The deadline for submission of abstracts is **20 June 2004**. They should be submitted by Email to **abstracts@caa-aca.ca** and should be no more than 250 words and include the usual contact information. Other details of requirements for abstracts will be posted on the CAA website. Notices of acceptance will be sent out shortly after this deadline.
The deadline for the subsequent submission of a two-page summary paper for the conference issue of Canadian Acoustics is **15 August 2004**. If you miss this deadline your two-page summary paper will not be included in the conference issue of Canadian Acoustics. This conference issue has become the archival record of new acoustical research activities in Canada each year. Make sure you are included!

Venue and Accommodation

The conference will be held at the newly renovated and enlarged Lord Elgin hotel, centrally located in Ottawa a few blocks from Parliament Hill. Participants registering with the hotel by September 5, 2004 will receive a room rate of $128/night (single or double). (1-800-267-4298). Please stay at this hotel to be with your friends and to support CAA.

Hospitality

CAA conferences are always an opportunity to meet old friends and to make new ones over a coffee during the conference, or over a drink after the sessions are over. There are many nearby bars and restaurants and of course there will be a banquet as part of the conference. Why not make it a holiday too, and stay on to see the Fall colours in Ottawa and Gatineau?

Contacts

Convenor	John Bradley	(john.bradley@nrc.ca)
Technical Chair	Brad Gover	(brad.gover@nrc.ca)
Publicity	Christian Giguère	(cgiguere@uottawa.ca)
Exhibits	Hugh Williamson	(hughwilliamson@sympatico.ca)
Webmaster	Alf Warnock	(alf.warnock@nrc.ca)
Audio-Visual	Frances King	(frances.king@nrc.ca)

Congrès annuel de l'ACA à Ottawa

6 au 8 octobre 2004

www.caa-aca.ca/ottawa-2004.html

Deuxième avis

Le congrès annuel de l'Association canadienne d'acoustique se tiendra à Ottawa du 6 au 8 octobre 2004. Avec comme site Ottawa et comme thème « L'Acoustique : Une question nationale », il s'agira sûrement d'un congrès des plus mémorables. Trois jours de communications scientifiques comprenant trois sessions parallèles sont prévus sur tous les domaines de l'acoustique. En plus des réunions habituelles, des visites de laboratoires seront au programme. Veuillez planifier dès maintenant de participer à cet événement !

Sessions spéciales

Des sessions spéciales seront structurées autour de conférenciers invités et des communications soumises par les délégués. Si vous désirez participer à l'une des sessions spéciales ci-dessous, veuillez communiquer avec le responsable de session. Pour organiser d'autres sessions spéciales, veuillez communiquer avec le Président du congrès ou le Directeur scientifique.

« Propagation sonore à l'extérieur »
responsable: Cameron Sherry, CWSherry@aol.com

« Instrumentation et Méthodes de mesures »
responsable: George Wong, George.Wong@nrc.ca

« Matériaux acoustiques : simulation et caractérisation »
responsable: Raymond Panneton, Raymond.Panneton@USherbrooke.ca

« L'Acoustique des établissements éducationnels »
responsable: John Bradley, John.Bradley@nrc.ca

« Systèmes audio et traitement du signal »
responsable: Scott Norcross, Scott.Norcross@crc.ca

« Aspects acoustiques et non-acoustiques de la perception de la parole »
responsable: Kathy Pichora-Fuller, kpfuller@utm.utoronto.ca

« Aides auditives »
responsable: Vijay Parsa, parsa@nca.uwo.ca

« Audition et Milieu de travail »
responsable: Christian Giguère, cgiguere@uottawa.ca

« Psychoacoustique musicale et écologique »
responsable: Frank Russo, frusso@utm.utoronto.ca

« Acoustique musicale »
responsable: David Gerhard, david.gerhard@uregina.ca

« Acoustique sous-marine »
responsable: Nicole Collison, Nicole.collison@drdc-rddc.gc.ca

« Applications du traitement du signal »
responsable Dave Havelock, Dave.Havelock@nrc.ca

« Lignes directrices pour le bruit environnemental »
responsable: Stephen Bly, Stephen_Bly@hc-sc.gc.ca

« Déclaration de l'émission sonore des équipements bruyants »
responsable: Stephen Keith, Stephen_Keith@hc-sc.gc.ca

Échéances

Résumés	20 juin 2004
Articles de deux pages	15 août 2004

Orateurs pléniers

Deux acousticiens canadiens renommés feront des présentations plénières dans le cadre de deux des sessions spéciales. Les détails définitifs sont en voie d'être fixés et seront présentés dans notre prochain communiqué.

