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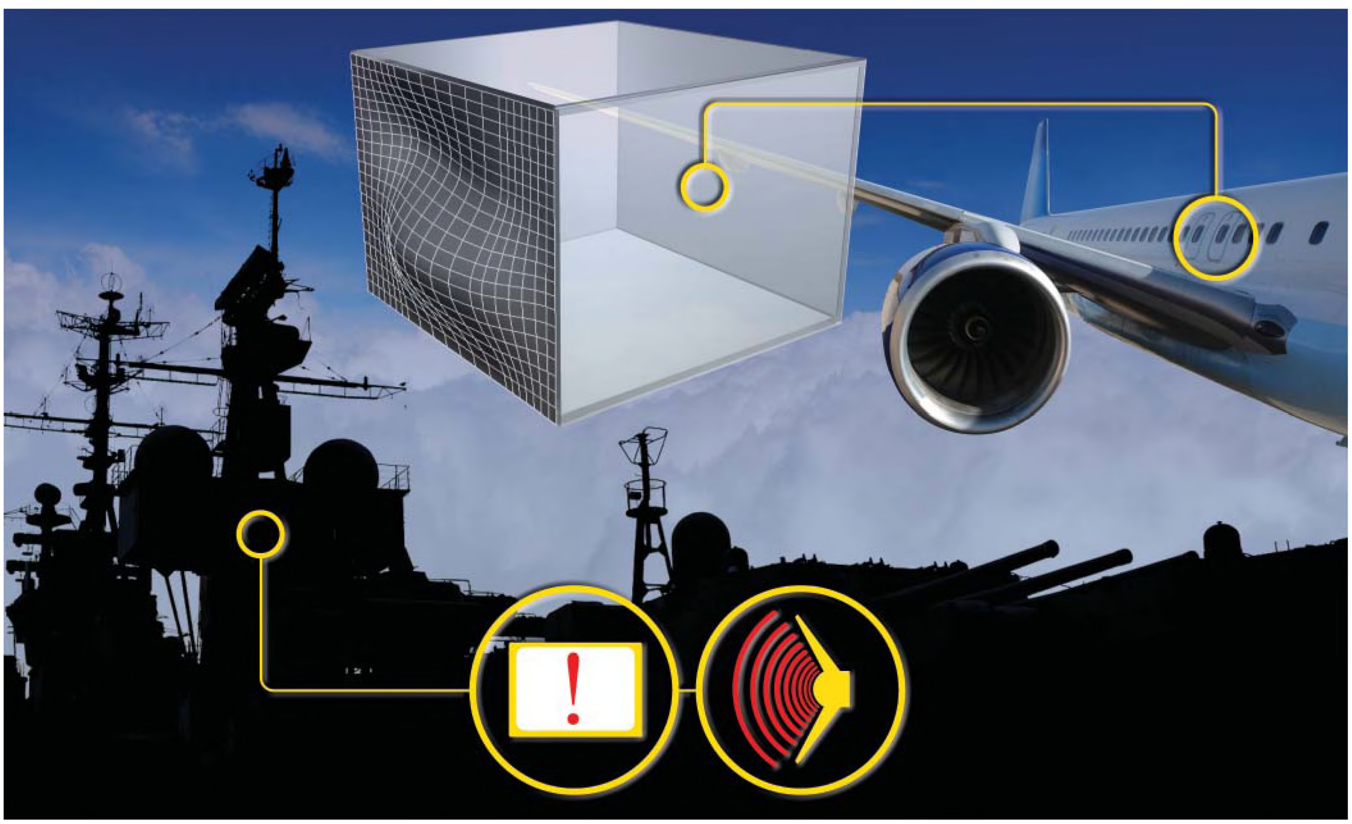

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CANADIAN ACOUSTICS publishes refereed articles and news items on all aspects of acoustics and vibration. Articles reporting new research or applications, as well as review or tutorial papers and shorter technical notes are welcomed, in English or in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

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PRESIDENT'S MESSAGE / MESSAGE DU PRÉSIDENT

In an intriguing turn of events, the CAA will return a little closer to its roots, tracing back to 1962, when a group of acousticians first met at NRC to coordinate acoustical standards activities in the country. The newly formed "Canadian Committee on Acoustics" met annually and grew well beyond to become the full-fledged Association that we now know. Over the years, standards activities were transferred to the Canadian Standards Association (CSA) and Standards Council of Canada (SCC). While our members were very active in these organizations and standards meetings remained an important event during Acoustics Week in Canada, the CAA no longer had a specific role regarding standards. This all changed this year when CSA decided to maintain activities only in a renamed Technical Committee on Occupational Hearing Conservation (Z94.2) and to disband the Technical Committee on Acoustics and Noise Control (Z107). The recently formed "CAA Acoustical Standards Committee" will take over to fill the gap left in several areas, coordinate Canadian involvement with US and international standards writing groups such as ISO, IEC, ASTM and ASA, and provide technical advice to the CAA Board on issues raised by I-INCE, ICA, and other organizations of which we are a member society. Without a doubt, this key development will help raise the profile of the CAA among Canadian and international acousticians. Now is the best time to get involved, and interested members should contact the current chair, Tim Kelsall. Long Live the Standards Committee!

We enjoyed a very successful Acoustics Week in Canada 2010 this past October in Victoria in a beautiful setting around the Inner Harbour. Special thanks go to Stan Dosso and the members of his organizing committee: Roberto Racca, Clair Wakefield, Lori Robson, Lara Berg, Brian Rideout and Michael Wilmut. Over 160 attendees and exhibitors participated, and 110 technical papers were delivered covering all areas of acoustics. There was again a large student participation this year and a strong turn out of Exhibitors. A special mention goes to the three plenary speakers: Christine Erbe presenting on marine soundscapes, Murray Hodgson on the acoustics of green buildings, and Gary Heard on Arctic acoustics, who all shared their contagious passion for acoustics with us.

Acoustic in Canada 2011 will be held in another beautiful setting in Quebec city, October 12-14th. It will be the 50th Annual Meeting of our organization and, on this occasion, it is an honour for me to convene this event with a group of Quebec acousticians. Please mark it down in your calendar, and consult the current and future issues of Canadian Acoustics or the website for more information.

Since 1999, our Editor-in-Chief, Ramani Ramakrishnan, has worked tirelessly on every issue and done a superb job at publishing our Quarterly Journal Canadian Acoustics. Thanks to his effort and personal vision, the Journal remains a pillar of

Une tournure toute particulière d'événements au cours des derniers mois ramènera l'ACA vers ses origines remontant à 1962 lorsqu'un petit groupe d'acousticiens se rencontra au CNRC pour coordonner les activités canadiennes en matière de normalisation en acoustique. Le tout nouveau « Comité canadien sur l'acoustique » connu une forte croissance pour devenir l'Association que nous connaissons maintenant. Au fil des ans, les activités de normalisation ont été transférés à l'Association canadienne de normalisation (ACNOR) et au Conseil canadien des normes (CCN). Bien que nos membres ont toujours été très actifs auprès de ces organismes et que des réunions sur la normalisation se sont toujours tenues durant la Semaine canadienne d'acoustique, l'ACA n'avait plus de rôle précis en matière de normalisation. Tout cela a changé cette année lorsque l'ACNOR a décidé de ne maintenir que les activités du Comité technique sur la préservation de l'ouïe en milieu de travail (Z94.2) et de dissoudre le Comité technique sur l'acoustique et le contrôle du bruit (Z107). Le tout nouveau « Comité des normes de l'ACA » prendra le relais pour combler un vide laissé dans plusieurs sphères d'activités, coordonner la participation du Canada auprès d'organismes de normalisation américains et internationaux tels que l'ISO, la CEI, l'ASTM et l'ASA, en plus de fournir des avis au Comité de direction de l'ACA sur toute question technique soulevée par I-INCE, l'ICA et les autres organisations dont nous sommes société membre. Cet important développement contribuera sans aucun doute à rehausser le profil de l'ACA auprès des acousticiens canadiens et internationaux. Les membres intéressés à participer aux activités de normalisation sont invités à communiquer avec le président du comité, Tim Kelsall. Vif succès au Comité des normes !

La Semaine canadienne d'acoustique 2010 s'est tenue à Victoria en octobre dernier sur un site enchanteur aux abords du port intérieur et a connu encore année un franc succès. Maints remerciements à Stan Dosso et aux membres de son comité organisateur : Roberto Racca, Clair Wakefield, Lori Robson, Lara Berg, Brian Rideout et Michael Wilmut. Le congrès a attiré plus de 160 participants et exposants et quelques 110 communications orales ont été présentées dans les différents domaines de l'acoustique. Nous avons bénéficié encore cette année d'un bon contingent d'étudiants et d'une forte participation à l'exposition technique. Il faut aussi souligner la prestation des trois conférenciers en plénière qui ont parlé avec grande passion de leurs travaux en acoustique: Christine Erbe sur l'écologie de l'environnement sonore marin, Murray Hodgson en acoustique des bâtiments verts et Gary Heard sur l'acoustique de l'Arctique canadien.

La Semaine canadienne d'acoustique 2011 se tiendra dans un endroit tout aussi charmant dans la ville de Québec du 12 au 14 octobre prochain. Il s'agira du 50ième congrès annuel de notre association et j'aurai l'honneur de présider ce colour Association. Retirement is on the horizon and Ramani would like to undergo a smooth transition to a new Editor-in-Chief. To this end, he is seeking to train an Assistant Editor to gradually take over part of the operations of producing the Journal, with a goal that the new Assistant would stand election as Editor-in-Chief by 2013. We thereby invite any CAA member with a special interest in eventually assuming the role of Editor-in-Chief to contact Ramani or myself. This is a unique opportunity to learn from the Great Master himself!

On behalf of our Editor-in-Chief, I would also like to thank Jason Tsang for several years of service as Advertising Coordinator for Canadian Acoustics. Journal ads are a crucial revenue component to allow distributing Canadian Acoustics to our wide membership at a reasonable cost. It is a pleasure to announce that Rich Peppin will now serve as the new Advertizing Coordinator. Rich wants to put in place new mechanisms to facilitate advertising operations for both the Association and our advertisers.

Having served three years as President of the Association, I now more than ever realize how much volunteering effort is required to conduct all our activities and provided for by a good number of people. I'd like to take the opportunity this year to thank everyone and invite any CAA member with a desire to contribute to the operation of the Association to contact me or any CAA Director.

Christian Giguère
CAA President

loque avec des collègues acousticiens du Québec. Veuillez inscrire dès maintenant cette date à votre agenda et consulter les annonces du congrès dans l'Acoustique canadienne pour de plus amples renseignements.

Depuis 1999, notre rédacteur en chef, Ramani Ramakrishnan, a travaillé sans relâche à la publication de notre revue trimestrielle l'Acoustique canadienne. Son grand dévouement et sa vision personnelle ont porté fruits, la revue demeure un pilier de notre Association. La retraite se pointe à l'horizon pour Ramani et il entend démarrer dès maintenant le processus de transition vers un nouveau rédacteur en chef. Il est à la recherche d'un adjoint pour assumer progressivement la production de la revue avec comme objectif que le nouvel adjoint soit fin prêt à se faire élire comme rédacteur en chef en 2013. Nous invitons ainsi tout membre de l'ACA intéressé à se préparer à devenir rédacteur en chef de notre revue à contacter Ramani ou moi-même. Il s'agit d'une occasion unique d'apprendre du Grand Maître lui-même !

Au nom de notre rédacteur en chef, je tiens aussi à remercier Jason Tsang qui a assumé le rôle de d'agent de publicité pour l'Acoustique canadienne depuis quelques années. Ces revenus publicitaires nous sont essentiels afin de distribuer notre revue à tous nos membres à coût raisonnable. Il me fait plaisir d'annoncer que Rich Peppin va maintenant agir comme nouvel agent de publicité. Rich veut mettre en place de nouveaux mécanismes pour faciliter les opérations de publicité tant pour l'Association que pour nos annonceurs.

Après trois ans comme président de l'Association, je réalise plus que jamais à quel point l'effort bénévole d'un grand nombre de personne est nécessaire pour réaliser l'ensemble de nos activités. Je profite de l'occasion pour remercier tous ceux-ci et inviter tout membre de l'ACA désirant contribuer au fonctionnement de l'association à communiquer avec moi ou avec un membre du conseil d'administration.

Christian Giguère
Président de l'ACA

POSITION OPEN - Assistant Editor, Canadian Acoustics Journal

The current Editor-in-Chief, Ramani Ramakrishnan, will be stepping down in 2012. A new Editor will be elected during the Annual General Meeting (AGM) in October 2012 and will become the new Editor-in-Chief from 2013 onwards. CAA is looking for an Assistant Editor who will work with Ramani Ramakrishnan and will be trained to be the next Editor. He or she will be nominated during the 2012 AGM and it is hoped that the membership will vote him/her to be the Editor-in-Chief.

The current proposal is to aid in the smooth transition from Ramani Ramakrishnan to the new Editor.

Interested person should contact either the president, Christian Giguère or Ramani Ramakrishnan.

EVALUATION OF AUDIO AND VISUAL ALERTS DURING A DIVIDED ATTENTION TASK IN NOISE

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ABSTRACT

The Halifax class frigate operations room is a demanding environment in which operators are required to monitor multiple visual displays and auditory communication channels. The current alerting system is ineffective, as the visual alerts tend to be ignored or dismissed without being read, and the auditory alerts are turned off completely. Visual alerting strategies have already been investigated. The current study compared the response times (RT) to visual, auditory and combined (audiovisual) alerts as subjects performed a visual divided attention task using three displays (secondary task). Another objective was to investigate the effects of alert type on the performance of the secondary task. The experiment was performed in quiet and in recorded frigate control room noise (69 dBA). There were no significant differences in RT between the visual and audiovisual alerts in quiet or noise. The RT for the auditory alert was significantly higher than the audiovisual alert in quiet, and than both the visual and audiovisual alerts in noise. There was no main effect of alert type on the performance of the secondary task. The audiovisual alert could be beneficial for detection in the operations room because 1) the RT was not significantly different from the visual alert, indicating that the auditory component was not distracting, and 2) it is more likely to be detected over the visual alert when the operators are looking away from the displays. Future studies should investigate the psychoacoustic properties of the auditory component of the alert for perceived urgency, in the interest of prioritizing the alerts.

RÉSUMÉ

La salle des opérations d'une frégate de classe Halifax est un environnement exigeant dans lequel les opérateurs doivent surveiller de nombreux affichages visuels et canaux de communication sonore. Le système d'alerte actuel n'est pas efficace, car les alertes visuelles sont souvent ignorées ou rejetées sans être lues, et les alertes sonores sont coupées complètement. Les stratégies d'alerte visuelle ont déjà été étudiées. La présente étude visait à comparer le temps de réaction (TR) aux alertes visuelles, sonores et combinées (audiovisuelles) de sujets qui effectuaient une tâche visuelle en situation d'attention partagée au moyen de trois afficheurs (tâche secondaire). Un autre objectif consistait à étudier les effets du type d'alerte sur le rendement pour la tâche secondaire. L'expérience a été réalisée dans un milieu silencieux et un milieu bruyant où jouait le bruit enregistré d'une salle de contrôle de frégate (69 dBA). On n'a remarqué aucune différence significative dans le TR entre les alertes visuelles et audiovisuelles dans un milieu silencieux ou bruyant. Le TR pour l'alerte sonore était beaucoup plus élevé que celui pour l'alerte audiovisuelle dans un milieu silencieux et ceux pour les alertes visuelles et audiovisuelles dans un milieu bruyant. Le type d'alerte n'a eu aucun effet principal sur le rendement de la tâche secondaire. L'alerte audiovisuelle aiderait à la détection dans la salle des opérations pour les raisons suivantes : 1) le TR était peu différent de celui pour l'alerte visuelle, ce qui indique que les éléments sonores ne détournaient pas l'attention; 2) l'alerte audiovisuelle a plus de chances d'être détectée que l'alerte visuelle lorsque les opérateurs ne regardent pas les afficheurs. De nouvelles études devraient porter sur les propriétés psychoacoustiques de l'élément sonore de l'alerte pour les urgences perçues, de façon à établir l'ordre de priorité des alertes.

1. INTRODUCTION

The Halifax class frigate operations room is a demanding, high-intensity environment, manned by about twenty Navy personnel. Many of the personnel are sensor operators, who are required to monitor several visual displays showing tactical and administrative data. The operators also use headsets to monitor two channels of communication while keeping track of face-to-face communication within the room. Automated systems, both auditory and visual, are in place to warn operators of impending system errors and tactical threats (e.g., detection of submarines, mines, torpedoes). However, recent discussions with Navy personnel indicate that the alerting system is not effective. The alerts are not prioritized, and all alerts are sent to all operators, regardless of relevance. The auditory alert is a single tone that is activated constantly and presents no information about the urgency of the alert. As a result, the operators tend to turn the auditory alerts off due to annoyance. Visual alerts appear as flashing text at the bottom of one of the screens, and can be easily ignored or dismissed without being read. The Halifax Class Frigate is undergoing a complete modernization upgrade and it is therefore of interest to explore methods of enhancing the way operators are alerted.

For high-priority alerts, it is important to capture the attention of the operator as quickly as possible. Although simple reaction time (RT) has been reported as faster for auditory than for visual stimuli (130 vs 170 ms), it has been argued that this is only the case when the subjective intensity differences between the modalities was not controlled (Wickens, 1992). When a visual stimulus is presented simultaneously with an auditory stimulus (bimodal), it has been shown that subjects tend to respond to the visual stimulus, and the auditory component is often ignored (Colavita, 1974). This phenomenon is known as visual dominance (Colavita and Weisberg, 1979; Sinnett et al, 2007). It is unknown if the results for experiments of simple or serial RT can be generalized to real-world environments in which a complex task must be accomplished in the presence of environmental stressors. Sinnett et al (2007) were able to replicate the visual dominance results of Colavita (1974) and extended the experiments by increasing the perceptual load in both the visual and auditory domains. This was done by adding a number of distracting (i.e., non-target) images and sounds to the presented stimuli. There were no significant differences in RT for auditory and visual alerts with the increased perceptual load (Sinnett et al, 2007). Anaesthetists have been shown to respond more quickly to auditory alarms from the monitoring equipment than visual (Morris and Montano, 1996); however, another study showed no differences in RT between the two modalities (Loeb and Tecumseh, 2002). Conflicting results have also been shown for the use of audiovisual alerts. Some studies have shown audiovisual alerts to elicit faster RT and better accuracy (fewer missed alarms) than both unimodal auditory and visual alarms (Chan and Chan, 2006), while other studies have found better accuracy, but slower RT for the audiovisual alerts (Sinnett et al, 2007).

Another factor to consider in real-world environments is the presence of background noise. Previous studies have shown that the presence of high-level background noise did not affect serial RT for visual alerts, but accuracy was adversely affected (Abel, 1990). Auditory alerts must be presented at levels that are sufficient to be heard above the background noise, but not so high that they become startling or distracting. It has been suggested that auditory warnings should be between 15 and 25 dB above the masked threshold to ensure detection without being aversive or disruptive to verbal communication (Patterson, 1982). Thus, the optimal level of the auditory alarm depends on the level and spectrum of the background noise. In addition to the level, there are a number of psychoacoustic properties known to affect the perceived urgency of an auditory alarm (Patterson, 1982; Edworthy and Hellier, 2000; Arrabito et al, 2004).

In the frigate operations room, the high workload in both the visual and auditory modalities (multiple visual displays and communication channels) and the presence of background noise will impact how the operators react to different types of alarms. Different types and locations of visual alerts using three displays have been investigated (Crébolder and Beardsall, 2009; Crébolder & Beardsall, 2009b; Roberts, 2008, Roberts and Foster-Hunt, 2008). It was found that a vertical red bar appearing on one or all the displays was detected more quickly than a red border around the displays. This was true whether the bar appeared at the top left side, or centered across the top or bottom of the display. In addition, static alerts (red bar or border displayed for four seconds) were detected more quickly than flashing alerts, and detection was faster when the alerts were shown simultaneously on all three displays than when shown on one. The purpose of the current study was to compare the RT for the previously tested visual alerts to simple auditory and audiovisual alerts. A secondary objective was to investigate the effects of alert type on the performance of a simulated operations room task.

2. EXPERIMENTAL PROTOCOL

2.1 Participants

Protocol approval was obtained from the Human Research Ethics Committee (HREC) of Defence Research and Development Canada (DRDC), and informed consent was obtained from all participants. Sixteen subjects (eight males, eight females) aged 21 to 51 years (30.6 ± 8.8 years) participated in the study. All subjects had normal or corrected-to-normal vision. A trained technician screened the subjects for a history of ear disease and hearing thresholds no greater than 25 dB HL (hearing level), bilaterally, at pure tone frequencies of 0.5, 1, 2 and 4 kHz. The hearing thresholds were measured with an audiometer (Interacoustics AC40, Eden Prairie, MN) using Békésy tracking (Brunt, 1985), in a double-walled, soundproof booth (Series 1200, Industrial Acoustics Company, Bronx, NY). All subjects were right-handed.

2.2 Experimental platform

The experiment was conducted using a personal computer with three 22” liquid crystal display (LCD) monitors. The software was programmed using E-Prime 2.0 (Psychology software tools, Pittsburgh, PA), running on a Windows XP operating system. The primary task required the subjects to respond to alerts by pressing the space bar (the alerts are described in the next section). The secondary task required the subjects to monitor a tactical display (center monitor) and classify contacts (unknown ships) as being neutral or hostile. A schematic of the experimental platform is shown in Figure 1. The task was designed to be a high-intensity task, involving multiple displays, similar to a task that a sensor operator might be required to perform. Contacts (triangles on the screen) originated in the periphery of the tactical display and moved toward the ownship (the ship to be protected), located at the centre of the display. Upon selecting a contact on the tactical display, information regarding the ship size, ship speed and whether or not weapons were on board, was displayed on the status display (left monitor). The subject used the information to determine if the contact was hostile or neutral, and then entered the corresponding three character code on the reporting display (right monitor). The codes were "qwe" for a neutral contact and "asd" for a hostile contact. The contact information is listed in Table 1. A contact was categorized as hostile if any two of its attributes were hostile, or neutral if any two attributes were neutral. When a contact was classified correctly, it disappeared from the tactical display; otherwise, it kept moving toward the ownship. The subject received feedback about the correctness of the entered response by a message displayed under the response box (correct or incorrect). If any contact, neutral or hostile, reached the ownship, it was destroyed and the displays were reloaded.

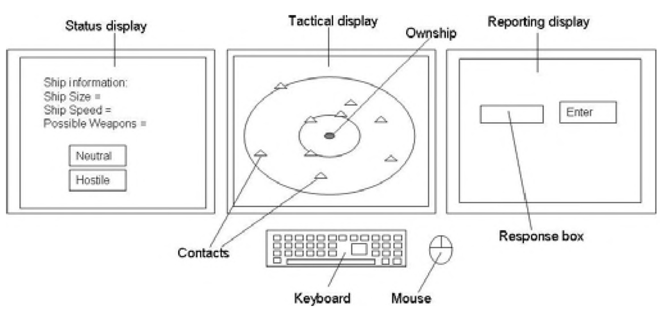

Figure 1: The experimental platform.

Table 1. Hostile and neutral attributes for contact classification.

	Categorized as Hostile	Categorized as Neutral
Ship Size	Small	Large
Ship Speed	Fast	Slow
Possible Weapons	Yes	No

2.3 Design

We employed a 3x2 repeated measures design (three alert types x two background noise conditions). The three alert types were visual, auditory and audiovisual, and will be described in greater detail below. The two background noise conditions were quiet and recorded Halifax class frigate operations room noise (69 dBA). The noise was a combination of speech and machinery noise, thus potentially providing both informational and energetic masking of the auditory alert (Brungart, 2001); the one-third octave band spectrum, as well as the total A-weighted and linear levels, are shown in Figure 2. Since most of the energy in the noise spectrum was below 1 kHz, an auditory alert composed of frequencies in this range would have to be presented at a relatively higher level to reduce the effects of masking. The auditory alert was a 1 kHz tone, presented at 75 dB SPL, which was well above the background noise level in that frequency band (54 dB SPL). We chose this simple auditory alert because it was similar to what is currently being used in the operations room, and therefore operationally relevant. Both the background noise and the auditory alerts were presented over headphones (Sennheiser HD 280), which were worn during both of the background conditions. The visual alert was a static red bar shown at the bottom of all three displays. We chose to show the visual alert on all three displays to match the omnidirectional nature of the auditory alert as closely as possible. Therefore, the three types of alerts were: audio only (1 kHz tone), visual only (a static red bar shown on all three of the displays) and audiovisual (1 kHz tone synchronized with the red bar). The alerts were displayed for four seconds, or until acknowledged by the subject by pressing the space bar. The duration of the alerts on the screen was chosen to maintain consistency with the previous studies (Crébolder and Beardsall, 2009; Crébolder & Beardsall, 2009b; Roberts, 2008, Roberts and Foster-Hunt, 2008). Alerts that were not acknowledged were recorded as being missed.