Événements particuliers

- Quoi de Neuf en Acoustique des Bâtiments à l'IRC-CNRC?
  organisateur: Dave Quirt (Vendredi en avant-midi, ouvert aux délégués du congrès de l'ACA)
- CSA Z107 Réunion du Comité en Acoustique et Contrôle du Bruit
  organisateur: Cameron Sherry, CWSherry@aol.com (mercredi soir)
- Visite de divers labos de le CNRC et de Santé-Canada, (vendredi en après-midi)

Exposition technique

Il y aura une exposition d'instruments et d'autres produits en acoustique. L'exposition durera toute la journée jeudi et le vendredi en avant-midi (7-8 octobre). La salle d'exposition agira comme lieu central lors des pauses. Veuillez communiquer dès maintenant avec le coordonnateur de l'exposition pour de plus amples renseignements ou pour la commandite d'événements particuliers lors du congrès.

Participation étudiante

L'ACA accorde beaucoup d'importance à la participation des étudiants. Les membres étudiants qui présenteront une communication pourront soumettre une demande de subvention pour frais de déplacement au congrès et pourront se voir mériter l'un des prix pour communications étudiantes. Veuillez consulter notre site Internet.

Appel de communications et Dates importantes

Les soumissions portant sur tous les domaines de l'acoustique sont les bienvenues. La date d'échéance pour la soumission de résumés est le **20 juin 2004**. Les résumés doivent être envoyés par courriel à **abstracts@caa-aca.ca**, inclure les renseignements habituels sur les auteurs et ne pas dépasser 250 mots. Les autres détails et exigences pour les résumés seront affichés sur le site Internet de l'ACA. Les avis d'acceptation seront envoyés peu après la date d'échéance. La date d'échéance pour la soumission de l'article de deux pages pour la revue Acoustique Canadienne, édition spéciale du congrès, est le **15 août 2004**. Si vous ne rencontrez pas cette échéance, votre article de deux pages ne sera pas publié dans l'édition spéciale du congrès. Cette édition spéciale est un portrait des nouvelles recherches en acoustique de l'année. Soyez certain d'en faire partie!

Lieu du congrès et Hébergement

Le congrès se tiendra à l'hôtel Lord Elgin, tout récemment agrandi et rénové, situé au coeur d'Ottawa à proximité de la colline parlementaire. Les délégués qui réserveront leur chambre à cet hôtel (1-800 267-4298) avant le 5 septembre 2004 bénéficieront d'un tarif préférentiel de $128/nuit (occupation simple ou double). Choisissez cet hôtel pour participer pleinement au congrès et encourager l'ACA.

Votre séjour à Ottawa

Le congrès de l'ACA est toujours une excellente occasion de renouer avec vos collègues acousticiens et de faire de nouvelles connaissances. Il y a plusieurs bistros et restaurants à proximité de l'hôtel et il y aura aussi le banquet du congrès. Pourquoi aussi ne pas en profiter et rester un peu plus longtemps pour découvrir les couleurs d'automne de la région Ottawa-Gatineau?

Personnes contacts

Président	John Bradley	(john.bradley@nrc.ca)
Directeur scientifique	Brad Gover	(brad.gover@nrc.ca)
Publicité	Christian Giguère	(cgiguere@uottawa.ca)
Exposition	Hugh Williamson	(hughwilliamson@sympatico.ca)
Webmaster	Alf Warnock	(alf.warnock@nrc.ca)
Audio-Visuel	Frances King	(frances.king@nrc.ca)

The Canadian Acoustical Association
L'Association Canadienne d'Acoustique

PRIZE ANNOUNCEMENT • ANNONCE DE PRIX

A number of prizes and subsidies are offered annually by The Canadian Acoustical Association. Applicants can obtain full eligibility conditions, deadlines, application forms, past recipients, and the names of the individual prize coordinators on the CAA Website (http://www.caa-aca.ca). • Plusieurs prix et subventions sont décernés à chaque année par l'Association Canadienne d'Acoustique. Les candidats peuvent se procurer de plus amples renseignements sur les conditions d'éligibilités, les échéances, les formulaires de demande, les récipiendaires des années passées ainsi que le nom des coordonnateurs des prix en consultant le site Internet de l'ACA (http://www.caa-aca.ca).