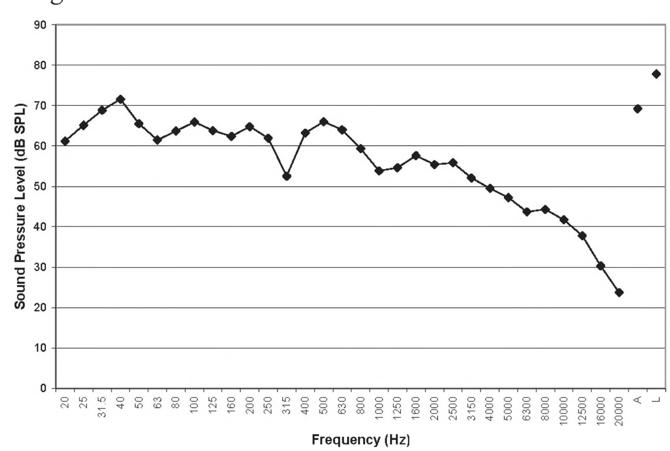

Figure 2: Background noise recorded in the Halifax Class Frigate operations room.

The experiment was divided into blocks. Between 14 and 16 alerts were presented within a block and each block lasted approximately three minutes. Only one type of alert was presented within a block. The alert types were randomized between blocks, such that nine blocks were presented in total with three of each alert type. The subjects performed the experiment twice: once in quiet, and once in noise. The order of the quiet and noise sessions was counterbalanced between subjects.

2.4 Procedure

All subjects participated in a training session to familiarize themselves with the displays, functions, and task. Using a training experiment consisting of three blocks (one of each type of alert), an experimenter talked them through the first run, and then allowed them to run through a second time without help. During the second run, the subject wore the headphones with the background noise turned on. Most subjects were comfortable with the task after completing the training experiment twice (once in quiet, once in noise).

During the experimental session, subjects first performed a warm-up set of three blocks. They then completed two sets of nine blocks: one in quiet and one in noise, with a 10-minute break in between.

2.5 Statistical Analysis

The main outcome measures of this study were RT to alert type, and efficiency of contact classification (accuracy and speed). The numbers of missed alerts, false alarms and destroyed ownship were also analyzed. Within-subjects analyses of variance (ANOVA) were applied to the data to evaluate the significance of variation in alert type and noise condition. Non-parametric analyses (Friedman) were used to analyze the contact classification data. The effects of age on the main outcome measures were calculated using correlations. All analyses were calculated using SPSS 17.0 (Statistical package for social sciences, SPSS Inc., 2008) and p values < 0.05 were considered to be significant.

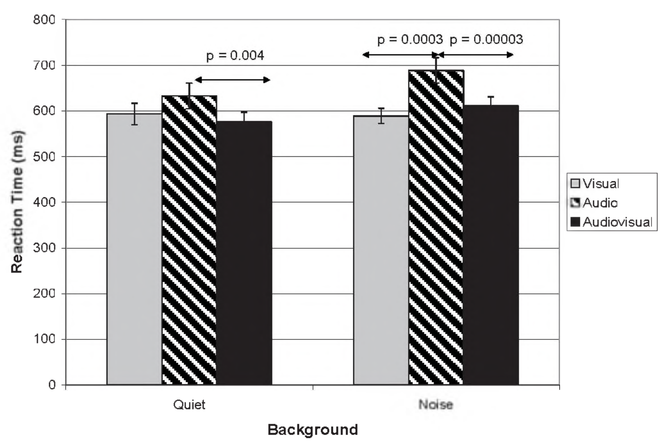

Figure 3: Reaction time in quiet and noise by alert type. The error bars indicate standard error.

3. RESULTS

3.1 Primary Task – Alert Detection

The RTs for the three types of alerts, grouped by background noise condition, are shown in Figure 3. A repeated-measures ANOVA with two factors (alert type and background) was applied to the RT data. Because sphericity was violated for alert type, a Greenhouse-Geisser correction was applied to the degrees of freedom ($\varepsilon = 0.688$). There were main effects for alert type, $F(1.38, 20.65) = 18.65$, $p = 0.0001$, partial $\eta^2 = 0.554$ and background, $F(1, 15) = 12.16$, $p = 0.003$, partial $\eta^2 = 0.448$. There was also a significant interaction between type and background, $F(2, 30) = 9.321$, $p = 0.001$, partial $\eta^2 = 0.383$. Post-hoc analyses using a Bonferroni correction showed significantly longer RT for the audio alert (633 ±113 ms) than the audiovisual alert (575 ± 84 ms) in quiet ($p = 0.004$). In noise, the mean RT for the audio alert (689 ±110 ms) were significantly higher than both the visual (589 ± 67 ms, $p = 0.0003$) and audiovisual (611± 78 ms, $p = 0.00003$) alerts.

The mean RTs, grouped by alert type, are shown in Figure 4. T-tests were calculated to look at the effect of background noise on RT for each of the alert types. There were significant differences between the RT for the auditory alert in quiet (633 ± 113 ms) and noise (689 ± 110 ms), $t(15) = 0.001$ (two-sided) and the audiovisual alert in quiet (576 ± 84 ms) and noise (611 ± 78 ms), $t(15) = 0.004$ (two-sided).

Pearson correlation coefficients were calculated between age and RT for each alert type and background condition (Table 2). Significant positive correlations were found for the audiovisual RT in quiet ($r = 0.514$, $p = 0.04$), and all alert types in noise (visual: $r = 0.558$, $p = 0.025$; audio: $r = 0609$, $p = 0.012$; audiovisual: $r = 0.663$, $p = 0.005$). The correlations indicated an increase in reaction time with age, especially in noise.

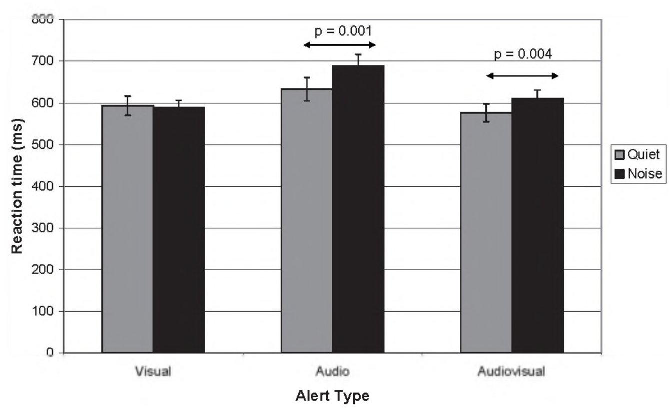

Figure 4: Reaction time by alert type in quiet and noise. The error bars indicate standard error.

There was only one occurrence of a missed alert by one subject (a visual alert in the quiet condition), so no analysis was performed on the miss data. The number of false alarms (hitting the space bar when no alert was presented) was also small. The average number of false alarms in quiet was 1.3± 1.5 (collapsed across alert type) and 2.4 ± 1.7 in noise, out of approximately 135 presentations (nine blocks of 14 to 16 presentations).

Table 2. Pearson correlation coefficients between subject age and reaction time.

Alert type (Background)	Correlation coefficient	P
Visual (quiet)	0.458	0.074
Audio (quiet)	0.465	0.069
Audiovisual (quiet)	0.514	0.04*
Visual (noise)	0.558	0.025*
Audio (noise)	0.609	0.012*
Audiovisual (noise)	0.663	0.005*

3.2 Secondary Task – Contact Classification

Performance on the secondary task was analyzed based on the accuracy and the total number of contacts classified (neutral and hostile combined). The accuracy data (percentage correct) for contact classification were non-normally distributed. Non-parametric Friedman tests applied to the data in quiet and noise showed no main effect of alert type for either background condition. In addition, there was no main effect of background on accuracy. The average accuracies (collapsed across all alert types) were 99.0 ± 0.8% and 98.6 ± 1.3% in quiet and noise, respectively. Similarly, Friedman tests applied to the data for number of contacts classified showed no main effect of alert type or background. The average numbers of contacts classified (collapsed across all alert types) were 335 ± 17 and 335 ± 18 in quiet and noise, respectively.

Ownship explosions occurred if a contact reached the ownship without being classified. There were only two occurrences of exploded ownship in quiet (one subject, once each during the visual and audio alert conditions) and four in noise (four different subjects, twice each during the visual and audiovisual alert conditions).

4. DISCUSSION

It has been demonstrated that the auditory modality is superior to the visual for alerting (Wickens, 1992); therefore, one might expect that a unimodal auditory alert would be responded to more quickly than a unimodal visual alert. However, factors such as intensity differences between the stimuli, workload in the two modalities, and environmental noise, seem to complicate these general conclusions. In the current study, there was no significant difference in RT between the audio and visual alerts in quiet, but the RT was significantly higher for the audio alert than the visual in noise. This finding might be explained by the fact that the perceptual load in the auditory modality is low in the quiet condition, allowing more modality-specific resources to process the target (Sinnett et al, 2007). In the noise condition, since the perceptual load increased in the auditory modality but not the visual, audio RT increased.

The choice of modality for an alert in part depends on the attentional resources being consumed by the task at hand. The auditory modality is omnidirectional, meaning that an auditory alert can be heard regardless of the source location. By contrast, in order for a visual alert to be effective, the subject must process it by actively attending to the spatial location of the alert (Posner, Nissen, & Klein, 1976). For example, it has been suggested that anaesthetists spend less than one-third of their time looking at the monitors in the operating room, which places limitations on the use of unimodal visual alarms (Loeb and Fitch, 2002). For this type of environment, it may be safer to rely on auditory alarms when a time-sensitive (i.e., high priority) response is required (Morris and Montano, 1996). In the current study, the subjects were engaged in a visually intensive task, requiring them to attend to the displays at all times. This was likely why the RT for visual and audio alerts were not significantly different in the quiet condition.

The use of a bimodal (e.g., audiovisual) alert can help to enhance detection in complex, multimodal environments. It is often necessary to use auditory alerts in conjunction with visual alerts because operators need to read or otherwise examine the context of the warning message (Chan and Chan, 2006). The literature has suggested that audiovisual alerts lead to fewer detection errors, but may or may not be detected more quickly than their unimodal counterparts (Chan and Chan, 2006; Sinnett et al, 2007). Our results showed that the RT for the audiovisual alert in both the quiet and noise conditions was significantly faster than the RT for the unimodal audio alert, but not significantly different from the unimodal visual alert. It is possible that the subjects displayed visual dominance in their response to the audiovisual alerts (responded to the visual component while ignoring the audio); however, since the RT for audiovisual alerts was not significantly different from the visual alerts, it appeared that the auditory component did not hinder detection. None of the alert types affected performance on the secondary task. A combined alert would likely be advantageous because operators will shift their attention from the displays for various reasons, such as when talking to someone in the room.

Our results for the unimodal auditory alert (slower RT than visual and audiovisual) seem to support the behaviour of the sensor operators, who turn the auditory alerts off due to annoyance. However, since the audiovisual alert did not cause any performance decrements over the visual alert in terms of RT or secondary task performance, the design and implementation of audiovisual alerts in the operations room should be further investigated. Specifically, the addition of a divided attention auditory task to simulate the monitoring of communication channels should be added to the experimental platform. The additional load on the auditory channel will make it more difficult to alert the operator. In addition, different types of auditory alerts could be introduced to demonstrate urgency or priority.

5. CONCLUSION

An investigation of visual, audio and audiovisual alerts in the Halifax class frigate control room environment showed that while performance of the secondary visual task was not different across alert types, reaction time was slowest for the auditory alert in both quiet and noise. Reaction time for the audiovisual alert was not significantly different than for the visual alert, and the audiovisual alert may be more easily detected when the operators are not looking directly at the displays. The design and implementation of audiovisual alerts in the modernized frigate control room should therefore be further investigated.

ACKNOWLEDGEMENTS

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TURBULENT BOUNDARY LAYER INDUCED NOISE AND VIBRATION OF A MULTI-PANEL WALLED ACOUSTIC ENCLOSURE

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ABSTRACT

Flow-induced noise in aircraft cabins can be predicted through analytical models or numerical methods. To date, analytical methods have been used for simple structures and cabins, where usually a single panel is vibrating due to the flow excitation, and coupled with an acoustic enclosure. The present work investigates the analytical prediction of turbulent boundary layer induced noise and vibration of a multi-panel system. The objective is to investigate the coupling between individual panels and the acoustic enclosure. Each panel is coupled with the acoustic enclosure, which consists of a large rectangular room, with five rigid walls and one flexible wall. The properties of the panels and acoustic enclosure represent a typical fuselage skin panel and a cabin section, respectively. It is shown that identical panels located at different positions have dissimilar contributions to the cabin interior noise, showing that the panel position is an important variable for the accurate prediction and suppression of cabin noise. Analytical predictions were obtained for both the space-averaged interior sound pressure level and local interior sound pressure level. The space-averaged sound pressure level is usually accepted to provide the necessary information for the noise prediction; however, in some real life applications, the local sound pressure may also be desirable.

RÉSUMÉ

Le bruit à l'intérieur d'es cabines d´es avions induite par écoulement externe peut être prédit par modèles analytiques ou méthodes numériques. À ce jour, les méthodes analytiques ont été utilisés pour structures et chambre simples, où, normalement, un seul panneau est considéré à vibrer en raison de l'écoulement externe, et couplé avec la chambre acoustique. Cet article étudie la prévision analytique des vibrations et du bruit dans un système avec plusieurs panneaux. L'objectif est d´examiner le couplage entre panneaux individuels et la chambre acoustique, en considérant de l'emplacement du panneau comme une variable. La cabine acoustique est une grande chambre rectangulaire et les panneaux rectangulaires sont considérés simplement appuyés. Les propriétés des matériaux et les dimensions des panneaux et de chambre acoustique sont représentatives d'un panneau de fuselage typique d'un avion et un compartiment de la cabine, respectivement. Il est conclu que panneaux similaires situés dans des positions différentes de la cabine ont contributions différentes du bruit intérieur, montrant que la position du panneau est une variable importante pour une prévision précise de bruit et de suppression de bruit dans la cabine. Ont été obtenu des prévisions analytiques des valeurs localisées du niveau de pression sonore à l'intérieur, et la moyenne de ces valeurs en l'espace. Le niveau moyen de pression acoustique à l'intérieur est habituellement utilisé pour obtenir information de la prévision du bruit; cependant, dans certaines situations et applications réelles, la valeur du niveau de pression acoustique d'un point précis dans l'espace peut être souhaitable.

1. INTRODUCTION

The turbulent boundary layer (TBL) induced vibration in transport vehicles, particularly in aircraft, is a major source of interior noise, and thus an important topic of investigation. As confirmed by flight measurements in [1], the interior noise in the cabin of a jet transport aircraft, during cruise flight, is mostly generated by the external TBL excitation. While during takeoff the engine noise is the dominant source of cabin noise, the airflow sources become the major contribution for the interior noise during cruise flight. For subsonic flight, the TBL pressure levels on the exterior of the fuselage increase with the flight speed [2–4], representing a major source of aircraft interior noise for frequencies below 1000 Hz [1, 5]. Specifically, in [6], flight test measurements in an aircraft cockpit indicated that interior noise was dominated by low-frequency noise (<500 Hz), and that the main noise source was the external turbulent flow. As referred in [7], TBL excitation is regarded as the most important noise source for jet powered aircraft at cruise speed, particularly, as new quieter jet engines are being developed.

The main objective of the present work is the development of accurate analytical models for the prediction of TBL-induced noise in the interior of a real scale rectangular cabin. The physical system considered corresponds to a rectangular shaped acoustic enclosure, filled with air, with one flexible wall and five rigid walls. The flexible wall is composed of 50 identical simply supported panels. The dimensions and properties of the plates are similar to those of typical aircraft fuselage skin panels. A larger acoustic enclosure, compared with the plate's dimensions, was studied in order to simulate a more realistic approach of an aircraft fuselage section. Both the unpressurized and pressurized cabin are explored. The external flow excitation is representative of typical cruise conditions of a commercial aircraft, i.e., of the TBL wall pressure fluctuations in the aircraft fuselage, while in cruise and stabilized flight conditions. The TBL is assumed to be attached and completely developed over the aircraft structure. The amplitude of the wall pressure fluctuations is dependent on the TBL thickness, thus depends on the longitudinal position of the plate.

Previous work performed by the authors has validated the analytical models for single panel coupled with an acoustic enclosure, as in [8]. The analytical framework was successfully validated through comparison with several independent experimental studies. The present work adds a step forward compared to previous studies - it considers the TBL as the panel's excitation, while considering each panel (located at different positions in the flexible wall) coupled with a real scaled acoustic enclosure. The aim is to investigate the coupling between individual panels and the acoustic enclosure. It is shown that the position of the plate relative to the enclosure plays a crucial role in the accurate prediction of the interior pressure field. The difference between predicted space-averaged sound pressure level (SASPL) and predicted local sound pressure level (LSPL) is also explored.

The analytical formulation was developed from the fundamental equations and intrinsically derived as a structural-acoustic coupled model, i.e., it accounts for the natural modes of the plate and the acoustic modes of the enclosure. A convergence study was performed to calculate the minimum number of plate and acoustic modes needed for convergence of the predicted spectral quantities. Results were obtained for the prediction of vibration and sound pressure levels in the power spectral density (PSD) domain, up to a frequency of 1000 Hz. The model is able to predict the space-average plate displacement level, the space-average interior SPL, local plate displacement level (at a specified location on the panel surface), and local SPL. The occurrence of the hydrodynamic coincidence phenomenon is also investigated.

1.1 Turbulent Boundary Layer Excited Panels

Previous investigations on flow-induced noise and vibration have been reported, although, it is important to recognize how different these studies are, and how their nature can affect the development of a predictive mathematical model. Specifically, the study of the noise radiation by a isolated panel into free air, involves a different analysis compared with the study of cabin interior noise prediction due to the vibration of a panel. The later involves the coupling between the structural vibration and the cabin acoustic field. When the purpose is to develop analytical models for the prediction of cabin noise levels, one needs to consider the properties of: (1) the excitation, (2) the vibrating structure, and (3) the sound receiving room.

Several early experimental studies were performed to investigate the vibration and radiation of sound from structural panels, excited by the TBL, e.g. [9–12]. These studies provide knowledge about the shape of the spectrum, convection velocity and space-time correlation of the exterior TBL pressure fluctuations on aircraft panels, as well as displacement and acceleration spectra of the vibrating aircraft panels. Additionally, theoretical studies have also been performed to study the vibration and sound radiated by isolated panels (i.e., not coupled with an acoustic enclosure) excited by turbulent flows [13–17], and for random vibration of a single panel coupled with a small acoustic enclosure [7, 18, 19]. In these studies, the TBL excitation is usually described in terms of the statistical properties of the wall pressure fluctuations based on the Corcos formulation [20, 21]. Even though a number of new models were developed after the Corcos model for the TBL description, e.g. [22–25], the Corcos formulation is widely used in recent studies to describe the TBL pressure field [26–30], since it captures the fundamental pressure tendency along the frequency and requires significantly reduced computational effort to employ. Furthermore, Corcos formulation provides a good estimation for the TBL wall-pressure fluctuations levels at and near the convective peak, which is of fundamental importance for aircraft boundary layers (for high subsonic Mach numbers) [26], the case of the present study. In order to account with the streamwize variation of the boundary layer thickness, in the present study the Efimtsov model [22] is used to provide the point power spectrum. In the comparison of the several models available to describe the turbulent boundary layer wall pressure in [31], the model developed by Efimtsov is cited as a suitable candidate, being the only model derived from aircraft flight tests rather than laboratory measurements. More recently, flight tests performed in the Tupolev 144LL aircraft [32], demonstrated that the Efimtsov model shows the best agreement with the experimental data.

In the present study, the panels are considered to be simply supported in all four edges. Each panel represents the distance between adjacent stringers and frames (no additional stiffeners are considered), and is individually vibrating and coupled with the acoustic enclosure. As concluded in [11, 33], while jet noise induced vibration is highly correlated over several panels in both longitudinal and circumferential directions, the TBL induced vibration (in which the vibration correlation decays rapidly especially in the circumferential direction) is confined to one or two adjacent panels in the longitudinal direction. For the TBL excitation, the vibration of an isolated panel can be considered with the limitation that it is not necessarily valid at frequencies below the lowest natural frequency of a single bay of the fuselage structure (which in the present study is 61.44 Hz for the unpressurized cabin, and 355.45 Hz for the pressurized cabin).

1.2 Aircraft Cabin Noise Induced by Turbulent Flow

Several experimental studies were conducted to investigate the aircraft cabin interior noise induced by the TBL [34–37], providing measurements of the interior SPL and fuselage skin vibrations spectrum, at various locations in the cabin and cockpit of commercial aircraft, for aluminum and composite fuselages. These studies are a good database for comparison with theoretical predictions of interior noise levels induced by the TBL. The effect of aircraft speed on boundary layer induced interior noise can be seen to be dramatic, with the interior sound pressure levels being generally higher for higher flight speeds. The TBL wall pressure levels increase with the flight Mach number, as concluded in [2, 4, 33]. In the absence of hydrodynamic coincidence phenomenon, the interior noise level usually follows the same tendency, i.e., it increases with the flight Mach number [7, 16, 38]. In the presence of hydrodynamic coincidence, the tendency of increased interior noise with higher flight speeds is generally valid for frequencies below and above the neighborhood of frequency at which hydrodynamic coincidence occurs, as shown in [11].

Interior cabin noise is a challenging problem in most aircraft and many other transport vehicles. The reduction of cabin noise levels is desirable for both comfort and health-related reasons, and they are balanced with the cost, complexity, and physical constraints of noise control systems. As well known, passive noise control techniques are not effective in the low-frequency noise range, where the active noise control techniques have demonstrated better results, showing the ability to decrease sound levels without a big penalty in terms of weight. However, because of the complexity of the coupled structural-acoustic system, the implementation of these techniques is far from being straightforward. To efficiently design a noise control system, a clear understanding of the mechanisms of sound radiation and transmission of the coupled structural-acoustic system is crucial. In this context, the objective of the present study is to contribute for the understanding of the sound transmission phenomenon involved in the multi-panel structural-acoustic system.

2. MATHEMATICAL MODELS

In this section, the mathematical models developed are presented. Three models need to be defined: (1) the TBL model, which represents the external force applied to each panel, (2) the structural model, representing each individual plate coupled with the acoustic enclosure, and (3) the acoustic model, consisting of a rectangular acoustic cabin. In the following subsections, the mathematical models involved in the analysis are presented. Since previous work was performed in order to validate the analytical model, for simple systems, in this work the new developments on the model are emphasized. If the reader wishes to see more details of the mathematical manipulations involved in the analysis, please refer to [8].

2.1 Turbulent Flow Model

As previously referred, modeling the TBL wall pressure has been a subject of study for many years. Since the TBL is a random process, the resultant wall pressure, p(x, y, t), is usually statistically described in terms of the pressure power spectral density (PSD). These models were developed for turbulent flow over a flat plate, assuming fully developed flow and zero mean pressure gradient. For these conditions, the turbulent flow can be regarded as stationary in time and homogeneous in space. The cross-spectral density of the wall pressure over the (x, y) plane, for flow in the x-direction, can be defined through the Corcos formulation [20, 21], as follows

$$S(x,\xi_x,\xi_y,\omega) = S_{ref}(x,\omega)\, e^{-\frac{\alpha_x \omega |\xi_x|}{U_c}}\, e^{-\frac{\alpha_y \omega |\xi_y|}{U_c}}\, e^{-\frac{i \omega \xi_x}{U_c}}, \tag{1}$$

in which ξ_x and ξ_y are the spatial separations in the streamwise and spanwise directions of the plate, respectively, ω is the angular frequency, U_c is the convective speed of the TBL, and α_x and α_y are empirical parameters that denote the lost of coherence in the longitudinal and transverse directions, and are chosen to yield the best agreement with the reality. Recommended empirical values for aircraft boundary layers are $\alpha_x = 0.1$ and $\alpha_y = 0.77$ [31]. Note that, with relation to previous study [8], x was added as a variable for the $S(x,\xi_x,\xi_y,\omega)$ and $S_{ref}(x,\omega)$ terms. This variable needs to be added since the TBL pressure cross-spectral density dependents on the position of each panel; i.e., panels positioned at different x-coordinates have different S_{ref} values. Efimtsov model, defined in [22], is in good agreement with experimental data for the flow speed of interest in the present work, and provide the reference PSD as follows:

$$S_{ref}(x,\omega) = \frac{\tau_w^2(x)\, \delta(x)}{U_\tau(x)}\, \frac{0.01\,\pi}{1+0.02\, Sh^{2/3}(x,\omega)}, \tag{2}$$

with:

$$U_\tau(x) = U_\infty \sqrt{\frac{C_f(x)}{2}},\ \tau_w(x) = \frac{1}{2}\rho U_\infty^2 C_f(x),\ Sh(x,\omega) = \frac{\omega\, \delta(x)}{U_\tau(x)}, \tag{3}$$

where U_τ is the friction velocity, τ_w is the mean wall shear stress, C_f is the friction coefficient, δ is the boundary layer thickness, Sh is the Strouhal number, and U_∞ is the free-stream velocity. Functions $C_f(x)$ and $\delta(x)$ were computed using the following semi-empirical expressions for turbulent boundary layers, respectively from [39] and [9]:

$$C_f(x) = 0.37(\mathrm{Log}_{10}\mathrm{Re}_x)^{-2.584}, \tag{4a}$$

$$\delta(x) = 0.37 \text{ x } \mathrm{Re}_x^{-\frac{1}{5}}\left[1+\left(\frac{\mathrm{Re}_x}{6.9\times10^7}\right)^2\right]^{\frac{1}{10}}. \tag{4b}$$

Values of C_f and δ for each plate are shown in Fig. 1. Plates located at the same x-coordinate have same values of C_f and δ. In these figures, points x_1 through x_5 correspond to the panel center locations from the first to the fifth rows of panels along x-direction. Points x_i and x_f correspond, respectively, to the x-coordinate in which the first row of panels starts (i.e., to x = 9.14 m) and to the x coordinate in which the last row of panels ends (i.e., x = 11.64 m) - refer to Fig. 3 for more details about the physical system under study. Fig. 2 shows $S_{ref}(x,\omega)$ for different flight speeds and altitudes, given by Eq.(2), for the fifth row of panels. For this row of panels, the curve corresponding to the present study is the one with solid line and bold circle – refer to Table 2 for more information about external fluid parameters. As concluded in [32], the predictions provided by the Efimtsov model, using Eq.(2), show a weaker decay than the measured data at high frequencies (above 1000 Hz), over predicting the spectral quantities above this frequency. Since the present study considers only frequencies up to 1 KHz, this problem does not significantly affect the results.

2.2 Panels Displacement Model

All plates are considered to be flat panels, simply supported in all four boundaries (as in Fig. 3b) and are assumed to represent the distance between adjacent stringers and frames of a conventional aircraft skin-stringer-frame structure. The displacement of each panel is defined in terms of its natural modes, as follows

$$w(x,y,t) = \sum_{m_x=1}^{M_x}\sum_{m_y=1}^{M_y} \alpha_{m_x}(x)\,\beta_{m_y}(y)\,q_{m_x m_y}(t)\,, \tag{5}$$

where $\alpha_{m_x}(x)$ and $\beta_{m_y}(y)$ functions define the variation of w with the x and y, respectively, $q_{m_x m_y}(t)$ define the variation of w with t, and $M = M_x \times M_y$ is the total number of plate modes considered in the analysis. As shown in Fig. 3a, the direction of the plate displacement, w, was chosen to be the positive z-direction. Since the panels are simply supported plates, the spatial functions may be defined, in the plate (local) coordinates system, as:

$$\alpha_{m_x}(x_1) = \sqrt{\frac{2}{a}}\,\sin\left(\frac{m_x\,\pi\,x_1}{a}\right), \tag{6a}$$

$$\beta_{m_y}(y_1) = \sqrt{\frac{2}{b}}\,\sin\left(\frac{m_y\,\pi\,y_1}{b}\right), \tag{6b}$$

in which a and b are the dimensions of the plate in the x- and y-directions.

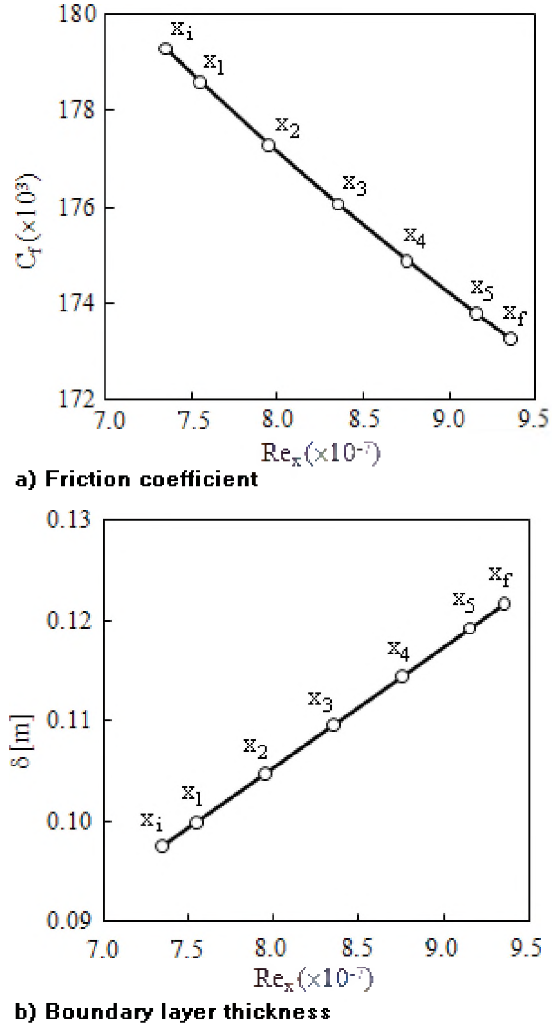


Figure 1. Comparison of the C_f and δ values along the several x panel rows.

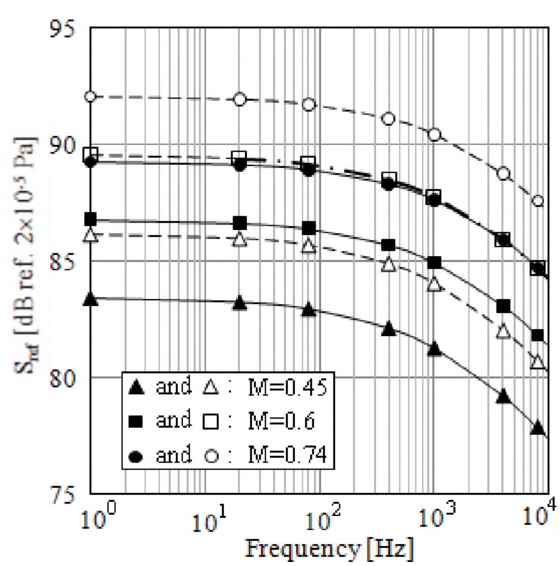


Figure 2. Reference PSD obtained from Efimtsov model, for altitudes 25000 ft (solid lines) and 16400 ft (dashed lines).

However, in order to compute the structural-acoustic coupling, it is convenient to work in the enclosure (global) coordinates system. To accomplish that, Eqs.(6) can be written in the global coordinates system, for each individual plate, as follows:

$$\alpha_{m_x}(x) = \sqrt{\frac{2}{a}}\,\sin\left(\frac{m_x\,\pi\,(x-x_{p_i})}{a}\right), \tag{7a}$$

$$\beta_{m_y}(y) = \sqrt{\frac{2}{b}}\,\sin\left(\frac{m_y\,\pi\,(y-y_{p_i})}{b}\right), \tag{7b}$$

where (x_{pi}, y_{pi}) is the position of the origin of the i^{th} plate coordinates system, written in the global coordinates system. The natural frequencies for the simply supported untensioned plate (unpressurized cabin) and tensioned plate (pressurized cabin), are given respectively by:

$$\omega^{p}_{m_x m_y} = \sqrt{\frac{D_p}{\rho_p h_p}}\,\left[\left(\frac{m_x\,\pi}{a}\right)^2 + \left(\frac{m_y\,\pi}{b}\right)^2\right], \tag{8a}$$

$$\omega^{p}_{m_x m_y} = \sqrt{\frac{1}{\rho_p h_p}\left[D_p({f_x}^2 + {g_y}^2)^2 + T_x {f_x}^2 + T_y {g_y}^2\right]}, \tag{8b}$$

in which ρ_p is the density of the panel, h_p is its thickness, $D_p = \frac{E_p h_p^3}{12(1-\nu_p^2)}$ is the panel stiffness constant, T_x and T_y are the in-plane tensions in x- and y-directions, respectively, and $f_x = \frac{m_x \pi}{a}$ and $g_y = \frac{m_y \pi}{b}$. The plate governing equations for a given applied external pressure, for the unpressurized and pressurized cabins, are respectively:

$$D_p \nabla^4 w + \rho_p\, h_p\, \ddot{w} + \zeta_p\, \dot{w} = p_{ext}(x,y,t), \tag{9a}$$

$$D_p \nabla^4 w + \rho_p\, h_p\, \ddot{w} + \zeta_p \dot{w} - (T_x {f_x}^2 + T_y {g_y}^2)\, w = p_{ext}(x,y,t), \tag{9b}$$

where ζ_p was added to account for the damping of the plate, and w is given by Eq.(5).

2.3 Acoustic Cabin Pressure Model

Following the same approach as for the plate's displacement models, the pressure inside the enclosure is defined in terms of the acoustic modes, as follows:

$$p(x,y,z,t) = \sum_{n_x=1}^{N_x} \sum_{n_y=1}^{N_y} \sum_{n_z=1}^{N_z} \psi_{n_x}(x)\, \phi_{n_y}(y)\, \Gamma_{n_z}(z)\, r_{n_x n_y n_z}(t), \tag{10}$$

where $\psi_{n_x}(x)$, $\phi_{n_y}(y)$ and $\Gamma_{n_z}(z)$ are spatial functions, $r_{n_x n_y n_z}(t)$ are functions of t, and $N = (N_x+1)\times(N_y+1)\times(N_z+1)$ is the number of acoustics modes considered. As shown in Fig. 3a, the direction of the interior pressure, p, was chosen to be the positive z-direction. The individual spatial functions are assumed to be orthogonal between each other, and given by the rigid body enclosure modes, as follows:

$$\psi_{n_x}(x) = \frac{A_{n_x}}{\sqrt{L_x}}\cos\left(\frac{n_x \pi\, x}{L_x}\right), \tag{11a}$$

$$\phi_{n_y}(y) = \frac{A_{n_y}}{\sqrt{L_y}}\cos\left(\frac{n_y \pi\, y}{L_y}\right), \tag{11b}$$

$$\Gamma_{n_z}(z) = \frac{A_{n_z}}{\sqrt{L_z}}\cos\left(\frac{n_z \pi\, z}{L_z}\right), \tag{11c}$$

where L_x, L_y and L_z are the dimensions of the enclosure in the x-, y- and z-directions, respectively, and the constants A_n were chosen to satisfy normalization. For a rectangular cavity, the natural frequencies are determined as follows:

$$\omega^{ac}_{n_x n_y n_z} = c_0 \sqrt{\left(\frac{n_x\,\pi}{L_x}\right)^2 + \left(\frac{n_y\,\pi}{L_y}\right)^2 + \left(\frac{n_z\,\pi}{L_z}\right)^2}, \tag{12}$$

where c_0 is the speed of sound inside the acoustic enclosure. The governing equation is the wave equation, defined by

$$\nabla^2 p - \frac{1}{c_0^2}\,\ddot{p} - \zeta_{ac}\,\dot{p} = 0, \tag{13}$$

in which ζ_{ac} was added to account for the acoustic damping in the enclosure.

2.4 Coupled System Model

To obtain the governing equations of the coupled structural-acoustic system, the equations of the individual uncoupled subsystems should be combined (please refer to [8] for the coupling details). The equations of the coupled plate-enclosure system can be written together in the matrix form, as follows:

$$\begin{bmatrix}\mathbf{M}_{pp} & \mathbf{0}\\ \mathbf{M}_{cp} & \mathbf{M}_{cc}\end{bmatrix}\begin{Bmatrix}\ddot{\mathbf{q}}(t)\\ \ddot{\mathbf{r}}(t)\end{Bmatrix} + \begin{bmatrix}\mathbf{D}_{pp} & \mathbf{0}\\ \mathbf{0} & \mathbf{D}_{cc}\end{bmatrix}\begin{Bmatrix}\dot{\mathbf{q}}(t)\\ \dot{\mathbf{r}}(t)\end{Bmatrix} + \begin{bmatrix}\mathbf{K}_{pp} & \mathbf{K}_{pc}\\ \mathbf{0} & \mathbf{K}_{cc}\end{bmatrix}\begin{Bmatrix}\mathbf{q}(t)\\ \mathbf{r}(t)\end{Bmatrix} = \begin{Bmatrix}\mathbf{p}_{tbl}(t)\\ \mathbf{0}\end{Bmatrix}, \tag{14}$$

with:

$$\mathbf{M}_{pp} = \mathrm{diag}\left[\rho_p\, h_p\right] \quad \text{and} \quad \mathbf{M}_{cc} = \mathrm{diag}\left[\frac{1}{c_0^2}\right], \tag{15a}$$

$$\mathbf{M}_{cp} = \rho_0 \left[\frac{(-1)^{n_z} A_{n_z}}{\sqrt{L_z}} \int_{x_{p_i}}^{x_{pf}} \alpha_{m_x}(x)\psi_{n_x}(x)dx \int_{y_{p_i}}^{y_{pf}} \beta_{m_y}(y)\phi_{n_y}(y)dy\right], \tag{15b}$$

$$\mathbf{D}_{pp} = \mathrm{diag}\left[2\rho_p h_p \omega_m \xi_p\right] \quad \text{and} \quad \mathbf{D}_{cc} = \mathrm{diag}\left[2\frac{1}{c_0^2}\omega_n \xi_{ac}\right], \tag{15c}$$

$$\mathbf{K}_{pp} = \mathrm{diag}\left[\omega_m^2\ \rho_p\ h_p\right] \quad \text{and} \quad \mathbf{K}_{cc} = \mathrm{diag}\left[\omega_n^2\ \frac{1}{c_0^2}\right], \tag{15d}$$

$$\mathbf{K}_{pc} = -\left[\frac{(-1)^{n_z} A_{n_z}}{\sqrt{L_z}} \int_{x_{p_i}}^{x_{p_f}} \alpha_{m_x}(x)\psi_{n_x}(x)dx \int_{y_{p_i}}^{y_{p_f}} \beta_{m_y}(y)\phi_{n_y}(y)dy\right], \tag{15e}$$

$$\mathbf{p}_{tbl}(t) = -\left[\int_{y_{p_i}}^{y_{p_f}} \int_{x_{p_i}}^{x_{p_f}} \alpha_{m_x}(x)\ \beta_{m_y}(y)\ p_{tbl}(x,y,t)\ dx\ dy\right]. \tag{15f}$$

In the previous equations, **M** represents mass matrices, **D** damping matrices, **K** stiffness matrices, subscripts *p* and *c* correspond respectively to *plate* and *cavity*. All matrices and vectors expressions were obtained analytically as shown in [8]. Additionally, $\zeta_p = 2\omega_m\xi_p$ is the structural modal damping and $\zeta_{ac}=2\omega_n\xi_{ac}$ is the acoustic modal damping. For notation simplicity, $\omega^p_{m_x m_y}$, $\omega^{ac}_{n_x n_y n_z}$, $q_{m_x m_y}(t)$, and $r_{n_x n_y n_z}(t)$ were substituted, respectively, by ω_m , ω_n, $q_m(t)$ and $r_n(t)$. Note that ω_m, in Eqs.(15), is given by Eq.(8a) for the unpressurized cabin and by Eq.(8b) for the pressurized cabin. The cross terms, i.e., matrices $\mathbf{M}_{cp}$ and $\mathbf{K}_{pc}$, are responsible for the coupling between each panel displacement and the enclosure pressure. Note that Eq.(14) accounts for the coupling of only one plate with the enclosure, i.e., the contribution of each panel for the interior sound pressure level is individually analyzed.

3. SOLUTION IN THE SPECTRAL DOMAIN

Since the TBL wall pressure field model is expressed in the frequency domain, one needs to transform the equation of the coupled system, Eq.(14), from the time domain to the frequency domain. As introduced in [8], this can be performed by assuming $q_m = Q_m\ e^{i\omega t}$ and $r_n = R_n\ e^{i\omega t}$. The spectral density of the system response is then given by:

$$\mathbf{S_{YY}}(\omega) = \mathbf{H}^*(\omega)\ \mathbf{S_{XX}}(\omega)\ \mathbf{H}^T(\omega)\ , \tag{16}$$

in which superscripts * and T denote, respectively, Hermitian conjugate and matrix transpose, $\mathbf{S_{XX}}(\omega)$ is the spectral density matrix of the excitation, and $\mathbf{S_{YY}}(\omega)$ is the spectral density of the system response. For mathematical convenience, matrix $\mathbf{S_{YY}}(\omega)$ can be divided in two matrices: (1) the PSD matrix of the coupled plate displacement, $\mathbf{S_{WW}}(\omega)$, and (2) the PSD matrix of the coupled acoustic enclosure pressure, $\mathbf{S_{PP}}(\omega)$. Similarly, matrix $\mathbf{S_{XX}}(\omega)$ can be divided in: (1) the PSD matrix of the turbulent boundary layer excitation, $\mathbf{S_{tbl}}(\omega)$, and (2) a null matrix. Considering this, Eq.(16) can be written in the following separate form:

$$\mathbf{S_{WW}}(\omega) = \mathbf{H_W}^*(\omega)\ \mathbf{S_{tbl}}(\omega)\ \mathbf{H_W}^T(\omega)\ , \tag{17a}$$

$$\mathbf{S_{PP}}(\omega) = \mathbf{H_P}^*(\omega)\ \mathbf{S_{tbl}}(\omega)\ \mathbf{H_P}^T(\omega)\ , \tag{17b}$$

where matrices $\mathbf{H_W}(\omega)$ and $\mathbf{H_P}(\omega)$ are defined by:

$$\mathbf{H_W}(\omega) = (\mathbf{A} - \mathbf{B}\ \mathbf{D}^{-1}\ \mathbf{C})^{-1}, \tag{18a}$$

$$\mathbf{H_P}(\omega) = -\ \mathbf{D}^{-1}\ \mathbf{C}\ \mathbf{H_W}(\omega)\ , \tag{18b}$$

and

$$\mathbf{A} = -\ \omega^2\ \mathbf{M}_{pp} + i\ \omega\ \mathbf{D}_{pp} + \mathbf{K}_{pp}\ , \tag{19a}$$

$$\mathbf{B} = \mathbf{K}_{pc}\ , \tag{19b}$$

$$\mathbf{C} = -\ \omega^2\ \mathbf{M}_{cp}\ , \tag{19c}$$

$$\mathbf{D} = -\ \omega^2\ \mathbf{M}_{cc} + i\ \omega\ \mathbf{D}_{cc} + \mathbf{K}_{cc}\ . \tag{19d}$$

The generalized PSD matrix of the TBL excitation, $\mathbf{S_{tbl}}(\omega)$, the PSD function of the plate´s displacement, and the PSD function of the enclosure interior pressure are defined, respectively, as follows:

$$\mathbf{S_{tbl}}(\omega) = \left[\int_{S_p}\int_{S_p} \alpha_{m_x}(x)\alpha_{m_{x'}}(x')\beta_{m_y}(y)\beta_{m_{y'}}(y')S(x,\xi_x,\xi_y,\omega)dSdS'\right], \tag{20}$$

$$S_{ww}(x_1, y_1, x_2, y_2, \omega) = \sum_{m_{x_1}, m_{x_2}=1}^{M_x^2} \sum_{m_{y_1}, m_{y_2}=1}^{M_y^2} \alpha_{m_{x_1}}(x_1)\alpha_{m_{x_2}}(x_2)\beta_{m_{y_1}}(y_1)\beta_{m_{y_2}}(y_2)\mathbf{S_{WW}}(\omega)_{m_1,m_2} \tag{21}$$

$$S_{pp}(x_1, y_1, z_1, x_2, y_2, z_2, \omega) = \sum_{n_{x_1}, n_{x_2}=1}^{N_x^2} \sum_{n_{y_1}, n_{y_2}=1}^{N_y^2} \sum_{n_{z_1}, n_{z_2}=1}^{N_z^2} \psi_{n_{x_1}}(x_1)\psi_{n_{x_2}}(x_2)\phi_{n_{y_1}}(y_1)\phi_{n_{y_2}}(y_2)\Gamma_{n_{z_1}}(z_1)\Gamma_{n_{z_2}}(z_2)\ \mathbf{S_{PP}}(\omega)_{n_1,n_2} \tag{22}$$

in which $S_p = a \times b$ is the plate surface area. Eqs.(21) and (22) can be used to predict the displacement PSD at a chosen point in the plate, and the pressure PSD at any given location of the acoustic enclosure, respectively. Finally, the space-averaged PSD functions can be found by integrating the individual power spectral densities over the plate area and the cavity volume, respectively, as:

$$S_{ww}(\omega) = \int_{S_p}\int_{S_p} S_{ww}(x_1, y_1, x_2, y_2, \omega)\ dS_1 dS_2, \tag{23}$$

$$S_{pp}(\omega) = \int_{V_c}\int_{V_c} S_{pp}(x_1,y_1,z_1,x_2,y_2,z_2,\omega)\ dV_1 dV_2, \tag{24}$$

in which $V_c = L_x \times L_y \times L_z$ is the enclosure volume. The analytical expressions derived for $\mathbf{S_{tbl}}(\omega)$, $S_{ww}(\omega)$ and $S_{pp}(\omega)$ are shown [8].

4. PHYSICAL SYSTEM

The geometry of the complete system is shown in Fig. 3. The turbulent flow is developed across the plates, at $z = L_z$, in the positive x-direction. The dimensions of the system are given in Table 1 and the physical parameters, including the external fluid, the plate, and the acoustic enclosure are listed in Table 2. The properties of the plates are for aluminum, and the parameters of the external fluid correspond to a cruise flight altitude of 25000 ft (i.e., 7628 m). Damping ratios for the structure and for the acoustic field of 1% were chosen to be representative of those in an aircraft [2, 26]. A large acoustic enclosure compared with the plates was chosen in order to simulate a more realistic approach of an aircraft cabin section. The flexible wall of the enclosure is composed by 50 simply supported identical plates (same dimensions and properties), with 5 panel rows along x-direction and 10 panel rows along y-direction. The panels have dimensions and properties similar to that of a typical commercial aircraft fuselage panel, representing the distance between adjacent frames and stringers. The fuselage cabin section is considered to start at x = 9.14 m from the nose of the aircraft, in order to consider a more realistic case of an aircraft fuselage section - as shown in [2], this is the start point of the forward section of a Boeing 727-200 airplane fuselage.

Table 1. Dimensions of the system.

Description	Symbol	Value, m
Plate length	a	0.5
Plate width	b	0.22
Plate thickness	h_p	0.00102
Enclosure length	L_x	2.5
Enclosure width	L_y	2.2
Enclosure height	L_z	2.1

Table 2. Properties of the physical system.

Description	Symbol	Value
External Fluid:		
Air density	ρ	0.54 Kg m^{-3}
Air kinematic viscosity	ν	2.85×10^{-5} m^2 s^{-1}
Speed of sound	c	309.6 m s^{-1}
Free stream velocity	U_∞	229.104 m s^{-1}
Convective velocity	U_c	0.7 U_∞
Empirical parameter	α_x	0.1
Empirical parameter	α_y	0.77
Panels:		
Density	ρ_p	2800 Kg m^{-3}
Elasticity Modulus	E_p	7.24×10^{10} Pa^2
Poisson's ratio	ν	0.33
Damping ratio	ξ_p	0.01
Longitudinal tension	T_x	29300 N m^{-1}
Lateral tension	T_y	62100 N m^{-1}
Number of modes	M	
- unpressurized cabin:		44 (M_x=11, M_y=4)
- pressurized cabin:		27 (M_x=9, M_y=3)
Acoustic Enclosure:		
Air density	ρ_0	1.2 Kg m^{-3}
Speed of sound	c_0	340 m s^{-1}
Damping ratio	ξ_{ac}	0.01
Number of modes	N	2912 (N_x=16, N_y=14, N_z=13)

5. RESULTS

5.1 Convergence

A convergence study was performed to determine the number of structural and acoustic modes required for the calculation of the spectral quantities. It was found that, for the frequency range of interest, [0; 1000] Hz, some non-resonant modes need to be considered.