Deadline: Shaw, Bell, Fessenden, Eckel and Hétu Prizes: **15 April 2004**
Échéance: Prix Shaw, Bell, Fessenden, Eckel et Hétu: **15 Avril 2004**

EDGAR AND MILLICENT SHAW POSTDOCTORAL PRIZE IN ACOUSTICS • PRIX POST-DOCTORAL EDGAR AND MILLICENT SHAW EN ACOUSTIQUE

$3,000 for full-time postdoctoral research training in an established setting other than the one in which the Ph.D. was earned. The research topic must be related to some area of acoustics, psychoacoustics, speech communication or noise. • $3,000 pour une formation recherche à temps complet au niveau postdoctoral dans un établissement reconnu autre que celui où le candidat a reçu son doctorat. Le thème de recherche doit être relié à un domaine de l'acoustique, de la psycho-acoustique, de la communication verbale ou du bruit.

ALEXANDER GRAHAM BELL GRADUATE STUDENT PRIZE IN SPEECH COMMUNICATION AND BEHAVIOURAL ACOUSTICS • PRIX ÉTUDIANT ALEXANDRE GRAHAM BELL EN COMMUNICATION VERBALE ET ACOUSTIQUE COMPORTEMENTALE

$800 for a graduate student enrolled at a Canadian academic institution and conducting research in the field of speech communication or behavioural acoustics. • $800 à un(e) étudiant(e) inscrit(e) au 2e ou 3e cycle dans une institution académique canadienne et menant un projet de recherche en communication verbale ou acoustique comportementale.

FESSENDEN GRADUATE STUDENT PRIZE IN UNDERWATER ACOUSTICS • PRIX ÉTUDIANT FESSENDEN EN ACOUSTIQUE SOUS-MARINE

$500 for a graduate student enrolled at a Canadian academic institution and conducting research in underwater acoustics or in a branch of science closely connected to underwater acoustics. • $500 à un(e) étudiant(e) inscrit(e) au 2e ou 3e cycle dans une institution académique canadienne et menant un projet de recherche en acoustique sous-marine ou dans une discipline reliée à l'acoustique sous-marine.

ECKEL GRADUATE STUDENT PRIZE IN NOISE CONTROL • PRIX ÉTUDIANT ECKEL EN CONTRÔLE DU BRUIT

$500 for a graduate student enrolled at a Canadian academic institution and conducting research related to the advancement of the practice of noise control. • $500 à un(e) étudiant(e) inscrit(e) au 2e ou 3e cycle dans une institution académique canadienne et menant un projet de recherche relié à l'avancement de la pratique du contrôle du bruit.

RAYMOND HÉTU UNDERGRADUATE PRIZE IN ACOUSTICS • PRIX ÉTUDIANT RAYMOND HÉTU EN ACOUSTIQUE

One book in acoustics of a maximum value of $100 and a one-year subscription to *Canadian Acoustics* for an undergraduate student enrolled at a Canadian academic institution and having completed, during the year of application, a project in any field of acoustics or vibration. • Un livre sur l'acoustique et un abonnement d'un an à la revue *Acoustique Canadienne* à un(e) étudiant(e) inscrit(e) dans un programme de 1er cycle dans une institution académique canadienne et qui a réalisé, durant l'année de la demande, un projet dans le domaine de l'acoustique ou des vibrations.

CANADA-WIDE SCIENCE FAIR AWARD • PRIX EXPO-SCIENCES PANCANADIENNE

$400 and a one-year subscription to *Canadian Acoustics* for the best project related to acoustics at the Fair by a high-school student • $400 et un abonnement d'un an à la revue *Acoustique Canadienne* pour le meilleur projet relié à l'acoustique à l'Expo-sciences par un(e) étudiant(e) du secondaire.

DIRECTORS' AWARDS • PRIX DES DIRECTEURS

One $500 award for the best refereed research, review or tutorial paper published in *Canadian Acoustics* by a student member and one $500 award for the best paper by an individual member • $500 pour le meilleur article de recherche, de recensement des travaux ou d'exposé didactique arbitré publié dans l'*Acoustique Canadienne* par un membre étudiant et $500 pour le meilleur article par un membre individuel.