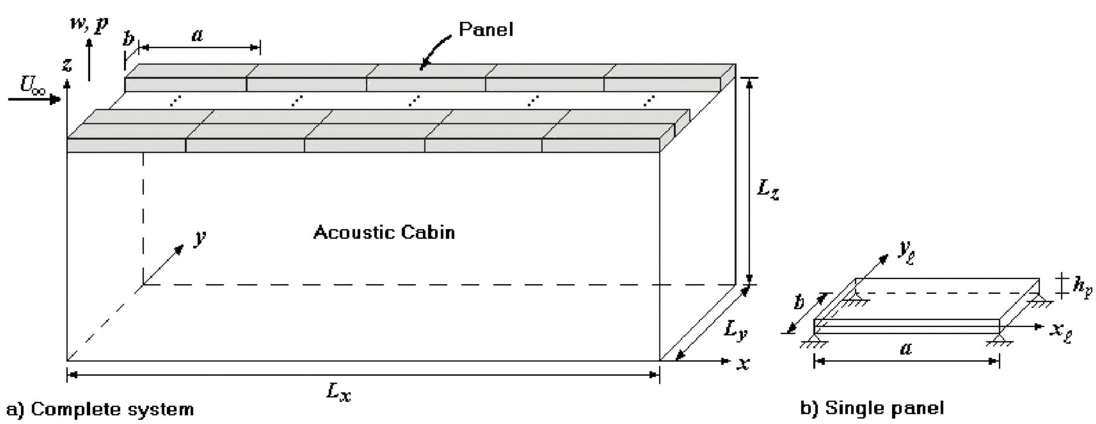


Figure 3. Geometry of the physical system.

A simple criterion to determine the number of structural modes required for convergence of the modal series up to a frequency f_{max} is the following: convergence is reached when the distance between two nodes of the structural mode shape is less than or equal to the half-wavelength of the bending wave on the plate, $\lambda_b/2$, at the analysis frequency, i.e.,

$$\frac{a}{M_x} \leq \frac{\lambda_b(\omega)}{2}, \tag{25}$$

$$\lambda_b(\omega) = 2\,\pi\,\omega^{-\frac{1}{2}} \left(\frac{D_p}{\rho_p h_p}\right). \tag{26}$$

Thus, for f_{max}=1000 Hz, from Eq.(25) one obtains $M_x \geq 11$. Another convergence criterion is presented in [16] - for a given upper frequency f_{max}, the number of modes (M_x,M_y) required for the calculation of the spectral quantities, for a tensioned panel, can be calculated by

$$\left\{\left[\left(\frac{M_x}{a}\right)^2 + \left(\frac{M_y}{b}\right)^2\right]^2 + \frac{T_x}{D_p}\left(\frac{M_x}{a}\right)^2 + \frac{T_y}{D_p}\left(\frac{M_y}{b}\right)^2\right\}^{1/4} \geq \left(\frac{\rho_p h_p}{D_p}\right)^{1/4} \sqrt{\frac{2\,f_{max}}{\pi}}. \tag{27}$$

For the untensioned plate case, the convergence criterion is obtained from Eq.(27) with in-plane tensions equal to zero. For the aircraft panel considered in the present study (for f_{max}=1000 Hz), the number of structural modes required to accurately calculate the PSD of the panel response is M_x=11 and M_y=4 for the untensioned plate, and M_x=9 and M_y=3 for the tensioned plate.

Table 3 displays the first 20 natural frequencies of the untensioned panel, and the corresponding frequencies for the in-tension panel. The number of enclosure acoustic modes required for the accurate calculation of the spectral quantities was also determined. Similarly to the plate, convergence is reached when the distance between two nodes of the acoustic mode shape is less or equal than half-wavelength of the acoustic wave in the interior of the enclosure, i.e.,

$$\frac{L_x}{N_x} \leq \frac{c_0}{2\,f_{max}}. \tag{28}$$

Thus, from Eq.(28) one obtains $Nx \geq 15$. For the aircraft cabin considered and f_{max}=1000 Hz, the number of acoustic modes required to accurately calculate the PSD of the acoustic response is N_x=16, N_y=14 and N_y=13. The 2912 acoustic modes considered are plotted in Fig. 4, as well as the plate's mode lines for the untensioned and in-tension cases. As can be seen, several non-resonant modes need to be considered.

5.2 Hydrodynamic Coincidence

To study the effect of hydrodynamic coincidence, it is important to first identify the degree of matching between the boundary layer and the plate modes. Figure 5 shows the plate natural frequencies plotted against longitudinal mode number, m_x, for modes with m_y=1,...,4. Also plotted in this figure are the hydrodynamic coincidence lines (representing $f = m_x U_c/2a$) for three cases: $U_c = 0.7\,U_\infty$ (reference case), $U_c = 0.75\,U_\infty$ and $U_c = 0.8\,U_\infty$.

Table 3. Panels first 20 natural frequencies [Hz].

(m_x,m_y)	Untensioned panel	In-tension panel
(1,1)	61.44	355.45
(2,1)	91.34	402.11
(3,1)	141.17	473.89
(4,1)	210.93	566.52
(1,2)	215.87	711.41
(2,2)	245.77	742.08
(3,2)	295.60	793.07
(5,1)	300.62	677.62
(4,2)	365.36	864.22
(6,1)	410.25	806.19
(5,2)	455.05	955.41
(1,3)	473.26	1115.82
(2,3)	503.15	1142.37
(7,1)	539.81	952.05
(3,3)	552.98	1186.99
(6,2)	564.68	1066.51
(4,3)	622.75	1250.11
(8,1)	689.29	1115.36
(7,2)	694.24	1197.46
(5,3)	712.44	1332.24

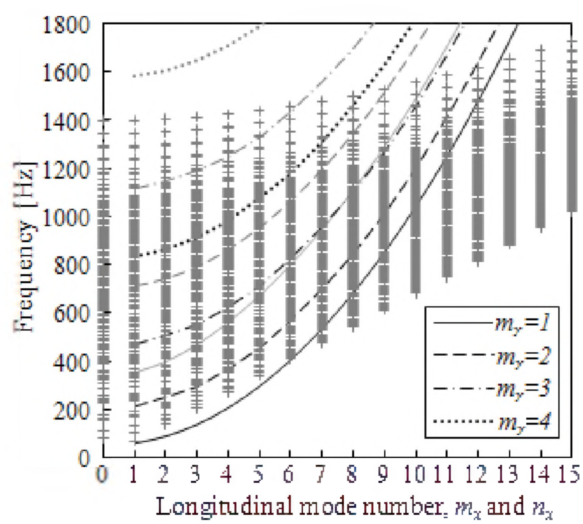
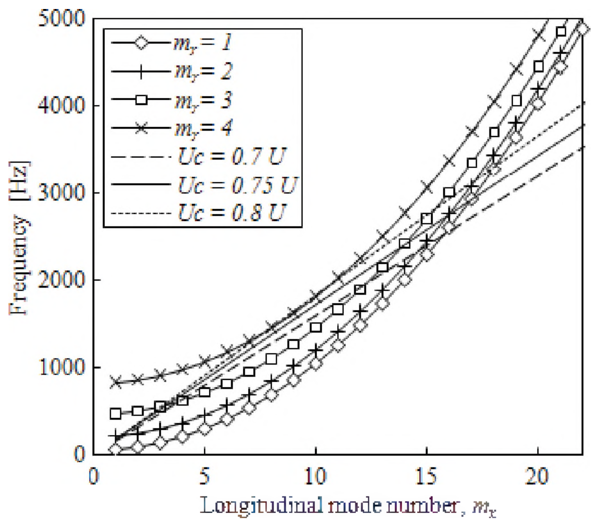
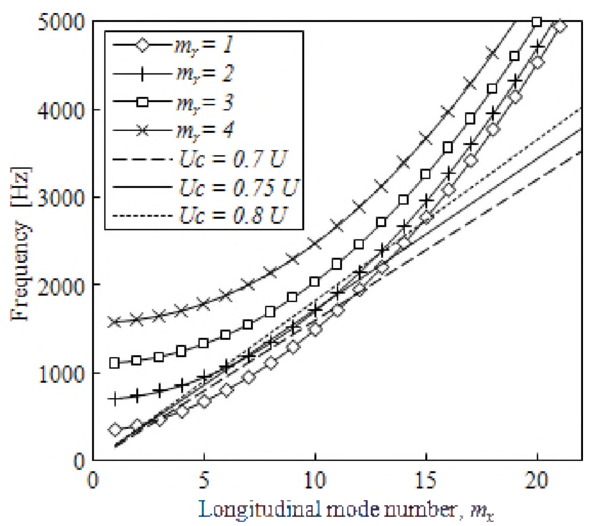


Figure 4. Matching between acoustic modes (+) and plates modes: untensioned (black lines), tensioned (grey lines).

From Fig. 5 one can confirm that hydrodynamic coincidence lines 'match' the plate modes over a large part of the frequency range, mainly for the untensioned plate case. As explained in [40], this confirms the importance of inefficient, but resonant and highly excited modes in the aircraft noise problem. In the reference case, for the untensioned case, the plate modes (4,3), (12,3), (15,2) and (16,1) provide the best matching with the turbulent convecting scales ($m_x U_c/a\omega \approx 1$), and are thus highly excited modes. Of these 4 modes only (4,3) has a resonant frequency in the range of study, [0; 1000] Hz. For the tensioned case, the plate mode (3,1) provides the best matching with the hydrodynamic coincidence line. In the reference case, the hydrodynamic coincidence frequency, $f_c=\frac{U_c^2}{2\pi}\sqrt{\frac{\rho_p h_p}{D_p}}$, is 2580.74 Hz. Thus, in the frequency range under study all resonant and nonresonant modes considered are inefficient radiators.



a) Untensioned Panel



b) In-tension Panel

Figure 5. Matching between hydrodynamic coincidence lines and plate natural frequencies lines.

5.3 Frequency Resolution

All spectral quantities were obtained for the frequency range [0; 1000] Hz. The frequency resolution for the PSD calculations was obtained through an adaptive algorithm to meet the damping coefficient constraint, both for the untensioned and tensioned plates. This algorithm was developed to guarantee enough frequency resolution to resolve all resonance peaks within the frequency range covered (maximum frequency was determined based on the convergence study), for both structural and acoustic modes.

5.4 Predicted Structural Vibration Levels

The space-averaged plates displacement power spectral density (ADPSD), expressed by Eq.(23), and the plate displacement power spectral density (DPSD) in several points on the plates, given by Eq.(21), were obtained.

Figure 6 shows the ADPSD for the panel (1,1), i.e., panel located at first row of panels and first row of columns. Panels in other locations have similar ADPSD, with panels located at bigger x-coordinates having a slightly higher ADPSD at all frequencies. This can be explained since an increase in x-station results in a higher value for the reference PSD of the TBL excitation. The first 3 ADPSD peaks correspond to the bending modes (1,1), (3,1) and (5,1), for both untensioned and tensioned plates. Additionally, considering pressurization effects results in a decreased radiated ADPSD for lower frequencies compared with the unpressurized cabin. For the untensioned panel, a large response due to resonant amplification of (4,3) plate mode does not occur. This can be explained because, in the present case of study, hydrodynamic coincidence is not well tuned at frequencies where the hydrodynamic matching line broadly coincides with the resonant modes. For the tensioned plate, the mode (3,1), which provides the best matching with the hydrodynamic coincidence line, corresponds to the second ADPSD peak. This may be explained because (3,1) is the only plate mode which provides 'matching' with the hydrodynamic coincidence line, while for the untensioned plate a larger number of modes provide this matching.

The results for the DPSD are shown in Fig. 7, for three different locations in the surface of the plate (1,1), specifically at: $(x_1, y_1)=(0.25,0.11)$m, $(x_2, y_2)=(0.1,0.06)$m, and $(x_3, y_3)=(0.4,0.06)$m. DPSD plots for the other plates show similar results and, as for the ADPSD, it shows a slight overall increase with the increase of the x-coordinate. As shown in this figure, point 1, located at the center of the panel, follow the same line as the ADPSD, with the peaks located at the same frequencies. However, the same does not occur for the other points considered, in which additional peaks can be observed for the DPSD curves. This can be explained since the point at the center of the plate is not affected when the longitudinal mode number, m_x, or the lateral mode number, my, is even. Thus, when evaluating the PSD of the plate response, it is important to know the position of interest in the plate, since its value is dependent on the position of measurement. Comparing points 2 and 3 (both located at y = 0.06 m), one can conclude that points at higher x have generally bigger DPSD.

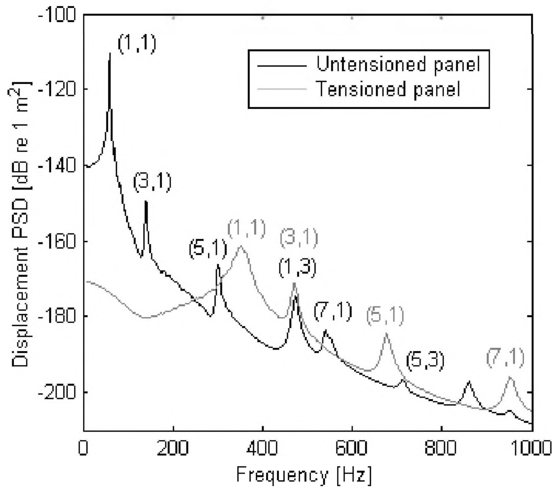
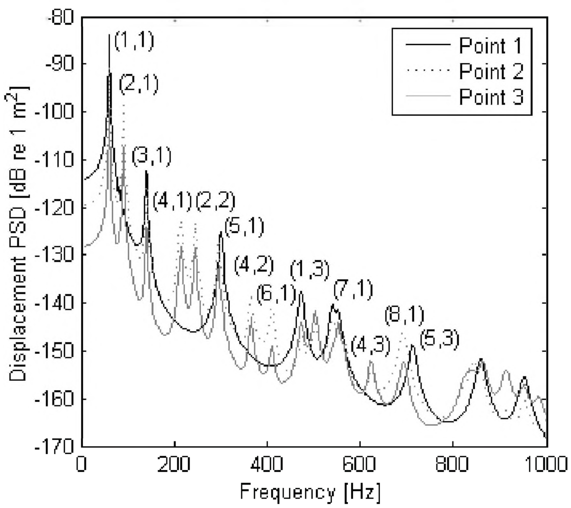
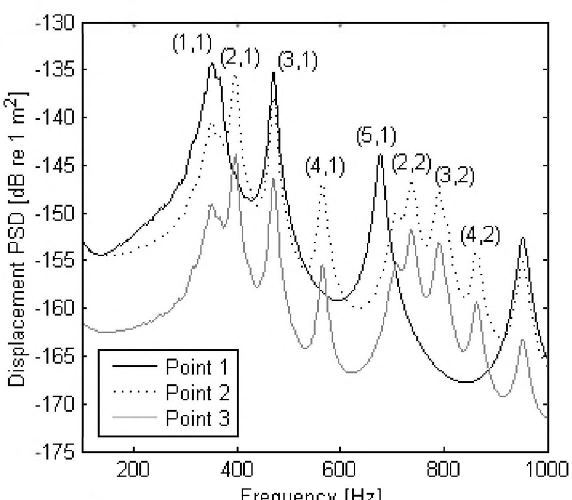


Figure 6. Space-averaged displacement PSD for plate (1,1).



a) Untensioned Panel



b) Tensioned Panel

Figure 7. Displacement PSD at 3 points located at the surface of panel (1,1).

5.5 Predicted Cabin Sound Pressure Levels

The acoustic enclosure space-averaged pressure power spectral density (APPSD) and the acoustic pressure power spectral density (PPSD) at specific point in the enclosure, were obtained through Eqs.(24) and (22), respectively. The acoustic enclosure APPSD, due to the individual contribution of the panels, located at two different positions in the flexible wall, is shown in Fig. 8. An important conclusion to draw from this figure is that some peaks correspond to plate natural frequencies and other to acoustic natural frequencies. This illustrates the importance of the structural-acoustic coupling for accurate prediction of the internal pressure in the interior of an enclosure, such as for example an aircraft cabin. The uncoupled study of the sound radiated by an individual plate, vibrating due to turbulent flow, does not give the total information when the main goal is to predict aircraft interior noise. Another conclusion is that plates located at different positions have dissimilar contributions to the enclosure interior pressure levels. For instance, plate (3,7) has a negligible contribution in the amplification of the acoustic mode (1,0,0) compared with plate (1,1). Since plates in row 3 are located in the centre of the enclosure in the x-direction, they do not add significant contribution to the frequency associated to this mode, which has a node at centre of the enclosure in this direction. For the same reason, plates in row 3 have a decreased contribution for the amplification of all other modes with longitudinal mode number, n_x, equal to 1, compared with the other plates. All other modes (i.e., with longitudinal mode number equal to 0 or to 2) have similar mode amplification, since for modes with $n_x = 2$ the middle point correspond to an antinode, while for modes with $n_x = 0$ the pressure is constant along x-direction. For untensioned plates the interior SPL is bigger for lower frequencies, while for tensioned panels the maximum SPL is observed at higher frequencies. This occurs since the first plate mode increases from 61.44 Hz, in the unpressurized case, to 355.45 Hz in the pressurized case. However, for the pressurized case, frequencies below 355.45 Hz cannot be ignored, since several acoustic modes are amplified below this frequency.

Figure 9 shows the results for the interior pressure power spectral density (PPSD), at four chosen points inside the enclosure, due to the individual radiation of plates, located at four different positions - specifically, plates (1,1), (3,1), (3,5), and (5,1) are analyzed. The points inside the enclosure under study are the following (defined in the global coordinates system): $(x_1, y_1, z_1)=(x_{pi}+a/2, y_{pi}+b/2, 2)$ and $(x_2, y_2, z_2)=(x_{pi}+a/2, y_{pi}+b/2, 1)$, $(x_3, y_3, z_3)=(1,1,2)$, and $(x_4, y_4, z_4)=(1,1,1)$. Note that points 1 and 2 are different for each plate, with x_{pi} and y_{pi} corresponding to the initial x and y coordinates of each plate. It can be observed that point (x_1, y_1, z_1) has higher PPSD at almost all frequencies, compared with the other points. As expected, decreasing z-coordinate (i.e., moving away from the plates) results in lower PPSD values. It is interesting to verify that the structural-acoustic coupling has an important role in the prediction of the interior SPL.

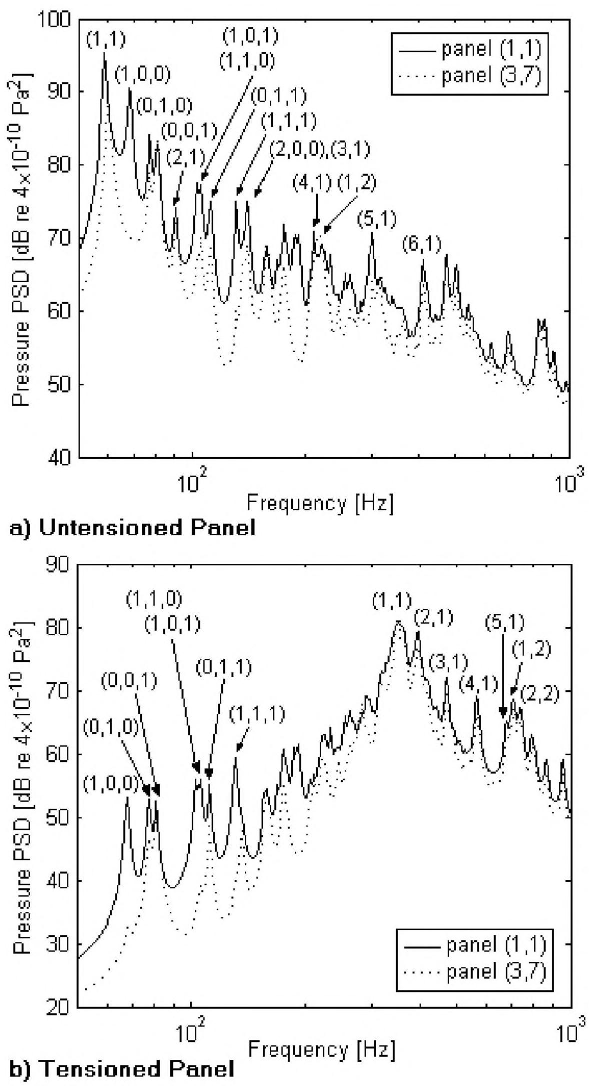


Figure 8. Space-averaged pressure PSD, for the contribution of panels (1,1) and (3,7).

Analyzing the results for the four different plates, and the same observing point (x_1, y_1, z_1), one can verify that the PPSD plot has some variations from plate to plate. Since point (x_1, y_1, z_1) is always at the same relative position at each plate, that difference can only be due to the enclosure acoustic modes. If only the plate modes were considered, one would obtain the similar curves for all plates in point (x_1, y_1, z_1), and as well in point (x_2, y_2, z_2), which is not what is observed. The fact that each plate is in a different position with relation to the enclosure global coordinate system, changes the way it couples with the acoustic enclosure. This explains why plates (1,1) and (5,1) have similar curves for points (x_1, y_1, z_1) and (x_2, y_2, z_2). As concluded for the DPSD, when evaluating the PSD of the interior pressure is important to know which is the position of interest in the enclosure, since the SPL value is dependent on the position of measurement.

6. CONCLUSIONS

An analytical study to predict the turbulent boundary layer-induced noise in the interior of a rectangular enclosure with one flexible wall, consisting of several panels, is presented. Predictions of the space-averaged PSD and localized PSD were obtained for the displacement of the plates and interior acoustic pressure in the enclosure. The characteristics of the physical system were selected to represent an aircraft cabin, and the external flow considered is representative of typical cruise conditions of a commercial aircraft. The analytical model is based on modal analysis, and it considers the structural-acoustic coupling for frequencies up to 1000 Hz. A convergence study was performed to determine the number of structural and acoustic modes required for the calculation of the spectral quantities, indicating that a large number of non-resonant modes need to be considered in the analysis. Also, it was found that hydrodynamic coincidence lines 'match' the plate modes over a large part of the frequency range, confirming the importance of inefficient, but resonant and highly excited modes in the aircraft noise problem.

This study leads to conclude that, for the accurate prediction of aircraft interior noise, the position of the panel as well as the structural-acoustic coupling effects are important factors to consider. Thus, the traditional approach of assuming a single panel vibrating to free air or coupled with an acoustic enclosure needs to consider these two factors. Additionally, the space-averaged PSD values only give information about the mean value. If one desires to determine the localized PSD values (for the plate vibration or for the interior pressure level), then the space-averaged values may not sufficiently accurate, since predicted averaged and localized values are dissimilar. For instance, one might want to predict the interior SPL at the passenger's head height, while in flight.

The analytical model presents also a solid basis for further analyzes, such as multidisciplinary design optimization analysis, and design and implementation of noise reduction techniques, as for instance: the use of added masses in the structure as passive noise control methods; the use of structural actuators embedded in the plates as active structural acoustic control methods; or loudspeakers installed at the interior of the cabin as active control noise methods.

ACKNOWLEDGMENTS

The authors are grateful for the financial support of the Foundation for Science and Technology, through a post-graduate scholarship program.