STUDENT PRESENTATION AWARDS • PRIX POUR COMMUNICATIONS ÉTUDIANTES

Three $500 awards for the best student oral presentations at the Annual Symposium of The Canadian Acoustical Association. • Trois prix de $500 pour les meilleures communications orales étudiant(e)s au Symposium Annuel de l'Association Canadienne d'Acoustique.

STUDENT TRAVEL SUBSIDIES • SUBVENTIONS POUR FRAIS DE DÉPLACEMENT POUR ÉTUDIANTS

Travel subsidies are available to assist student members who are presenting a paper during the Annual Symposium of The Canadian Acoustical Association if they live at least 150 km from the conference venue. • Des subventions pour frais de déplacement sont disponibles pour aider les membres étudiants à venir présenter leurs travaux lors du Symposium Annuel de l'Association Canadienne d'Acoustique, s'ils demeurent à au moins 150 km du lieu du congrès.

UNDERWATER ACOUSTICS AND SIGNAL PROCESSING STUDENT TRAVEL SUBSIDIES • SUBVENTIONS POUR FRAIS DE DÉPLACEMENT POUR ÉTUDIANTS EN ACOUSTIQUE SOUS-MARINE ET TRAITEMENT DU SIGNAL

One $500 or two $250 awards to assist students traveling to national or international conferences to give oral or poster presentations on underwater acoustics and/or signal processing. • Une bourse de $500 ou deux de $250 pour aider les étudiant(e)s à se rendre à un congrès national ou international pour y présenter une communication orale ou une affiche dans le domaine de l'acoustique sous-marine ou du traitement du signal.

WHAT'S NEW ??

Promotions	Retirements
Deaths	Degrees awarded
New jobs	Distinctions
Moves	Other news

Do you have any news that you would like to share with Canadian Acoustics readers? If so, send it to:

QUOI DE NEUF ?

Promotions	Retraites
Décès	Obtention de diplômes
Offre d'emploi	Distinctions
Déménagements	Autres nouvelles

Avez-vous des nouvelles que vous aimeriez partager avec les lecteurs de l'Acoustique Canadienne? Si oui, écrivez-les et envoyer à:

Steven Bilawchuk, aci Acoustical Consultants Inc., Edmonton, Alberta, Email: **stevenb@aciacoustical.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

Application for Membership

CAA membership is open to all individuals who have an interest in acoustics. Annual dues total $60.00 for individual members and $20.00 for Student members. This includes a subscription to Canadian Acoustics, the Association's journal, which is published 4 times/year. New membership applications received before August 31 will be applied to the current year and include that year's back issues of Canadian Acoustics, if available. New membership applications received after August 31 will be applied to the next year.

Subscriptions to *Canadian Acoustics* *or* Sustaining Subscriptions

Subscriptions to Canadian Acoustics are available to companies and institutions at the institutional subscription price of $60.00. Many companies and institutions prefer to be a Sustaining Subscriber, paying $250.00 per year, in order to assist CAA financially. A list of Sustaining Subscribers is published in each issue of Canadian Acoustics. Subscriptions for the current calendar year are due by January 31. New subscriptions received before August 31 will be applied to the current year and include that year's back issues of *Canadian Acoustics*, if available.

Please note that electronic forms can be downloaded from the CAA Website at **caa-aca.ca**

Address for subscription / membership correspondence:

Name / Organization ______________________________

Address ______________________________

____

City/Province ______________ Postal Code ______________ Country ______________

Phone ______________ Fax ______________ E-mail ______________

Address for mailing Canadian Acoustics, if different from above:

Name / Organization ______________________________

Address ______________________________

____

City/Province ______________ Postal Code ______________ Country ______________

Areas of Interest: (Please mark 3 maximum)

1. Architectural Acoustics
2. Engineering Acoustics/Noise Control
3. Physical Acoustics / Ultrasound
4. Musical Acoustics / Electro-acoustics
5. Psychological/Physiological Acoustic
6. Shock and Vibration
7. Hearing Sciences
8. Speech Sciences
9. Underwater Acoustics
10. Signal Processing / Numerical Methods
11. Other