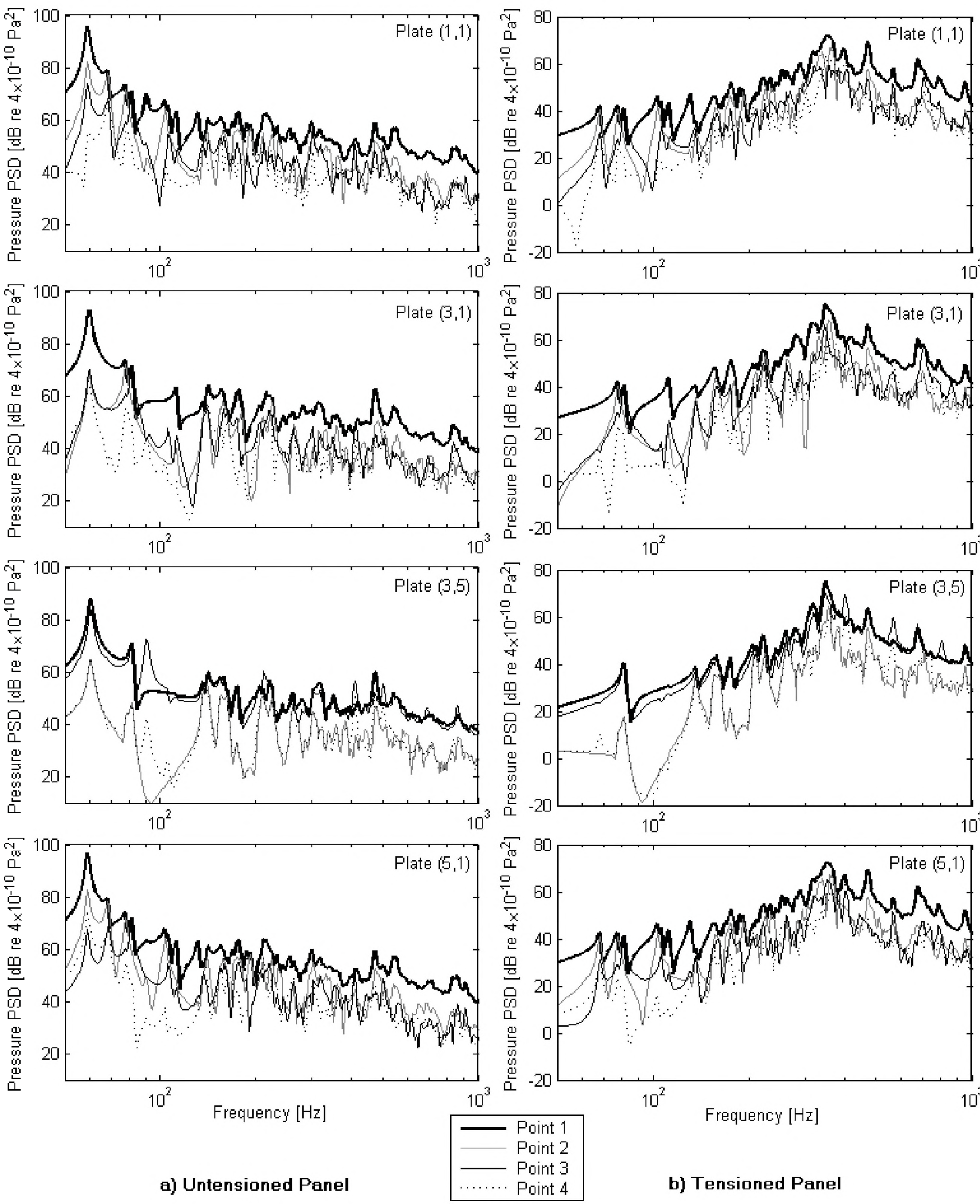


Figure 9. Pressure PSD at 4 points inside the enclosure, for the individual contribution of panels (1,1), (3,1), (3,5), and (5,1).

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SIMULTANEOUS MEASUREMENTS OF AMBIENT SOUND LEVELS AND WIND SPEEDS

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ABSTRACT

Continuous measurements of ambient sound level and wind speed were made for about 17 months at a wind farm site prior to construction, to obtain baseline sound levels. The site is in the middle of an agricultural field on a meteorological tower. Wind was measured at 3, 10, 30, 40, and 50 m above the ground. As expected the diurnal pattern showed high values of wind shear at night, compared to daytime in summer, but little day/night variation in winter. Relating sound level to wind speed indicated that the Ontario MOE approach of increasing the noise criteria with wind speed is appropriate and that above 5m/sec., ambient sound levels exceeded the MOE wind turbine sound limits due to wind action, in the absence of any wind turbines.

SOMMAIRE

Des mesures continues du niveau de sons ambiants et de la vitesse du vent ont été faites au cours de 17 mois à un site de ferme d'aérogénérateurs avant la construction, afin d'obtenir les niveaux de sons de base. Le site est au milieu d'un champ agricole sur un tour météorologique. Le vent a été mesuré à 3, 10, 30, 40 et 50 mètres au dessus de la terre. Comme prévu, la tendance quotidienne a montré des hautes valeurs de décalage de vent la nuit, par rapport aux journées pendant l'été, mais peu de variation jour/nuit pendant l'hiver. Comparant le niveau de son avec la vitesse du vent indique que la méthode du Ministre de l'Environnement de l'Ontario d'augmenter le principe de bruit avec la vitesse du vent est approprié et qu'après plus de 5 mètres par seconde, les niveaux de sons ambiants ont excédés les limites éolienne du Ministre de l'Environnement à cause du mouvement du vent dans l'absence d'aucune éolienne.

1. INTRODUCTION

Continuous measurements of sound level and wind speeds at different heights have been made between May 2007 and October 2008 to provide baseline reference information on ambient sound levels as a function of wind speed on a major wind power project site (wind farm). The current analysis presents information on measured wind and ambient sound data up until the end of October 2008, after which the operation of the wind turbines began. The measurement program is still on-going.

The measurement results were used to examine the validity of the Ontario Ministry of the Environment (MOE) sound limit criteria for wind turbines. These sound limits are based on ambient sound levels that increase with local wind speed at the sensitive receptor locations. The validity of this approach has been questioned due to the possible diurnal reduction in wind speed close to the ground that would result in reduced ambient sound levels, while wind speeds at wind turbine hub height show lesser or no reduction.

2. THE MEASUREMENTS

Sound level was measured at a height of about 3 metres (m) above ground with an integrating sound level meter sampling continuously and set to provide hourly summaries of Leq and cumulative probability (L_n) values. Wind speed was measured at heights of 3 m, 10 m, 30 m, 40 m and 50 m above ground.

3. THE SITE

The area is quite flat and used primarily for agriculture. There is very little road traffic on the nearby roads. The site is in the middle of an agricultural field. Thus, other than grass/weeds at the base of the measurement mast (a round-pipe) and crops during the growing season, there is very little major foliage in the immediate vicinity. There are hedgerows and trees along the border of fields and property lines and in the vicinity of a farm and other sparsely located houses. Figure 1 shows the sound measurement set up and the anemometers at 3 m.

4. WIND PROFILES

4.1 Wind Speeds

The MOE sound level limits for wind turbines are referenced to the wind speed at 10 m height. The IEC standard for measurement and rating of wind turbine sound emission also requires reporting the data referenced to wind speed at 10 m height. This appears to be an arbitrary height to introduce standardization because the average driving effect is wind speed at wind turbine hub height which varies with turbine type and installation. Figure 2 shows the measured hourly

Figure 1. Measurement Set-up.

wind speeds, at 10 m height, on a monthly basis. Figure 3 shows a histogram of wind speed for the whole time period measured.

Generally, wind speed was higher during the day (roughly 0700 to 1800 hours). However, this effect varied by month/season. The effect was greatest during summer (June to September) and least during winter (December and January) when wind speeds were more constant around the clock. Typically, the wind speeds were higher during winter, especially at night.

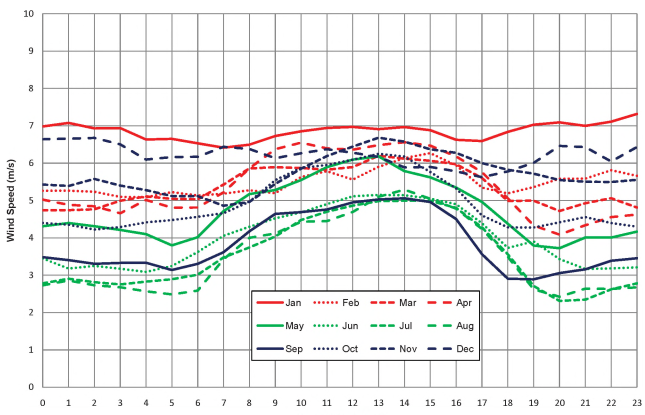

Figure 2. Wind Spped at 10 m Height.

4.2 Wind Shear

Normally, the wind speed increases with height. The equation that is commonly used to relate wind speeds at different heights is:

$$\frac{V_u}{V_l} = \left(\frac{H_u}{H_l}\right)^{\alpha} \qquad 1$$

where V_u is the wind speed at the upper height H_u. V_l is the wind speed at lower height H_l and α is the wind shear exponent (sometimes referred to simply as the wind shear).

This results in a logarithmic wind profile of speed vs. height. The values of α were calculated using the wind speed at all heights by an exponential curve fit, for each hour, on a monthly basis. Figure 4 shows the results. The pattern is believed to be typical of an open flat area in Ontario, although the specific absolute values of α will be site dependent. As might be inferred from Figure 4, the day values of α were low (0.1 ±) all year round. For winter, α remained low around the clock. During summer (June to September), the nighttime values rose to 0.4 to 0.5.

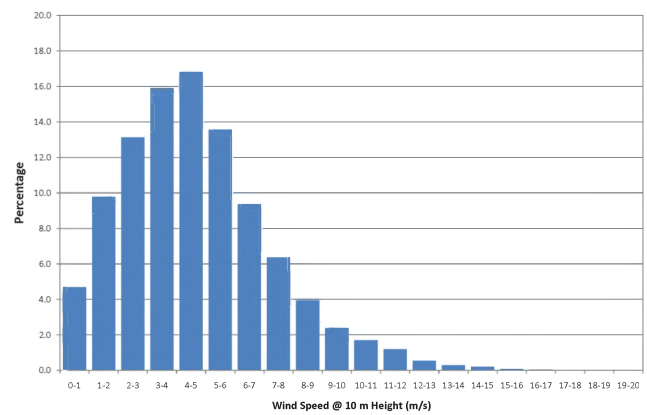

Figure 3. Wind Speed Profile.

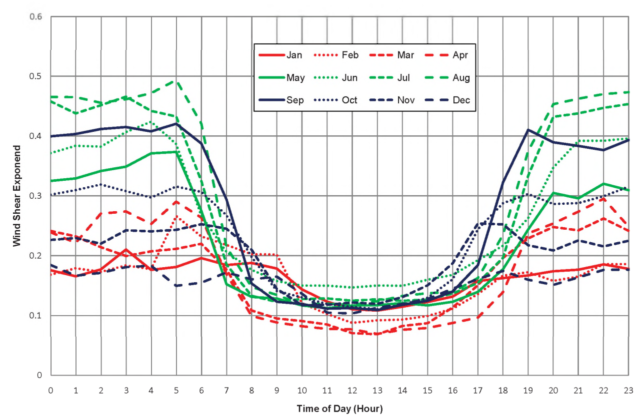

Figure 4. Wind Shear.

5. WIND SPEED AND SOUND LEVEL

5.1 Time History

The relationship between ambient sound level and wind speed can be examined for wind at any height. The patterns remain the same. The wind speed values are a function of height. For direct comparison to the MOE guidelines, the wind speed at 10 m height was used. Figure 5 shows a sample segment of time history, over two weeks, of hourly sound levels in terms of L_{eq} and L_{90} and hourly wind speed averages at 10 m and 3 m heights. The wind speeds at 3 m, which are more representative of what people and objects at ground level would experience, tracks that at 10 m but at lower levels.

The L_{90} values track the L_{eq} values very well. This leads to the conclusion that it is the wind that is the prime determinant of the measured sound levels, in this quiet, rural

Figure 5. Time History of Sound Levels.

environment. In a typical urban environment, with various activities, including traffic on other than expressways, it is common to have elevated values of L_{eq} (peaks) with more steady values of L_{90}. This is because many high sound level (noisy) events (e.g., vehicle pass-bys) that elevate L_{eq} do not last long enough (i.e., at least 90% of the time) to affect L_{90}. Subjectively, as illustrated in Figure 5, the sound level values and wind speeds also tracked very well. The area is very quiet, with minimum sound levels as low as 20 dBA, when wind speeds were negligible.

5.2 Ambient or Artefact

One of the concerns with sound measurements of this type is to be sure that the observed sound levels are, in fact, true ambient and not artefacts resulting from air flow over the microphone windscreen or the microphone itself. The sound level meter manufacturers do not provide data about the minimum sound levels that can be measured with their windscreens in the presence of air movement. It is known that the bigger the windscreen, the lower the potential for spurious readings. Hessler (2008) studied the sound levels generated by different air speeds flowing over a variety of windscreens in a specially built "quiet" wind tunnel [1]. Figure 6 shows the data and curve fit from the Hessler study for a windscreen similar to that was used in the current study. The comparison of the current sound level data to this curve showed that the measured sound levels are ambient and not artefacts.

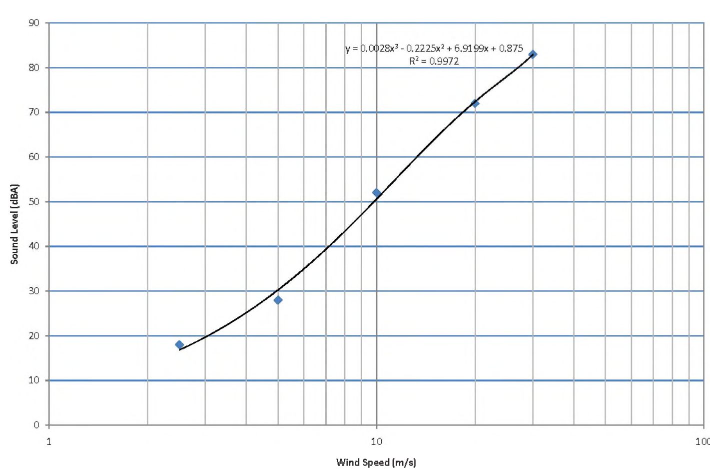

Figure 6. Wind Screen Noise Levels.

5.3 Results

Figure 7 shows a plot of all hourly sound data points (some 13,000 plus data points) as a function of the corresponding wind speed, at 3 m, the same height as the microphone. Particularly above 5 m/sec there is a definite trend pattern of increasing sound level with wind speed. At lower wind speeds there is more scatter and variation because wind generated sound levels are lower and other sources would be expected to be more dominant. Also shown on Figure 7 is the curve of the Hessler, laboratory-determined sound levels attributable to the air flow over the windscreen. In general, the measured data is well above this curve. Some measured sound levels were less than 10 dBA above the "windscreen line". Thus, to be rigorous, all data points were corrected for the sound

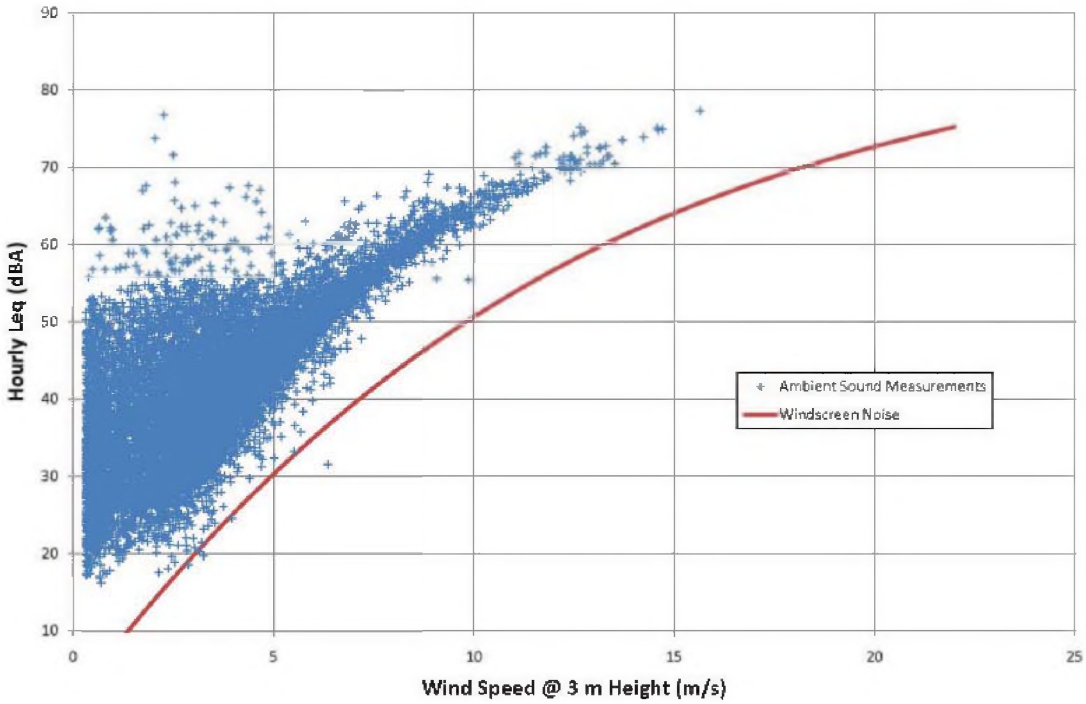

Figure 7. Hourly Noise Levels at 3 m.

level attributable to the windscreen.

Figure 8 shows all of the hourly sound level data points plotted against the wind speed at 10 m height, with a polynomial curve fit to the data. Figure 9 shows this data curve as well as the Ontario MOE wind turbine sound limit curve. (Recall the MOE criteria are referenced to wind speed at 10 m height.) Above 5 m/sec wind speed the ambient sound levels exceed the MOE criteria. Below 5 m/sec, the ambient sound levels were lower than the MOE criterion, which remains constant at 40 dBA at and below 6 m/sec wind speed. This approach is consistent with the MOE stationary source exclusion sound limit of 40 dBA in quiet rural areas, where a source is not required to attenuate below 40 dBA, regardless of the ambient sound level.

5.4 Analysis Intervals

Because the MOE noise guidelines are based on hourly time periods, the ambient sound levels were measured as one hour L_{eq} and related to hourly averages of wind speed. In addition

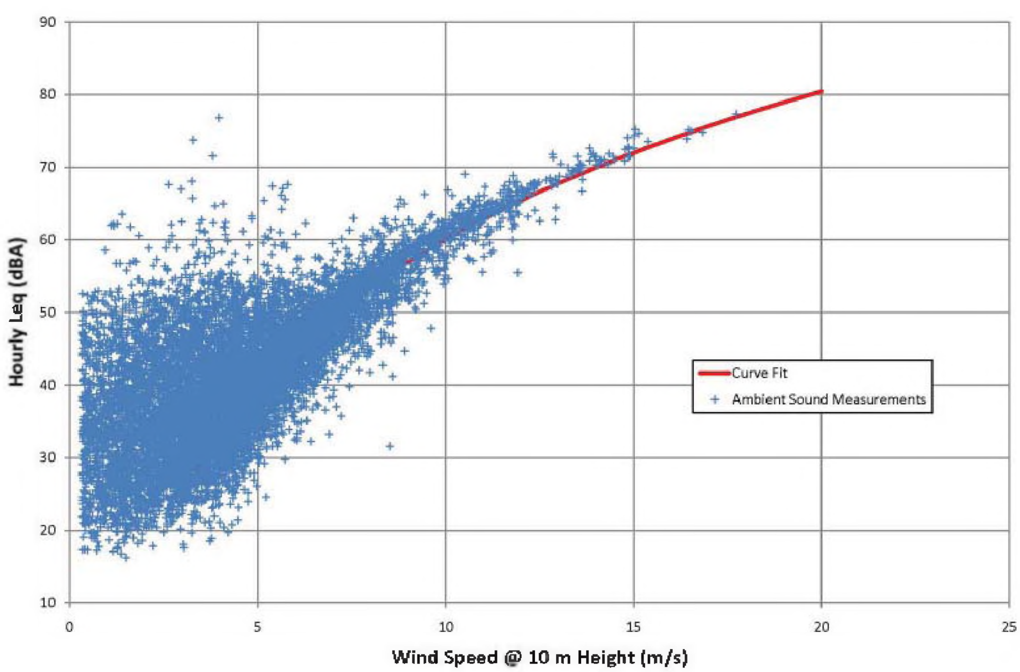

Figure 8. Hourly Noise Levels and Wund Speed at 10 m.

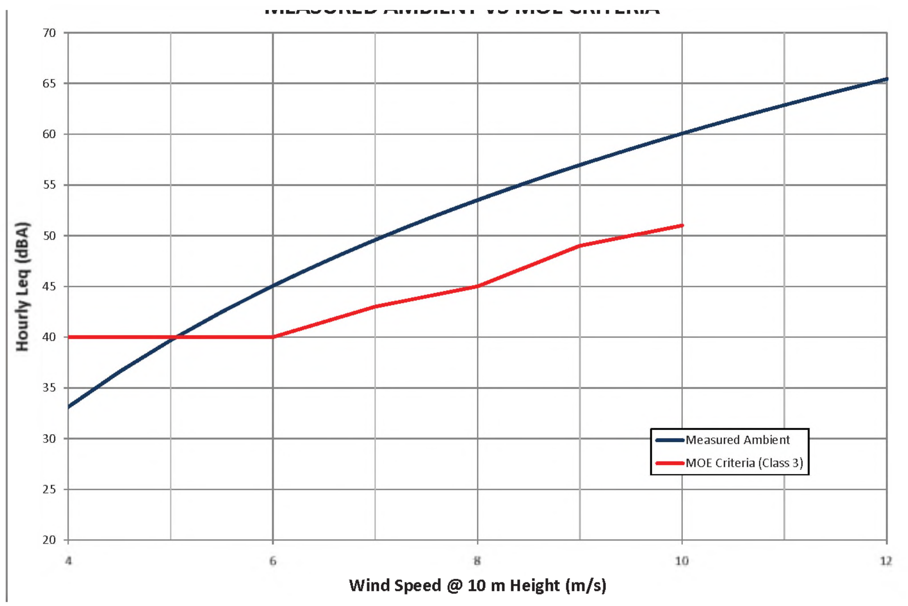

Figure 9. Hourly Noise Levels and MOE Noise Criteria.

to hourly L_{eq}, various hourly L_n sound level parameters were also recorded. The wind speed data was actually obtained as 10 minute averages; that is, as six samples per hour that were averaged together. The variability of the data within the hourly periods was examined. It may be surmised that during gusty conditions, the wind speed may vary significantly over short periods (within the hour). Correspondingly, the ambient sound levels may fluctuate significantly over the same time.

Figure 10 shows a plot of L_5 vs L_{eq}. A linear relationship fits well; basically L_5 is L_{eq} + 3.8 dBA, with very little scatter. For any hour, the difference between L_5 and L_{99} is the range of sound levels that existed for most (94%) of the time. Figure 11 plots the range of sound level vs. hourly L_{eq}. For any given hour there was a wide range of instantaneous sound levels contributing to the hourly Leq value.

Figure 12 plots the Standard Error (SE) and the Standard Deviation (SD) of the 10 minute wind speed values, binned to integer values for each hour. The SE is close to zero and the SD is small. That is, the variation in wind speed in any hour was small. However, the corresponding range of sound levels is relatively large (about 15 dBA). This apparent discrepancy may be due to significant wind speed variations that are averaged out using the 10 minute averaging periods.

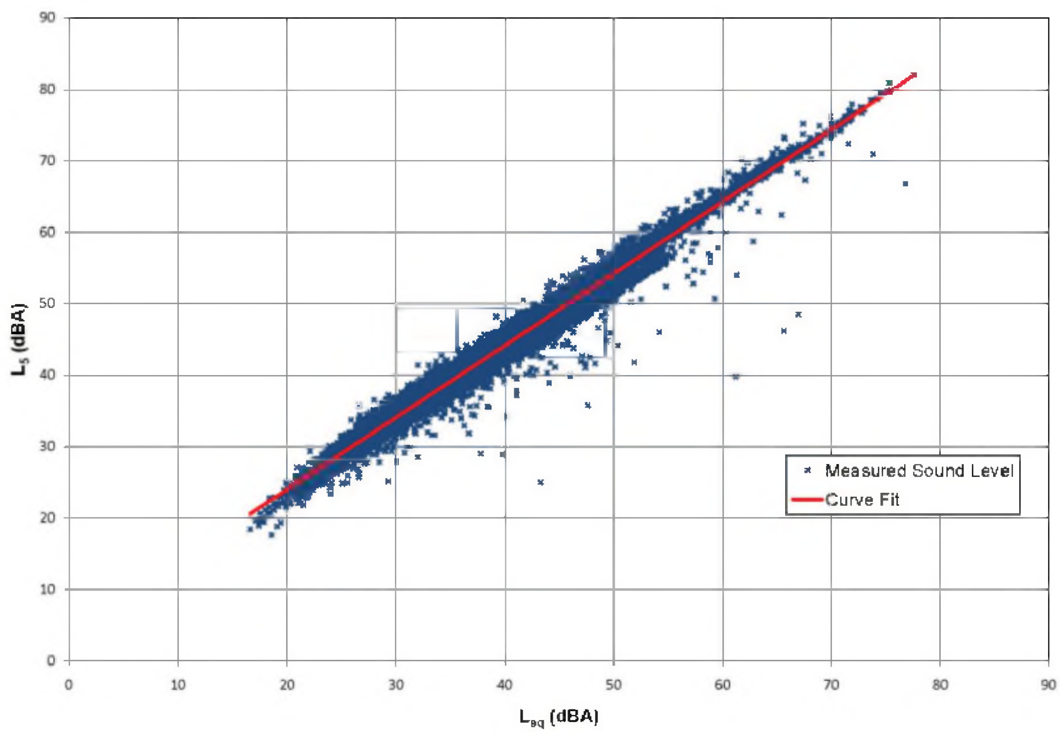

Figure 10. Hourly Noise Levels L_5 and L_{Eq}

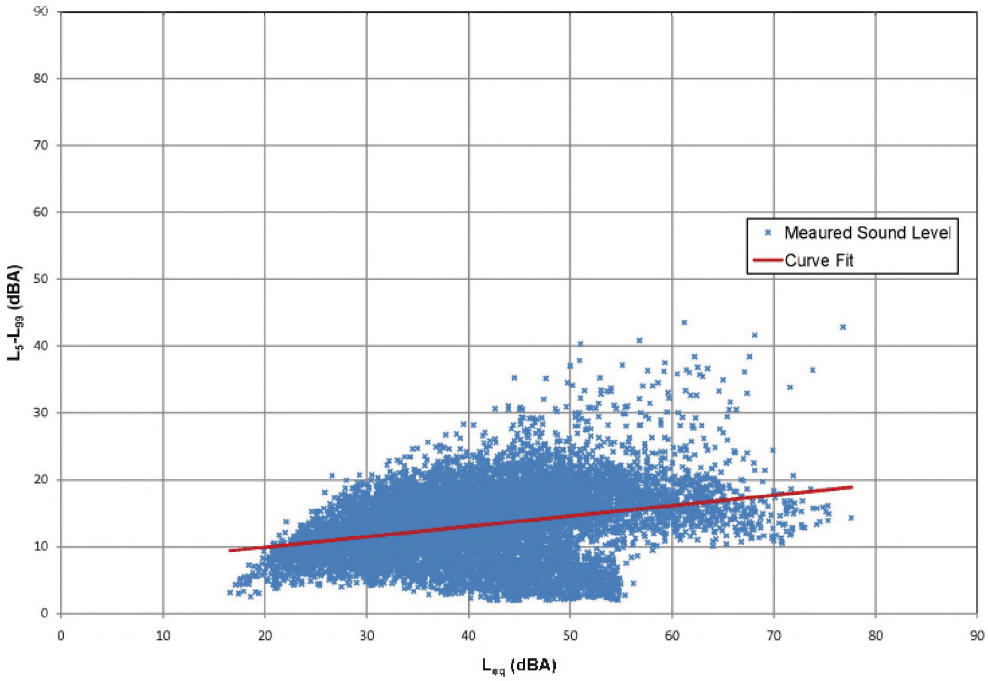

Figure 11. Hourly Noise Levels Range and L_{EQ}

6. CONCLUSIONS

1. Care must be taken in the selection of microphone windscreens, to measure low ambient sound levels in the presence of, or due to wind. There is the potential for air flow over the microphone/ windscreen assembly (or in fact over or past other objects close to the microphone) to produce spurious sound level readings. Of course, the resulting sound levels due to turbulent flow over objects such as residential buildings or trees, etc., that are part of a receptor's environment are legitimately part of the ambient environment.
2. As expected, wind speeds were generally higher in winter than in summer, with spring and fall being intermediate.
3. The expected diurnal variation in wind shear exponent was observed. This effect was strongest in summer, with wind shear exponent variation of 0.4 or 0.5 to 0.1, between night and day, and negligible in winter, with very little diurnal variation. The other seasons exhibited intermediate effects.
4. Above 5 m/sec wind speed, the ambient sound levels attributable to wind at a flat, open, agricultural site, were above the Ontario MOE sound limits for wind turbines. At and above 6 m/sec, the increment was at least 5 dBA, increasing with wind speed.
5. At and below 5 m/sec wind speed, the ambient sound levels were below 40 dBA, the applicable MOE criterion limit. The 40 dBA criterion is consistent with the "exclusion limit" used by the MOE noise guidelines for other types of stationary sources in quiet areas where the

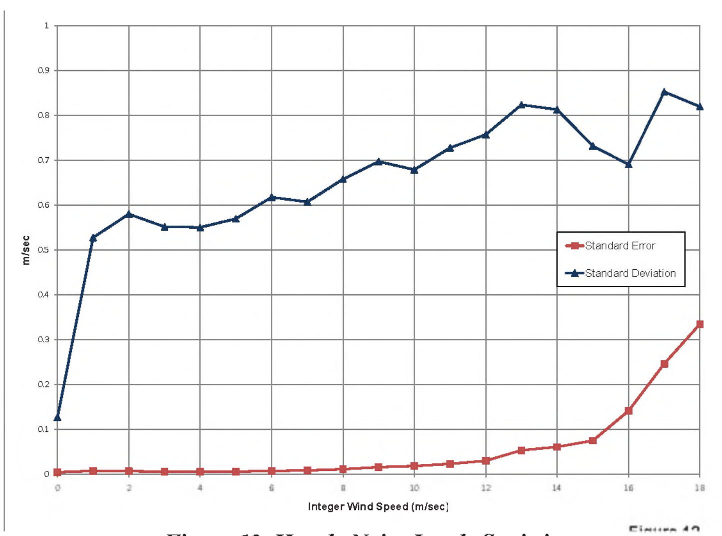

Figure 12. Hourly Noise Levels Statistic.

ambient can be expected to be lower.

6. It is concluded that, at least for a flat, quiet, rural, agricultural environment in Ontario, the MOE sound level limits for wind turbines are appropriate and are consistent with the notion that the sound limits should increase with wind speed above 6 m/sec, due to increasing ambient sound.
7. For measuring ambient sound in a quiet area, hourly averages of wind and sound (energy) data are acceptable. During gusty wind conditions it would be expected that ambient sound levels would follow in step with changes in wind speed and be appropriately reflected in the averages. However, large commercial wind turbines would not be expected to respond to rapid wind speed changes; in effect, averaging them out. Thus, significant fluctuations in sound level may be observed due to the ambient. To do a valid sound audit of a wind farm, and properly account for ambient sound levels, it appears that relatively short sampling periods for both sound level and wind speed are needed; possibly one minute or less, so that measured sound levels and wind speeds can be correlated. Further research is required to determine an appropriate data sampling rate.

ACKNOWLEDGEMENTS

We appreciate the cooperation of Mr. Bob Simpson and Enbridge Ontario Wind Power LP in allowing this data to be made public.

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SOUND ABSORPTION IMPROVEMENT FOR CEMENTITIOUS MATERIALS

H. S. Seddeq
Housing & Building Research Center, Building Physics Department, Egypt, hudasoliman@yahoo.com

ABSTRACT

Acoustical material plays a number of roles that are important in acoustic engineering such as the control of room acoustics, industrial noise control and studio acoustics. Sound absorptive materials are generally used to counteract the undesirable effects of sound reflection by hard, rigid and interior surfaces and thus help to reduce the reverberant noise levels. Cementitious materials may be used as interior finishing for interior surfaces in buildings. This paper review of sound absorption studies for cementitious materials for their potential benefits in sound absorption and investigate some finishing cementitious materials added with porous and fibrous materials to improve the sound absorption performance. Sprayed cement mortar containing cotton fibers and perlite of an amount of perlite in the region of 80% in relation to the cotton fibers gave the best results. Also sprayed fibrous cement mortar based on mixture of mineral wool and cement binders achieved high sound absorption.

RÉSUMÉ

Matériau acoustique joue un certain nombre de rôles qui sont importants dans l'ingénierie acoustique tels que le contrôle de l'acoustique des salles, contrôle du bruit industriel et acoustique de studio. Absorbants phoniques sont généralement utilisés pour contrer les effets indésirables de la réflexion du son par les surfaces dures, rigides et de l'intérieur et ainsi contribuer à réduire les niveaux de bruit de réverbération. matériaux à base de ciment peut être utilisé comme finition intérieure pour les surfaces intérieures des bâtiments. Cette revue papier des études d'absorption acoustique de matériaux cimentiers pour leurs avantages potentiels de l'absorption acoustique et d'enquêter sur certains matériaux de finition à base de ciment a ajouté avec des matériaux poreux et fibreux pour améliorer les performances d'absorption acoustique. mortier de ciment projeté contenant des fibres de coton et de perlite d'un montant de perlite dans la région de 80% par rapport aux fibres de coton a donné les meilleurs résultats. Également pulvérisé mortier de ciment fibreux à base de mélange de laine minérale et de liants de ciment atteint haute absorption acoustique.

1. INTRODUCTION

Sound absorptive materials are generally used to counteract the undesirable effects of sound reflection by hard, rigid and interior surfaces and thus help to reduce the reverberant noise levels [1], [2]. In order to have good sound absorption performance, the material should be in porous or fibrous form; the energy is lost by viscous dissipation when sound waves propagate into the material [3]. Sound absorbing materials have been developed as an engineering control to reduce reverberation and overall sound levels [4].

2. MEASUREMENT OF SOUND ABSORPTION COEFFICIENT

The performance of sound absorbing materials in particularly is evaluated by the sound absorption coefficient (α) [5], [6]. Alpha (α) is defined as the measure of the acoustical energy absorbed by the material upon incidence and is usually expressed as a decimal varying between 0 and 1.0. Values are usually provided in the literature at the standard frequencies of 125, 250, 500, 1000 and 2000 Hertz [5], [7].

In comparing sound absorbing materials for noise control purposes, the noise reduction coefficient (NRC) is commonly used. NRC is the average usually stated to the nearest multiple of 0.05, of the coefficient at four frequencies 250, 500, 1000 and 2000 Hz [8]. It is intended for use as a single number index of the sound absorbing efficiency of a material. The NRC value provides a decent and simple quantification of how well the particular surface will absorb the human voice [9].

Harris [8] describes the four factors that affect the sound absorption coefficient:

- Nature of the material itself
- Frequency of the sound
- The angle at which the sound wave strikes the surface of the material
- Air gap

Measurement techniques used to characterize the sound absorptive properties of a material are [10], [11]:

- Reverberation room method [ASTM C423]
- Impedance tube methods [ASTM E 1050]

Impedance tube method uses plane sound waves that strike the material straight and so the sound absorption coefficient is called normal incidence sound absorption coefficient, NAC [7]. Impedance tube method is faster and generally reproducible and, in particular, requires relatively small circular samples, either 100 or 29 mm in diameter for low and high frequency measurements. In the impedance tube method, sound waves are confined within the tube and thus the size of the sample required for test needs only be large enough to fill the cross section of the tube. Thus this method avoids the need to fabricate large test sample with lateral dimensions several times the acoustical wavelength.

Two fixed microphone impedance tube or transfer function method (ASTM E 1050), which is relatively recent development can be used. In this technique, a broadband random signal is used as a sound source. The normal incidence absorption coefficients and the impedance ratios of the test materials can be measured.

For this investigation, PULSE acoustic material testing tube, type 4206 (B&K) and impedance Tube Kit (50Hz - 6.4 kHz) were used for sound absorption measurements in conjunction with the software 7758 for determining the sound absorption coefficients for the tested samples [12].

Reverberant Method for measuring sound absorption is concerned with the performance of a material exposed to a randomly incident sound wave, which technically occurs when the material is in diffusive field. However creation of a diffusive sound field requires a large and costly reverberation room. Since sound is allowed to strike the material from all directions, the absorption coefficient determined is called random incidence sound absorption coefficient. This method is clearly explained in [ASTM C423]. The measurements of reverberation time in room under consideration were carried out in the reverberation room without and with the sample according to [ASTM C423]. Where the tested sample was applied to a substrate and tested according to the mounting methods stated in the standard in ASTM E795.

The analyzer of B&K's portable PULSE connected with condenser microphone type 4189, omni directional loudspeaker type 4292 (B&K), power amplifier 2716 (B&K) and the soft ware type 7842 have been used for measuring the reverberation time. Where the noise signal generated from the pulse generator that excites the reverberation room with and without the sample. The reverberation time for decay 60 dB is determined with and without the tested sample then the sound absorption coefficients is calculated using these measurements.

3. LITERATURE REVIEW

N. Neithalath, J. Weiss, and J. Olek evaluated three classes of specialty cementitious materials for their potential benefits in sound absorption including a Foamed Cellular Concrete (FCC), Enhanced Porosity Concrete (EPC) incorporating 20-25% open porosity, and a Cellulose Cement Composite (CCC). The FCC specimens showed absorption coefficients ranging from 0.20 to 0.30, the higher value for lower density specimens. The closed disconnected pore network of FCC resulted in a reduced absorption, even though the porosity is relatively high. The most beneficial acoustic absorption was observed for EPC mixtures. By engineering the pore structure by careful aggregate grading as in EPC, or incorporating porous inclusions like morphologically altered cellulose fibers, cementitious materials [13].

L. Arnaud and V. Cerezo study the acoustical properties of various formulations of concrete containing vegetable particles. Such material is made up with hemp shives mixed with lime binders. Thus, this concrete presents a high porosity related to the microscopic porosity of the shives and the macroscopic porosity due to the arrangement of particles resulting in sound absorption between 0.5 and 1 [14].

Knapen E., Lanoye R., Vermeir G., Lauriks W., Van Gemert D showed that polymer-modified porous cement mortars can be an alternative for the more conventional sound absorbing materials. They linked the acoustic behaviour to the polymer/cement ratio, the sand/cement ratio, compaction and size of the sand. These were connected to pertinent physical parameters (porosity, flow resistance, tortuosity, etc.) and those parameters were in turn linked to measured sound absorption [15].

Piti Sukontasukkul investigated the sound properties of crumb rubber concrete panel. The crumb rubber was used to replace fine aggregate at ratios of 10%, 20% and 30%. Results indicated that sound absorption coefficients α-values is low at the low frequency ranges of 125 and 250 Hz, However, at the mid-frequency (500 Hz), the crumb rubber concrete began to show slightly higher α-values. The ability to absorb sound by all crumb rubber concrete lightweight concrete was found to be much better than that of plain concrete for frequencies greater than 1000 Hz. This indicated that crumb rubber concrete is a better sound absorber at the high-frequency range than plain concrete [16].

4. EXPERIMENTAL WORK

The following cementious materials were investigated in this study:

1- Conventional cementsand mortar
2- Cementitious mortar altered with cotton fiber and expanded mineral (perlite)
3- Cementitious mortar altered with mineral fiber

The measurements have been carried out according to ASTM C423 ASTM E 1050 standards.

4.1 Conventional cements and mortar

The materials used in this investigation are ordinary Portland cement (OPC) and ultra fine sand (UFS). The UFS was added to the OPC at different weight ratios from 0 to 10% and the best sound absorption was as shown in table (1) and figure (1). The measurements have been made using pulse acoustic material testing tube type 4206 (B&K). The results of measurements show that the conventional cementsand mortar have low sound absorption coefficient at all frequency range from 100 to 6300 as shown in figure (1).

Frequency, Hz	α of Mortar with cotton
125	0.03
250	0.04
500	0.06
1000	0.07
2000	0.05
4000	0.07
NRC	0.06

Table 1: Sound absorption coefficient of the tested ordinary cement mortar

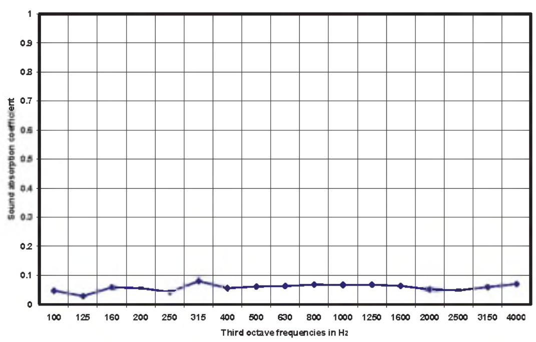

Figure 1 Sound absorption coefficient of ordinary cement mortar.

4.2 Cementitious material altered with cotton fiber and perlite

This investigation relates to a finishing mortar for sound-absorbing coating of inner walls, ceilings and the like in buildings. It may be applied directly on concrete or some other carrying material or on underlying insulation material, such as mineral wool. The finishing mortar according to the investigation is characterized in that it comprises cotton fibers and expanded mineral, such as perlite. Cement finishing mortar consisting essentially of: perlite; and cotton fibres, wherein a weight ratio of perlite to cotton fibres is in the range of 10%-250% preferably 80%. The mortar was present as a water dispersion with a content of solids which makes it suitable for spraying, the content of solids then preferably being 200-300 g/l. A dry volume weight of perlite was in the range of 35 to 125 kg/m3 dependent on particle size

The mortar sprayed on to a metal plate of length 3 and width 3.5. The metal substrate fixed to the floor of the reverberation room. The perimeter edges sealed with acoustic sealant. The thickness of the tested mortar was 10 mm. The measurements have been carried out in acoustics laboratory of housing and building research center according to ASTM C423.

Figure (2) shows the sound absorption coefficient for the tested mortar containing cotton fibers and perlite of different weight ratio. Where the highest sound absorption achieved for mortar containing cotton fibers and perlite of weight ratio 80%

Also the sound absorption coefficient for a tested mortar with cotton fibers only have been measured where the cotton fiber is applied by spraying on in two steps with intermediate drying. The cotton fiber mixture which is sprayed consists of cotton having suitable grinding degree, water, mica, biolite, muscovite, and silicaber to obtain different effect. The thickness of cotton layer was 4 mm.

Figure (3) shows the sound absorption coefficient of the tested mortar with cotton fibers only. The sound absorption effect of the tested mortar containing cotton fibers and perlite according to this research is shown in Table 2 and also compared to finishing mortar containing only cotton fibers.

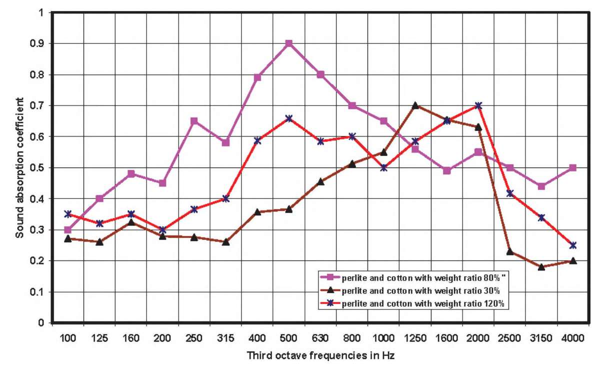

Figure 2. Sound absorption coefficient of the tested mortar with different containing perlite and cotton with weight ratio

Frequency, Hz	α of Mortar with cotton	α of Mortar with cotton fibres and perlite
125	0.40	0.40
250	0.55	0.65
500	0.75	0.90
1000	0.50	0.65
2000	0.25	0.55
4000	0.20	0.50
NRC	0.5	0.7

Table 2: Sound absorption coefficient of the tested mortar containing cotton fibers and perlite

From the results of measurements it is clear that the sound absorption of mortar altered with cotton fiber and perlite achieved good sound absorption due to the increase of porosity. Also the sound absorption of mortar altered with expanded material (perlite) are better than mortar with cotton because during admixing of the expanded material the air penetration of the mortar after drying will be maintained, which means that improve the acoustic properties

4.3 Cementitious material altered with mineral fiber

Two sprayed cement mortar have been tested in the reverberation room according to ASTM C423 as follows:

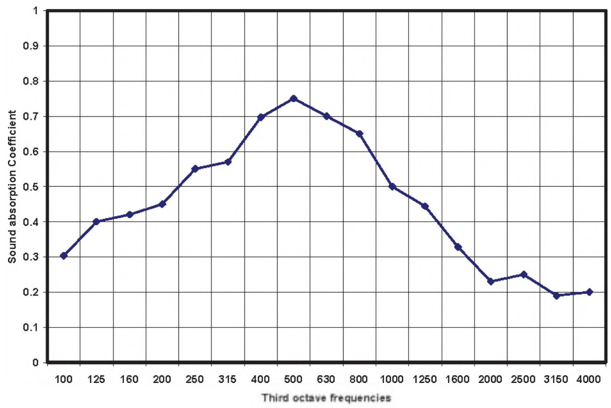

Figure 3 Sound absorption coefficient of the tested mortar with cotton fibers only

1- Sprayed cement material type 1

The sound absorption coefficient of fibrous cement material has been measured. This sample is a sprayable blend used for acoustic correction and made of mineral wool and hydraulic and inorganic binders of 25 mm thickness. The mortar sprayed on to a metal substrate of length 3 m and width 3.5 m. The metal plate fixed to the floor of the reverberation room. The perimeter edges sealed with acoustic sealant. The measurements are carried out in the reverberation room of acoustics laboratory in housing and building research center where the tested sample is sprayed on metal panel on the floor of the reverberation room. Figure (4) and Table 3 shows the sound absorption at third octave frequencies from 100 to 4000.

2- Sprayed cement materials type

The sound absorption coefficient of fibrous sprayed material is based on mixture of mineral wool and cement binders. This sample has been sprayed of thickness 10 mm on metal plate of area 10.5 m^2 on the floor of the reverberation room. Figure (5) and Table 4 shows the sound absorption at third octave frequencies from 100 to 4000.

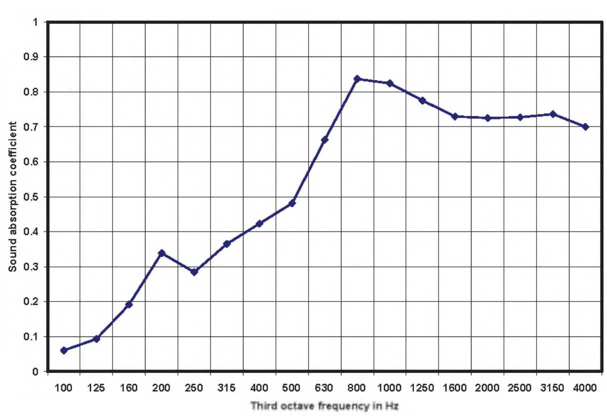

Figure 4 Sound absorption coefficient of sprayed fibrous cement sample type 1 of 25mm thickness

Figure (4), (5) show the sound absorption coefficient of sprayed fibrous cement samples where the sound absorption coefficient achieved good sound absorption. But the sound absorption coefficient of the sprayed fibrous cement mortar type 1 is better than sprayed fibrous cement mortar type 2 at the low frequencies due to the increase of thickness.

5. CONCLUSION

This paper investigated the sound absorption of some cementitious materials that may be used as interior finishing for interior surfaces in buildings improve the indoor acoustic performance. The experimental work indicated that the sound absorption of sprayed cement mortar can be improved by adding cotton fibers and perlite. An amount of perlite in the region of 80% in relation to the cotton fibers gives the best results. Also sprayed fibrous cement material based on mixture of mineral wool and cement binders can achieve high sound absorption that can be used as interior finishing specially for coating high ceiling

Frequency, Hz	α of Mortar with cotton
125	0.1
250	0.28
500	0.48
1000	0.82
2000	0.73
4000	0.7
NRC	0.55

Table 3: Sound absorption coefficient of sprayed fibrous cement sample type 1 of 25mm thickness

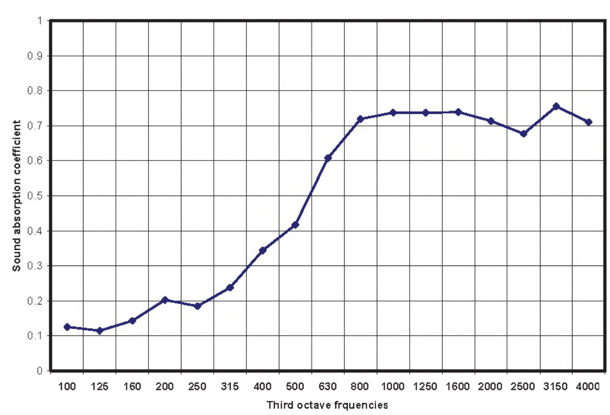

Figure 5 Sound absorption coefficient of sprayed fibrous cement sample type 2 of 10 mm thickness

Frequency, Hz	α of Mortar with cotton
125	0.1
250	0.18
500	0.42
1000	0.74
2000	0.71
4000	0.7
NRC	0.5

Table 4: Sound absorption coefficient of sprayed fibrous cement sample type 2 of 10mm thickness

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The **18th International Congress on Sound and Vibration (ICSV18) will be held from 10 - 14 July 2011 at the five-star Windsor Barra Hotel in Rio de Janeiro, Brazil. Rio de Janeiro** is an international, cosmopolitan metropolis known for its scenic coastlines graced by the instantly recognizable sugarloaf Mountain. The Brazillian culture is vivid, expressive, and welcoming to visitors who travel business or leisure. **The ICSV18 Scientific Programme will include the following keynote lectures:**

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The Canadian Acoustical Association
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Joint Standards Meeting

CSA Occupational Noise Technical Committee & CAA Acoustical Standards Committee

Laurel Point Inn, Victoria BC, 5:00-10:00 pm October 13, 2010

Minutes of Meeting

Tim Kelsall
Hatch, 2800 Speakman Drive
Mississauga, Ontario L5K 2R7, tkelsall@hatch.ca

This was the second joint meeting of the CSA Technical Committee on Occupational Hearing Conservation S304 and the CAA Acoustical Standards Committee. The first was held in the spring at CSA headquarters.