For student membership, please also provide:

(University) *(Faculty Member)* *(Signature of Faculty Member)* *(Date)*

I have enclosed the indicated payment for:

[ ] CAA Membership $ 60.00

[ ] CAA Student Membership $ 20.00

[ ] Institutional Subscription $ 60.00

[ ] Sustaining Subscriber $ 250.00 includes subscription (4 issues /year) to ***Canadian Acoustics.***

Payment by: [ ] **Cheque**

[ ] **Money Order**

[ ] **VISA credit card** ***(Only VISA accepted)***

For payment by VISA credit card:

Card number ______________________________

Name of cardholder ______________________________

Expiry date ______________________________

(Signature) (Date)

Mail this application and attached payment to:

D. Quirt, Secretary, Canadian Acoustical Association, PO Box 74068, Ottawa, Ontario, K1M 2H9, Canada

l'Association Canadienne d'Acoustique
The Canadian Acoustical Association

Formulaire d'adhésion

L'adhésion à l'ACA est ouverte à tous ceux qui s'intéressent à l'acoustique. La cotisation annuelle est de 60.00$ pour les membres individuels, et de 20.00$ pour les étudiants. Tous les membres reçoivent l'Acoustique Canadienne, la revue de l'association. Les nouveaux abonnements reçus avant le 31 août s'appliquent à l'année courante et incluent les anciens numéros (non-épuisés) de l'Acoustique Canadienne de cette année. Les nouveaux abonnements reçus après le 31 août s'appliquent à l'année suivante.

Abonnement pour la revue Acoustique Canadienneet abonnement de soutien

Les abonnements pour la revue Acoustique Canadienne sont disponibles pour les compagnies et autres établissements au coût annuel de 60.00$. Des compagnies et établissements préfèrent souvent la cotisation de membre bienfaiteur, de 250.00$ par année, pour assister financièrement l'ACA. La liste des membres bienfaiteurs est publiée dans chaque issue de la revue Acoustique Canadienne. Les nouveaux abonnements reçus avant le 31 août s'appliquent à l'année courante et incluent les anciens numéros (non-épuisés) de l'Acoustique Canadienne de cette année. Les nouveaux abonnements reçus après le 31 août s'appliquent à l'année suivante.

Pour obtenir des formulaires electroniques, visitez le site Web: caa-aca.ca

Pour correspondence administative et financière:

Nom / Organisation ____________________

Adresse ____________________

Ville/Province ____________________ *Code postal*

Pays ____________________

Téléphone ____________________ *Téléc.*

Courriel ____________________

Adresse postale pour la revue Acoustique Canadienne

Nom / Organisation ____________________

Adresse ____________________

Ville/Province ________ *Code postal*________ *Pays* ________

City/Province ________ Postal Code ________Country ________

Areas of Interest: (Please mark 3 maximum)

1. Acoustique architecturale
2. Génie acoustique / Contrôle du bruit
3. Acoustique physique / Ultrasons
4. Acoustique musicale/Electro-acoustique
5. Physio / Psycho-acoustique
6. Chocs et vibrations
7. Audition
8. Parole
9. Acoustique sous-marine
10. Traitement des signaux /Méthodes numériques
11. Autre

Prière de remplir pour les étudiants et étudiantes:

(Université) (Nom d'un membre du corps professoral) (Signature du membre du corps professoral) (Date)

Cocher la case appropriée:

[ ] Membre individuel $ 60.00

[ ] Membre étudiant(e) $ 20.00

[ ] Abonnement institutionnel $ 60.00

[ ] Abonnement de soutien $ 250.00

(comprend l'abonnement à L'acoustique Canadienne)

Méthode de paiement:

[ ] **Chèque au nom de l'Association Canadienne d'Acoustique**

[ ] **Mandat postal**

[ ] **VISA** (Seulement VISA)

Pour carte VISA: Carte n° ____________________

Nom: ____________________

Date d'expiration:____________________

(Signature): ____________________(Date): ________

Prière d'attacher votre paiement au formulaire d'adhésion. Envoyer à:

D. Quirt, Secrétaire exécutif, Association Canadienne d'Acoustique, Casier Postal 74068, Ottawa, K1M 2H9, Canada

The Canadian Acoustical Association
l'Association Canadienne d'Acoustique

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