This meeting continues a tradition since the forming of the CAA of having standards meetings as part of Acoustics Week in Canada. The CSA meeting was an informal review, rather than a formal meeting, although it is hoped that at future joint meetings both committees will hold full meetings.

Now that the CAA has a Standards Committee the intent is to publish the minutes in Canadian Acoustics. While the reading may be dry, many members will likely find information on current activities within their specialties around Canada and around the world.

Present:

Tim Kelsall	Hatch	tkelsall@hatch.ca	CSA (vice chair)	CAA (chair)
Tony Brammer	Enviro-Health Solutions	Anthonybrammer@hotmail.com	CSA	CAA
David Quirt	Chair	dave.quirt@nrc.gc.ca	CSA	CAA
Rob Joswlak	Aercoustics	robj@aercoustics.com		CAA
Werner Richarz	Aercoustics	Werner@aercoustics.com		CAA
Christian Giguère	University of Ottawa	cgiguere@uottawa.ca	CSA	CAA
Lixue Wu	National Research Council	Lixue.wu@nrc.ca	CSA	CAA
Brian Howe	HGC Engineering	bhowe@hgcengineering.com	CSA	CAA
Sasha Brown	Worksafe BC	Sasha.Brown@worksafebc.com	CSA	
Stephen Keith (part time by phone)	Health Canada	Stephen_Keith@hc-sc.gc.ca	CSA	CAA
Stephen Bly (part time by phone)	Health Canada	S_Bly@hc-sc.gc.ca	CSA	CAA

Sponsorship: Hatch Associates gratefully contributed toward the cost of holding the joint meeting.

1. CSA Technical Committee On Occupational Hearing Conservation S304 (informal review meeting)

1.1 Update since last meeting

CSA Z107 has been disbanded, with the Occupational Hearing Loss Technical Committee taking over several standards and CAA taking over Z107.10.

1.2 SC 1 (S304.3) – Hearing Protection - Van Volsen

The chair is leaving – need new chair – several names were discussed. Subsequent to the meeting Alberto Behar has proposed that he step down as Committee Chair and take on chairmanship of the Hearing Protection Subcommittee. This proposal is now being taken up by CSA.

EPA 40 CFR Part 211 Product Noise Labelling Hearing Protection Devices; Proposed Rule was briefly discussed

1.3 SC 2 (S304.4) – Noise Exposure Assessment and Control - Tim Kelsall

New appendix for Z107.56 covers assessment of noise exposure for workers using headsets. Tim Kelsall put forward a draft which incorporated assessment of drivers in cabs using radios (because the signal to noise estimate used under headsets would also apply in this case). It was agreed that it was better to put the assessment of such workers in the main body of the standard and limit the appendix to headsets only. Tim Kelsall agreed to do this and circulate a final version to the subcommittee.

C/M G. (Joe) Principato, Assistant Project Manager, “K” Division Radio Renewal, RCMP/GRC has subsequently been asking how soon this appendix can be passed and the standard updated because of the need for the standard.

1.4 Z107.58 – Stephen Bly

Health Canada - Consumer and Clinical Radiation Protection Bureau (Acoustics) in August 2010 posted an announcement on the Health Canada website as follows:

Health Canada - Notice to Stakeholders

Subject: Noise from Machinery Intended for the Workplace

The purpose of this Notice is to further strengthen ongoing efforts to help reduce the number of workers per year who suffer hearing impairments, such as permanent hearing loss, resulting from exposure to occupational noise.

Across Canada, approximately 9,000 workers each year suffer from some form of hearing impairment, including tinnitus (ringing in the ears), due to an overexposure to occupational noise. Excessive occupational noise has additionally been shown to increase the risk of accidents within the workplace, when workers fail to hear warning sounds.

Health Canada recommends that machinery, intended for the workplace be sold, leased or imported into Canada, with accompanying standardized noise emission declarations in both the technical sales literature and the instructions for use.

The Canadian Standards Association's (CSA) Standard *Z107.58 Noise Emission Declarations for Machinery* is the National Standard for Canada. It provides manufacturers with a means to determine and to create noise emissions declarations for the machinery they produce. Noise emission declarations for machinery help to support noise reduction guidance provided by provincial authorities. These declarations enable purchasers to select machines that are compliant with their noise-level requirements, and affords them the opportunity to reduce the level of noise within their workplaces by helping them to purchase quieter machinery and plan noise controls.

The information contained in the CSA Standard *Z107.58* is intended to be consistent with: (i) the European Union (EU) Machinery Directive; (ii) the EU Directive 2003/10/EC on workplace noise; and (iii) numerous international standards supporting these EU Directives.

Comments pertaining to this notice should be directed to: ccrpb-pcrpcc@hc-sc.gc.ca

Stephen Bly asked the committee generally how they could publicize this announcement. There was a general discussion and several suggestions were made, including the need to involve the provincial ministries of labour, workers compensation boards and safety associations, the CCOHS, etc. There was also the suggestion that 1 day course be provided across the country either in person or electronically.

1.5 SC 3 (S304.5) – Hearing Surveillance (Audiometry) - Christian Giguère

Reaffirmation of CSA Standard Z107.6 - Pure Tone Air Conduction Threshold Audiometry For Hearing Conservation was voted before Z107 disbanded.

A list of ANSI/ISO/IEC/OSHA standards overlapping with

Z107.4 and Z107.6 was posted on the CSA website with the help of Dave Shanahan.

The relevance of CSA Standard Z107.4 – Pure Tone Air Conduction Audiometers for Hearing Conservation and for Screening is currently being analyzed against ANSI S3.6-2004 and ISO 60645-1, the main US and international standards on the technical specifications of audiometers. The main observations are:

(1) The mandatory paragraphs of Z107.4 seem fully redundant with ANSI S3.6-2004 and ISO 60645-1 or earlier versions of these standards.
(2) Two supplementary appendices within Z107.4 (A.2 on technician training and B on maintenance and calibration) are valuable and not usually found in audiometer standards like ANSI S3.6 or ISO 60645-1. However, the two appendices are mandatory (paragraphs 7.0 and 4.2) in other audiometric standard Z107.6. Therefore, the appendices are redundant.
(3) Updating Z107.4 will at best bring us to the level of specifications already contained in ANSI S3.6 or ISO 60645-1.
(4) It seems preferable to work at the level of ANSI or IEC if we feel a need for a change in the technical characteristics of audiometers. Our market may be too small to justify a Canadian-based standard for audiometer manufacturers to take notice.

Abandoning Z107.4 is therefore an option the subcommittee is currently considering. A recommendation to the Main CSA Committee on Hearing Conservation may follow before the next meeting in May 2011.

1.6 SC 4 (S304.6) – Vibration Exposure Assessment and Control - Tony Brammer

There was a discussion leading to agreement that if possible there be a section in the hearing conservation standard discussing occupational vibration exposure.
The subcommittee considers Whole-Body Vibration Exposure, and operates in parallel with the CAA subcommittee on Human Vibration, which is harmonized with the Canadian Advisory Committee on ISO/TC 108/SC 4 "Human Exposure to Mechanical Vibration and Shock". The subcommittee continues to direct its efforts in support of the development of international standards. In this role, members of the subcommittee serve as conveners of two of the Working Groups (WG5 - Biodynamic Modeling, and WG8 - Vibrotactile Perception).

ISO/TC 108/SC 4 last met in London in September 2010 (see section 3.7 for the most significant developments at the London meeting).

A list of international standards prepared by ISO/TC 108/SC 4 is appended to the CAA report (see section 3.7).

1.7 SC 5 (S304.7) – HC Management - Jeffrey Goldberg (not attending)

The CAALL-OSH committee - representing OHS regulators from all jurisdictions across Canada - have decided to fund CSA Z1007 – Hearing Conservation Management Standard. They will provide the funds necessary to cover the costs of development, French translation, and publication. This funding demonstrates the support of the regulatory authorities for some of the objectives of the new Technical Committee.
It was agreed that among other things this standard should either amalgamate or at least point to all the standards, Canadian and International, which are encompassed by the Occupational Hearing Conservation Technical Committee.

2. CAA Standards Committee Meeting

2.1 Items from CAA board

Next year's meeting will be in Quebec City in October 2011. The ICA will be held in Montreal, June 3-7, 2013.

The Standards Committee minutes will be published in Canadian Acoustics.

The committee chair will be invited to CAA board meetings (when Tim Kelsall ceases to be a director) to report each year.

Handling of Z107.10 and other potential standards: It was agreed to re-label the standard as CSA S100. It was also agreed that charging for the standard might prove counter-productive. Instead we will look for sponsors.

2.2 Environmental Noise (B.H. for Bill Gastmeier) and Wind turbines – Brian Howe

The Ontario Ministry of Environment is starting to look at how sound propagation over water should be modelled and sought input in two meetings in Toronto.

2.3 CAC TC43 SC1, 2 – Stephen Keith

SCC Advisory committee for ISO TC43 and TC43/SC1

Current CAC membership: Alberto Behar (vice chair), Stephen Bly, John Bradley, Bill Gastmeier, Christian Giguère, Dalila Giusti, Stephen Keith (chair), Tim Kelsall, Emanuel Mouratidis, Colin Novak, Dave Quirt, Cameron Sherry, Jeremy Voix

New CAC members: Helen Ule: ISO532-x Loudness evaluation, ISO16254 Measurement of minimum noise emitted by road vehicles.

ISO active working group memberships:

Stephen Keith:
- TC43 Technical Advisory Panel
- ISO374x, ISO1120x Machinery noise emission
- ISO26101 Characterization of anechoic chambers
- ISO1996-x Environmental noise

Alberto Behar, Christian Giguère, Jeremie Voix:
- ISO4869 Hearing Protectors

Colin Novak, Helen Ule:
- ISO532-x Loudness evaluation
- ISO16254 Measurement of minimum noise emitted by road vehicles

2009 ISO Plenary meetings in Seoul, Korea.

New working group activity

- o New standard on "Measurement of minimum noise emitted by road vehicles"
- o New standard on "Compatibility between indoor and outdoor testing of road vehicles"
- o To be upgraded to full standard DTS 28961 "Acoustics – Statistical distribution of normal hearing thresholds under free-field listening conditions"
- o Planned revision of ISO 226:2003 "Acoustics – Normal equal-loudness-level contours"
- o Planned revision of ISO 389-7:2005 "Acoustics – Reference zero for the calibration of audiometric equipment – Part 7: Reference threshold of hearing under free-field and diffuse-field listening conditions"
- o Planned revision of ISO 17201-2:2006 "Acoustics – Noise from shooting ranges - Part 2: Estimation of muzzle blast and projectile sound by calculation".

Next ISO Plenary meetings: London, England, April 2011.

2.4 IEC – Lixue Wong

This report mainly summarizes the committee work of CSC/IEC/TC 29 since May 2010.

IEC Documents Revision

- o IEC 62489-2 Ed.1: Electroacoustics - Audio-frequency induction loop systems for assisted hearing - Part 2: Methods of calculating and measuring the low-frequency magnetic field emissions from the loop for assessing conformity with guidelines on limits for human exposure (Close Date: 2010-12-10)
- o IEC 61672-3Ed.2: Electroacoustics - Sound level meters - Part 3: Periodic tests (Close Date: 2011-02-28)
- o IEC 61672-2: Electroacoustics - Sound level meters - Part 2: Pattern evaluation tests (Close Date: 2011-02-18)
- o IEC 61672-1: Electroacoustics - Sound level meters - Part 1: Specifications (Close Date: 2011-02-18)
- o IEC 62585: Electroacoustics - methods to determine corrections to obtain the free-field response of a sound level meter (Close Date: 2011-02-04)
- o New Work Item Proposal on Hearing Instruments and Hearing Systems (Close Date: 2010-11-19)

Voting results

- o IEC 60118-15 Ed.1: "Electroacoustics - Hearing aids - Part 15: Methods for characterising signal processing in hearing aids with a speech-like signal"
  Final Canadian Position - Support with Comments

Next IEC/TC29 meeting: London, England
March 28 – April 1, 2011.

2.5 Z107.10 / Editorial – Cameron Sherry

Cameron Sherry could not make the meeting as he was at his son's wedding. David Quirt agreed to act as vice chair for the Editorial Subcommittee.

2.6 Building Acoustics – David Quirt

This report presents an overview of immediate suggestions for Z107-10, together with updates on key standardization activity in ISO/TC43/SC2 and ASTM E33, the two standards committees of obvious relevance for Canada.

Building Acoustics in "document formerly known as CSA Z107-10": Summaries for 13 ASTM standards were in CSA Z107-10, as published in 2006; eight of these have since been revised or reapproved and those entries should be updated in the next revision, at least to the extent of identifying the current version.

At least two other standards should be added to the document CSA Z107-10:

- o ANSI S12.60-2002, "Acoustical Performance Criteria, Design Requirements, and Guidelines for Schools". A draft entry for Z107-10 has been prepared.
- o ANSI-ASTM E2638-2008, "Standard Test Method for Objective Measurement of the Speech Privacy of Closed Rooms". A draft entry for Z107-10 can easily be prepared, including reference to the related requirements for federal government buildings.

Issues in ISO/TC43/SC2:

Steady advance of the ISO standards beyond their ASTM counterparts invites serious consideration of eventually basing the noise control provisions in the National Building Code on ISO standards, but meanwhile they provide technical content for ASTM to use.

More members joining Canadian Advisory Committee to ISO/TC43/SC2 would be nice, but there has been no systematic recruiting. Voting by current members has been erratic. Those interested in participating in the building acoustics CAC are encouraged to contact Dave Quirt.

The ISO meetings in Seoul Korea in November 2009 brought advances in ISO drafts and added several useful new work items. Next meeting is in London in April, and JDQ will attend.

TC43/SC2/WG18/AHG3 is dealing with restructuring of the ISO 140 series of laboratory standards for airborne and impact sound insulation, to facilitate their use as the basis for product test standards and eliminate current inconsistencies among the parts. The new series (ISO 10140) has 5 parts (test codes for products, airborne transmission, impact transmission, measurement procedures, laboratory & equipment). The FDIS was approved and corresponding parts of ISO 140 have been withdrawn. Acceptance in North America, and/or harmonization with ASTM counterparts, remains contentious. After formal approval, these will become the standards for testing products for noise control in buildings in Europe. To help Canadian exporters, these should be referenced in Z107.10 as part of the information about corresponding ASTM standards.

Several new work items are underway in TC43/SC2:

- o Precision for measurement of airborne and impact noise transmission, under AHG2 of WG18. First draft has been accepted as ISO/CD 12999 and includes new round robin information to deal with precision of airborne and impact noise measurements in lab and field. It seems obvious that error estimates are an important part of specification, compliance, and codes, and ISO leads ASTM E33 in this, with strong support from PTB (Germany). Brad Gover (BNG) is the Canadian participant, with JDQ as alternate.
- o Sound transmission through gaps and slits (pertinent for fire stops and for door or window seals): a new ad hoc group has been formed, but probably years away from CD.
- o Revision of ISO 717 (ratings for sound transmission) is beginning; BNG has been nominated as a Canadian participant, with JDQ as alternate.
- o Revision of field sound transmission standards (remaining parts of ISO 140) is beginning; JDQ is acting as formal Canadian participant (with others from NRC attending meetings so far). If the National Building Code changes from its current simplistic focus on the separating wall or floor assembly, then these standards (and their ASTM counterparts) will become the main focus foor noise control in buildings.

Issues in ASTM E33:

Members of our CAC have leading roles within ASTM Committee E33, which is responsible for standards in “Building and Environmental Acoustics”. Most recent meeting was the first week of October 2010. Trevor Nightingale is Chair of Subcommittee E33.03 (responsible for all ASTM standards pertinent to sound transmission in buildings, and hence building codes). BNG is leading several task groups in E33.03 and Chair of Subcommittee E33.05, Research (currently dealing mainly with issues for microphone specification and for statements of precision & bias).

Current activity in ASTM E33 includes work on ASTM E336 (airborne sound transmission in field), ASTM E1007 (field, impact transmission) and others.

Activity to maintain and revise ASTM standards is presented on the ASTM website, and for building acoustics, this is at http://www.astm.org/COMMIT/SUBCOMMIT/E33.htm.

For each current standard, there is a brief summary of significance and use, plus the scope, and an outline of the issues for any current revision.

3.7 Human Vibration – Tony Brammer

Members of the Canadian Advisory Committee are: Dr. Alberto Behar (ON), Dr. Paul-Emile Boileau (IRSST, QC), Dr. Anthony Brammer (Chairman), Dr. Tammy Eger (Laurentian University, ON), Dr. Ron House (St. Michael’s Hospital, Toronto, ON), Mr. Ed Lehtinen (Impacto Protective Products, ON), Dr. Pierre Marcotte (IRSST, QC), Dr. Jim Morrison (Shearwater Human Engineering, BC), Dr. Subhash Rakheja (Concordia University, QC), Dr. Dan Robinson (Robinson Ergonomics, BC), Mr. Mike Robichaud (Chairman of CAC/ISO/TC 108), and Dr. Vic Schroter (MoE, ON). The subcommittee is looking for more members.

The subcommittee is harmonized with the Canadian Advisory Committee on ISO/TC 108/SC 4 “Human Exposure to Mechanical Vibration and Shock”, and operates in parallel with SC4 of CSA Technical Committee on Occupational Hearing Conservation S304. The subcommittee continues to direct its efforts in support of the development of international standards. In this role, members of the subcommittee serve as conveners of two of the Working Groups (WG5 - Biodynamic Modeling, and WG8 - Vibrotactile Perception).

ISO/TC 108/SC 4 last met in London in September 2010. The most significant developments were:

(1) A decision to re-open the possibility of revising the omnibus standard on whole-body vibration (ISO 2631), and re-allocate the subject material so that all applications, e.g., comfort, health, and motion sickness are treated in separate parts of the standard, or Annexes. The “main” standard (Part 1) might therefore contain little more than frequency weightings.

(2) A decision to revise the standard on exposure to repeated shocks (ISO 2631-5) to change the biodynamic model used for estimating the effects of shocks on the spine. Two models are being proposed for the “z-direction” (i.e., along the axis of the spine): the first for shocks of magnitude up to about 20 m.s-2 , for use in assessing occupational exposures in industry, and a second model for

larger shocks such as encountered in military vehicles and fast boats.

(3) A decision to revise the standard on hand-transmitted vibration (ISO 5349-1) to include a frequency weighting specifically to assess the potential of vibration at different frequencies to precipitate vascular and neuro-sensory symptoms in the hands ("vibration-induced white finger" - VWF). In a related development, members of the CAA subcommittee are organizing a workshop on the suitability of the existing ISO frequency weighting for assessing the risk of VWF at the forthcoming International Conference on Hand-Arm Vibration to be held at Ottawa in June 2011. It is expected that the outcome of the workshop on the need for a second frequency weighting and, if appropriate, a proposed weighting function will have a large influence on the acceptance of such a change being accepted for the international standard.

(4) A decision to revise the standard that describes the biodynamic response of the hand to vibration (ISO 10068). The revision will include estimates of the frequency dependency of vibration that produces equal energy absorption in substructures of the hand (e.g., fingers, palm, wrist), and can be used to predict a frequency weighting for injury in the fingers. This frequency weighting is an important source of information for the revision of ISO 5349 (see 3, above).

List of International Standards prepared by ISO/TC 108/SC 4

- ISO 2631-1:1997 Mechanical vibration and shock — Evaluation of human exposure to whole-body vibration — Part 1: General requirements
- ISO 2631-2:2003 Mechanical vibration and shock — Evaluation of human exposure to whole-body vibration — Part 2: Vibration in buildings (1 Hz to 80 Hz)
- ISO 2631-4:2001 Mechanical vibration and shock — Evaluation of human exposure to whole-body vibration — Part 4: Guidelines for the evaluation of the effects of vibration and rotational motion on passenger and crew comfort in fixed-guideway transport systems
- ISO 2631-5:2004 Mechanical vibration and shock — Evaluation of human exposure to whole-body vibration — Part 5: Method for evaluation of vibration containing multiple shocks
- ISO 5805:1997 Mechanical vibration and shock — Human exposure — Vocabulary
- ISO 5982:2001 Mechanical vibration and shock — Range of idealized values to characterize seated-body biodynamic response under vertical vibration
- ISO 6897:1984 Guidelines for the evaluation of the response of occupants of fixed structures, especially buildings and off-shore structures, to low frequency horizontal motion (0,063 to 1 Hz)
- ISO 8727:1997 Mechanical vibration and shock — Human exposure — Biodynamic coordinate systems
- ISO 9996:1996 Mechanical vibration and shock — Disturbance to human activity and performance — Classification
- ISO 10227:1996 Human/human surrogate impact (single shock) testing and evaluation -Guidance on technical aspects
- ISO 10326-1:1992 Mechanical vibration — Laboratory method for evaluating vehicle seat vibration — Part 1: Basic requirements
- ISO 10326-2:2001 Mechanical vibration — Laboratory method for evaluating vehicle seat vibration — Part 2: Application to railway vehicles
- ISO 13090-1:1998 Mechanical vibration and shock — Guidance on safety aspects of tests and experiments with people — Part 1: Exposure to whole-body mechanical vibration and repeated shock

Future event: In June 2011 there will be a Human Vibration meeting in Ottawa.

2.8 Loudness Evaluation – Colin Novak

No report

2.9 CSA Z94.2 – Alberto Behar

See CSA section 1.1 above

3.10 New Business

3.11 Next Meeting and Adjournment

It was suggested that the standards meeting be held just before the CAA board meeting in the spring, preferably sequentially.

The next meeting will be held in the Spring in conjunction with the CSA TC meeting.

DST A NOVEL APPROACH FOR NOISE DEPENDENT HEARING PROTECTORS

Engbert Wilmink, Pieter van 't Hof
Dynamic Ear Company, Delft, The Netherlands, www.dynamic-ear.com, ewilmink@dynamic-ear.com

For most employees, noise levels are continuously changing. Putting machinery on and off, walking in and out high noise level areas etc. Giving those employees hearing protection during working day unnecessarily diminishes their quality of communicating for a large part of the day. However inserting hearing protectors and removing them a couple of times a day is uncomfortable and may easily lead to irritated ears. Besides a lot of noises arise unexpectedly, leading to unprotected ears exposed to noise. Secondly, because of liability, employers have to oblige their employees to wear hearing protection when they are exposed to a daily sound dose ≥ 85 dB in Holland. For this reason various professionals (pilots, dentist, musicians, craftsmen etc.) asked for a noise level dependent hearing protector solution.

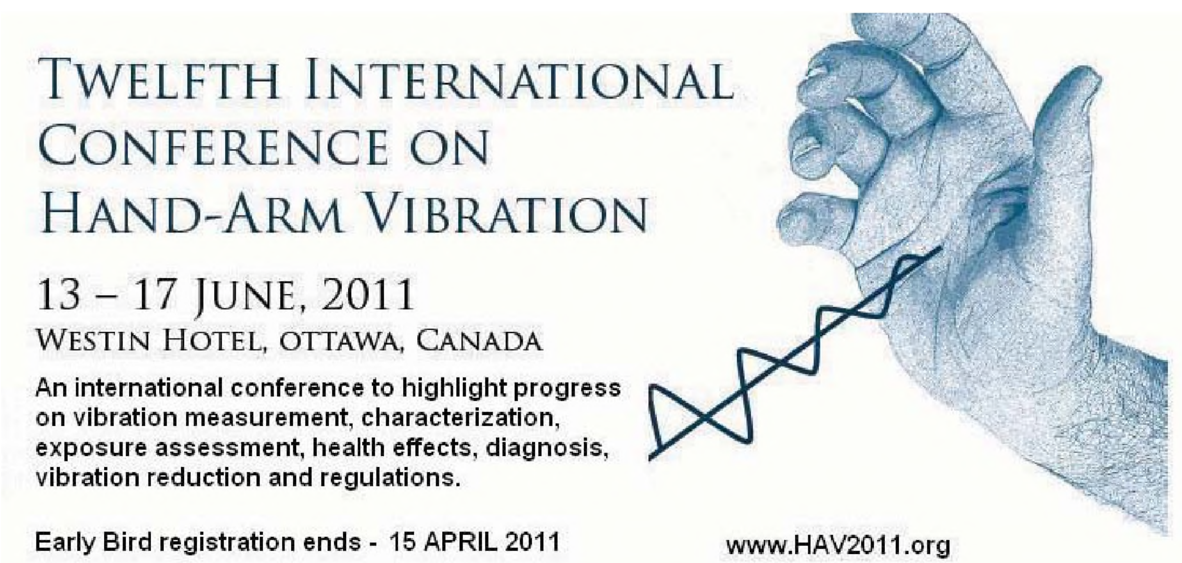
At Dynamic Ear Company we developed the novel concept of Dynamic Sound Technology (DST). With DST, the momentary sound level determines the attenuation of the hearing protector, giving the user no more attenuation than necessary. Main advantage of the system is that is gives the user the possibility to communicate freely, when the noise levels are acceptable, without removing the hearing protectors from the ear.

DST is a mechatronical system that is based on an automatically opening and closing gate. With the help of a microphone our custom-made IC calculates the sound pressure level (SPL). The SPL is compared to a reference level (e.g. 80 dB). The difference between the SPL and the reference level determines whether the acoustic valve needs to be (partly) opened or can be closed. Within the dynamic range of the damper (20 dB), it will try to keep the sound level in the ear on or below the reference level.

SPL is calculated using a B-weighted filter. Although A-weighting is mostly used in Europe to present sound levels, B-weighting was chosen for DST. This is because it is somewhere in between the required A-weighting and C-weighting. The latter was developed for the high sound levels where hearing protection is required. More important, B-weighting pronounces low frequency noise more. This is preferred, because noise caused by machinery usually has the most energy in the lower frequencies and low-frequency noise of a certain high SPL is as harmful to the ear as high-frequency noise of the same SPL.

The acoustic valve reacts on the momentary SPL outside the ear with a programmable attack and decay time. For dynamic hearing protection required to decrease the noise level, the attack time is short (~ 30 ms), to keep the daily sound dose at the ear as low as possible. For the same reason, the decay time is chosen to be rather long (> 5 s), because it is very likely that a certain noise will be repeated. For dynamic hearing protection required for musicians and visitors of concerts, we expect to use an attack time comparable to the decay time, because as much of the sound and the dynamics should be intact. To find suitable attack and decay times field tests will be conducted with a Dutch orchestra.

According to EU regulation Hearing protection needs to be certified. Standard REAT measurements are not possible, because the dynamic hearing protector is open at threshold levels. However, REAT measurements can be conducted when the device is turned off to determine the maximum attenuation level. A level dependant test should be carried out as well in order to find the reference level.



PROVIDING 'GOOD', 'BETTER' OR 'BEST' ACOUSTICAL PLUMBING SYSTEM PROPOSALS FOR COST-SENSITIVE CLIENTS

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1. INTRODUCTION

Throughout the conceptual and design stages of a building's construction cycle, numerous decisions are made and modified, based on cost/benefit measurements. Early on, the Building Owner and the design team make determinations as to the quality level target for a project, based on market needs and trends as well as financial capabilities and interests of investors. For instance, in the case of an office building, will it provide "Class A" or a "Class B" office space? These decisions affect nearly every aspect of a building's parts and pieces. In this case, we will concentrate on the noise and vibration options and choices related to a building's plumbing and piping systems.

When it comes to the costs associated with effective acoustical isolation of a plumbing system there are a variety of choices available. Though effective isolation materials and methods are available for very modest costs, there is often the challenge of "Value Engineering" to face. As in most facets of building construction, there are a variety of quality levels available when it comes to plumbing system's acoustic isolation options.

Becoming familiar with "Good", "Better" and "Best" materials and methods, in order to be helpful during the budgeting and design stages of a building project can make you a valuable asset to the entire design team and to the Building Owner. Learn to provide valuable input during the Cost–Benefit analysis for a building project. Base your input to your client upon solid laboratory test data arranged by specific plumbing system applications, such as through-stud isolation, riser clamp isolation, shower head attachments, hanger isolation, etc.

This paper will provide fundamental presentation points and cost control advice for any plumbing or piping system. This information takes into account both labor and material factors, in order to generate a real world "Installed Cost Analysis", while specifying a proven engineered system for your client.

2. Common sources of plumbing noise

Both "Airborne" and Structure-borne" noises are involved, but for the most part we will focus on the structure-borne noise component. The airborne side of the issue is best left to discussions relating to wall and floor assemblies, etc. Plumbing system noise can affect both "STC" and "IIC" ratings, of course. The portions of plumbing and piping systems that come into play include: Drainage systems (Sanitary Waste and Storm Drain/Rain leader), water distribution systems, fixtures, faucets & appliances, valves, pumps and equipment.

The main issue to be addressed is the breaking of direct contact between the piping system components ant the buildings multiple components and surfaces. Over the years, when contractors have been directed to take action regarding acoustical isolation of plumbing systems they have made a wide variety of attempts to one level or another and with widely varying results. When contentiously performed, effective acoustical noise and vibration isolation of these systems can reduce the noise perception by more than half! Some of the comparative examples of failed and successful methods include the ones shown in Figures 1.1 through 1.4.

Fig 1.1 Water tubing isolation using foam insulation and successful use of an engineered isolator

Fig 1.2 Paper compared to an isolated tube clamp

Fig 1.3 Shower head support compared to an engineered anchor point

Fig 1.4 Mid-span support (vertical/horizontal) showing a questionable use of felt and tie wire, and to engineered tube clamps

3. Why do contractors' bid prices skyrocket when an acoustical spec is included in the project scope?

The answer is simple. Most people tend to resist change. Contractors are no different. They prefer to continue with status quo and when told they must modify their means and methods of installation and install specialized materials they tend to retaliate by sharply increasing their prices. They do this because, for the most part, Contractors and individual tradesmen have little or no knowledge of acoustics and how to effectively succeed in meeting the requirements being proposed…which they don't understand in the first place and for which their installers have little or no training to perform. As a result they believe that they will be corrected and will have to re-work a great deal of their installations after their initial attempts are deemed ineffective by an engineer. Please remember, plumbing codes typically do not include requirements related to comprehensive noise and vibration isolation. Why do they resist? Most often it is because the direction given in the project's specification documents in vague and without detail. They also view an acoustical engineer/consultant as "just another authority to answer to", which they believe will translate into productivity slow-down and countless correction notices. Productivity slowdowns translate into the need to charge higher prices to cover their anticipated costs.

Additionally, many of these contractors fear and may have actually experienced litigation resulting from their failure to succeed in meeting the criteria imposed by an acoustical specification.

4. Tiered Cost Options in Today's Economy

In today's economy, Building Owners and Developers are looking for value as much or more than ever. Much of this is due to market uncertainly and low "ROI" expectations. As a result, many feel they cannot afford the "Best". As a result, when the project is in its design and/or pre-construction stages it is often faced with the need for "Value Engineering". During this process, having the ability to offer tiered choices, or "Good", "Better", "Best" options may well result in your continuing to be retained as an acoustical engineer and/or consultant to one level or another, rather than possibly facing the reality of being completely removed from project's scope all together. "All or Nothing" is not a good place to find yourself in when it comes to the chopping block called "Value Engineering". "Good", "Better", "Best" options are available for plumbing and piping system acoustical isolation.

4.1 Providing Options for Building Owners

To help you determine your client's perceived value of a building that is well isolated against noise transfer, ask yourself these two questions: 1) Might investing in a quieter building garner increased rent or sales revenue? 2) Might providing a quieter building serve as a way to help avoid possible litigation at the hands of unhappy building occupants later on? Depending on the answers to those two Cost/Benefit questions, provide tiered options to your client. Here are some examples that might be employed for a multi-story multi-family building such as a condominium or a hotel.

4.1.1 Minimal isolation ("GOOD")

- Soft isolators at all penetration locations of plumbing supply water and drainage lines, such as through studs, joists & at hanger support points
- Rubber/neoprene isolators under equipment
- Flexible water flex connectors, rather than hard piped connections

Cost: Approx. $100 material and 1Hr of added labor (above code minimum requirements) in a 2-Bath dwelling

4.1.2 Mid-range isolation ("BETTER")

- Soft isolators between all piping and building assemblies, such as through wall studs, ceiling joists, hangers & under all pipe riser clamps
- Soft isolators at mixing valves, showerheads & similar attachment points
- Spring isolated pipe connections to equipment and concrete inertia pads at equipment bases
- Spring isolators at hangers in mechanical rooms_and within 50' of mechanical equipment connections
- Braided/non-metallic hose connections at equipment
- Cast iron drainage piping, rather than plastic
- Use of braided water connectors at all fixture faucet connections and toilet inlets

Cost: Approx. $300 of material (mostly related to the upgrade to cast iron pipe) and 5 Hrs of added labor (above and beyond code minimum requirements and mostly related to the slower installation of cast iron pipe) in a 2-bath dwelling

4.1.3 Premium isolation ("BEST")

- Soft isolators between piping and building assemblies, such as through wall studs, ceiling joists, hangers &

under all riser clamps

- Soft isolators at any and all possible contact points, including mixing valves and shower head piping
- Spring isolation and inertia pads at equipment bases
- Spring isolators at hangers on all drain lines, liquid pressure lines (such as domestic water & hydraulic lines) and on all suspended equipment hangers
- Braided/non-metallic hose connections at equipment
- Cast iron drainage piping, rather than plastic
- Isolate all plumbing fixtures and their attachment points to floors and walls with soft neoprene liner, such as at toilets, tubs, etc.
- Use of braided water connectors at all fixture faucet connections and toilet inlets

Cost: Approx. $400 of material (mostly related to the upgrade to cast iron pipe) and 8 Hrs of added labor (above and beyond code minimum requirements and mostly related to the slower installation of cast iron pipe and the fixture isolation) in a 2-bath dwelling.

5 Is “Good”, “Better”, or “Best” perceptible?

The following example shows 1 of 70+ tests performed by an ISO 3822 accredited lab, comparing common installation practices to installations employing acoustical materials readily available in today’s market:

- Water pressure for each test was 45 PSI, water flow rate was 4.6 FPS and the resulting acoustic spectra analyzed in 1/3 octaves bands, 80–10,000 Freq. Hz
- Application: Shower Head Installation


Fig. 5.1


Fig. 5.2


Fig. 5.3

Figures 5.1 thru’ 5.3 Three Anchoring Methods Tested and Compared

6 A quiet plumbing system

Many things can be done to help assure that your client gets the biggest bang for their buck. These things include: Influence pipe, fixture & equipment selections, influence locations of plumbing system components to avoid sensitive areas, select appropriate and effective acoustical isolation products to be used, assure material compatibility and ease of use and cost, customize your Project Specification language, include Installation Detail Drawings with instructions, require product submittals & samples for approval, inspect for compliance during construction, provide clear specification language, provide product Spec Sheets from manufacturers or specify model numbers, provide simple Installation Instructions from manufacturers, provide Installation Detail Drawings and produce a blueprint Installation Detail Page.

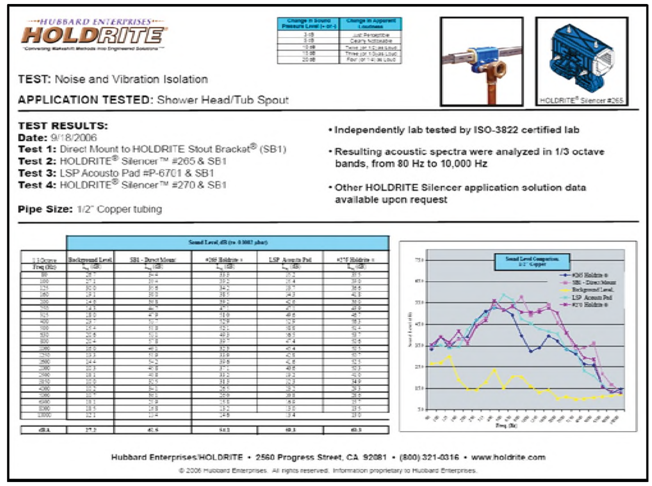

Fig 5.4 Noise Pressure Level Test Results
Background noise level- 27.2 dBA; Anchored directly to support bracket- 61.5 dBA Anchored to Acousto Pad #P-6701- 59.3 dBA Anchored to HOLDRITE Silencer #265- 54.1 dBA
NOTE: Sheet rock was installed on walls in each case, prior to noise level readings

7 Conclusion

Plumbing system noise mitigation should not be difficult. Partnering with committed manufacturers who can help you with “application specific” product solutions and the accompanying test data from neutral 3rd party labs will aid in your success as an acoustical consultant and help you avoid being “Value Engineered” out of the project scope.

Up-sell “sound quality” to “high-end” builders/owners by providing proof of affordability (positive R.O.I.) with “GOOD”, “BETTER” and “BEST” options.

REFERENCES

1. American Society of Plumbing Engineers (ASPE)- Plumbing Engineering Design Handbook, Volume 1 – Fundamentals of Plumbing Engineering, Chapter 10
2. ISO 3822: Laboratory Test of Noise Emissions from Appliances and Equipment used in Water Supply Installations. International Organization for Standards
3. Engineering Resource Binder, Hubbard Enterprises- HOLDRITE Silencer installation instructions

Canada Wide Science Fair

From File Reports

Cody Shaw is the winner of this year's Special Award from the Canadian Acoustics Association for his project "Phonic Crystals - Revisted."

Cody Shaw is a Grade Twelve student from St. Agathe, Manitoba. His interests include, computers, particle physics and electrical engineering. In addiiton to his extensive participation in Science fairs, he is part of a charity group, "The Speed Gamers," who do lengthy video game marathons for various charities. They have raised over $130,000 so far. Cody plans to do a double major in Particle Physics and Electrical Engineering. He also wants to pursue a PhD in Electrical Engineering with a goal to work with Large Hadron Colllidors.

Cody Shaw's's full article is reproduced below.

Phononic Crystals: Revisited*

Cody Shaw
Fort Richmond Collegiate, St. Agathe, Manitoba, Canada

Editor's Note: The submission by Cody Shaw was reformatted and edited to fit in to the Journal format.

1 INTRODUCTION

This project is an enhancement of last year's project, 'Phononic Crystals and Anti-Noise.' The intent of the project was to demonstrate the effects that a Phononic crystal has on audible sound propagating through air. The primary result from this project is to show the frequency dependant ratio of propagated and incident sound pressures (transmission coefficient). The experimental results correlate well with predictions, unlike the results of the previous year.

Phononic crystals utilise the periodicity of a crystal's structure and the resulting interferences of the sound waves to filter sound as it propagates through the crystal, creating a "band gap" at certain frequencies. The frequencies that are affected by the crystal depend on the distance between the rows of the crystal lattice (this is how the dimensions for the crystal's construction were determined). This has been demonstrated extensively in the ultrasonic range of frequencies, but t the audible frequency range for sound in air remains untested. The aim of this project was to demonstrate that phononic crystals can be scaled to affect audible sound as they do ultrasonic frequencies. One also hopes to produce more accurate experimental results from the focus placed on multidisciplinary work.

2 IMPROVEMENTS AND CHANGES

The enhancements made to produce this year's superior results are multidisciplinary in nature. Knowledge from physical acoustics, automation, electrical engineering, computer science and general science were applied to make the measurements. Enhancements were also produced by improvement in the methodology. An example of this is taking extensive measurements that define the placement of the crystal in the anechoic chamber, relative to the X-Y table, microphone, and loudspeaker. This ensured that whenever the crystal had to be removed from the chamber it could be replaced in the same position.

3 PROCEDURE

The experiments began with locating an anechoic chamber in which the measurements would be conducted. An anechoic chamber is essential to the project as it will "absorb" many of the reflections that biased my previous results. Further improvements of the experiment were the use of laboratory quality microphones, preamplifiers, and measurement instrumentation. Upgrades were also made to the excitation, including the use of a commercial loudspeaker and combination mixer/power amplifier. Automation in the form of an X-Y motor table under computer control was used to reduce biases from human interaction and enhance repeatability of the experiments. The tedious nature of the measurement process was greatly reduced as a by-product. The computer control that integrated motor control, source excitation, and data capture was achieved by adapting and enhancing a previously existing C++ program [1] to be used for the current application. Fig. 1 shows a picture of

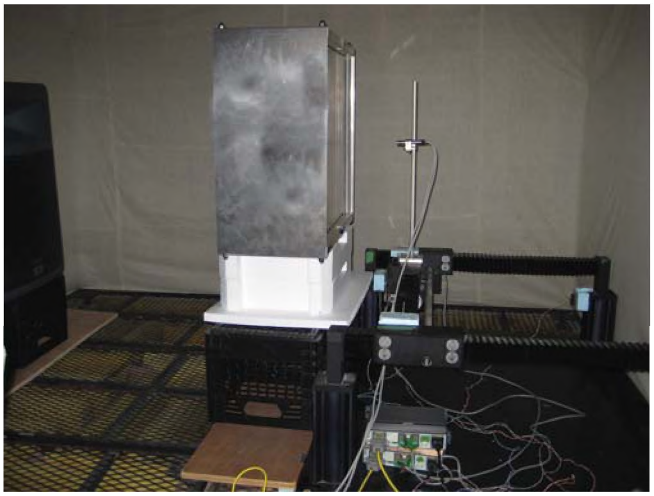

Figure 1: Experimental Set-up.

the crystal in position within the anechoic chamber, with the speaker on the left and the microphones attached to the X-Y motor table on the right. This was the consistent setup for every single one of the tests. The only changing variable with the tests and the apparatus was the inclusion of the crystal in the chamber, and the actual program that was swapped out for each test.

4. OBSERVATIONS

Fig. 2 shows a preliminary comparison of the pressure ratios, P_{res}/P_{ref}, that result when the phononic crystal is used. The solid line represents a measurement done using "Steady State" sine wave excitations while the dashed line represents measurements done using a Gaussian modulated sine wave. Sine wave testing is generally more accurate than Gaussian pulses, but Gaussian pulses are faster to implement. The graph of the Gaussian pulse demonstrates the same general trends as the graph using Sine waves. (Discrepancies can be explained by leakage effects from the DFT assuming a pulse to be periodic and unexpectedly large noise in the Sine wave testing due to uncontrollable building noise.) The generally good correlation implies that the Gaussian pulses are acceptable excitations to probe the crystal with. Position is also important, as a "comb filter" like effect is expected

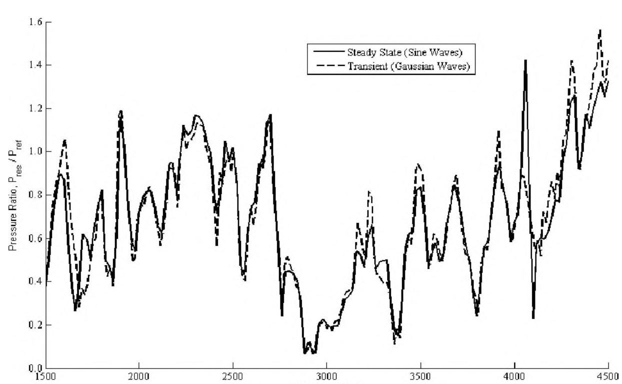

Figure 2: A comparison of Steady State and Transient preliminary test pressure ratios.

in front of the crystal, with some spots being louder than others. This is another improvement made, as it decreased test duration by hours, depending on the length of the test.

5. DISCUSSION

The main results from the tests are shown in Fig. 3 and Fig. 4. Fig. 3 shows a comparison between a reference pressure test (without the crystal in position within the chamber) versus a data pressure test (with the crystal in position) as a pressure ratio over a square area. The test is done two-dimensionally in front of the crystal face, to produce a three dimensional graph of position versus pressure ratio. This result is important, as it shows the expected sound filtering and periodicity of similar crystals that have been readily tested.

Fig. 4. shows the transmission coefficient across the face of the crystal. This, in essence, shows which frequencies the crystal is effective at blocking. As seen in the graph, there is a correlation between the experimental data and the computer generated theoretical data. The correlation in the band gaps between 2.5 and 4 kHz is very important, as it shows that the same principals apply to the sized crystals to the smaller crystals. This also shows that the larger crystals could be practically implemented in a way that would be beneficial to lower frequency problems that may arise.

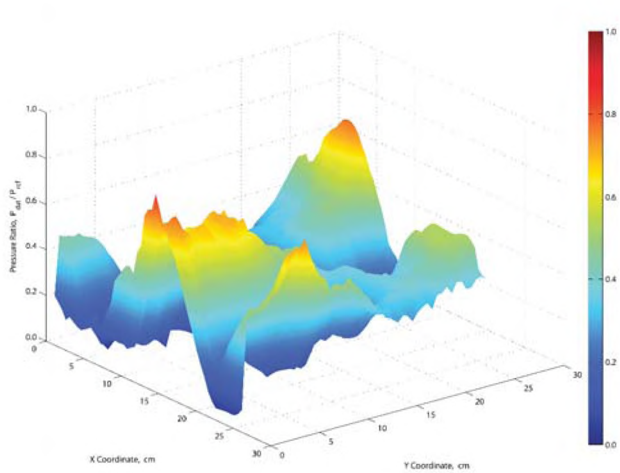

Figure 3: Comparison bnetween reference and data pressure tests.

6. CONCLUSIONS

The current results have definitely improved over earlier results due to a focus on methodological and multidisciplinary improvements. Being able to properly test, and then physically explain a property of a topic is extremely useful in the field of scientific research. The idea of turning this almost completely physics based project into a multidisciplinary project and mixing the knowledge of the disciplines helped the data acquisition and interpretation. The results show that crystals do have an impact in the audible ferquency range,

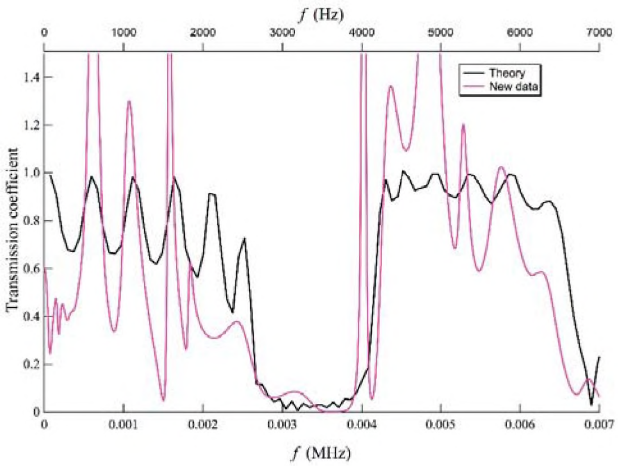

Figure 4: The Transmission coefficient of the crystal

but there is still room for improvement in the implementation of the testing process. An idea for a future enhancement would be to reduce or eliminate the ringing created by the loudspeaker when it is used to create excitations. The excess ringing by the speaker can give biased results if not taken into direct consideration as it adds unwanted data to the captured waveform graphs.

This project represents a substantial learning experience. Although last year's project was automated the softwares used for the movement or capturing of data were not prgrammed. MATLAB® is an extremely powerful array based coding language known for its ease of use in processing and reading graph data and it was used this year to manage and interpret the data. The propagation of Gaussian modulated Sine wave pulses required the use of Discrete Fourier Transforms (DFTs) to decompose the data into their corresponding frequency components. As copious amounts of data were collected, large files resulted. To implement effective storage strategies, the binary files were manipulated in both C++ and MATLAB®. This, combined with the Electrical Engineering and Physical aspect of the project, makes it multidisciplinary.

ACKNOWLEDGEMENTS

I wish to acknowledge contributions from Darryl Stoyko and Dr. Popplewell. Acknowledgements also go to Dr. Page, Anatoliy Strebeluvich, and Patrick Flemming, all of whom have helped and guided me to reach my goals with this project.

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The Effects of Low-Frequency Noise and Vibration on People

Edited by Colin H. Hansen
Multi-Science Publications, 2007
List price: $86 USD (Softcover)
416 pp., ISBN: 978-0-906-52245-5

For someone who works in the field of wind turbine noise generation, the above book is a timely compendium of research articles providing insight into the nature and results of exposure to Low frequency noise and vibration. Prof. Colin Hansen has ably edited together a collection of papers published in the Journal of Low Frequency Noise and Vibration. The compendium includes articles from 2000 through 2005.

The current collection includes 32 papers and cursory perusal shows that most of the research is conducted either in Europe or Japan. [Note: Detailed review of the 32 papers was not possible for this short book review]. The 32 papers are grouped together in five major headings and each chapter contains a brief introduction by Prof. Hansen as well as a brief summary of each research article. The summary also includes the page number of each article as they appear in the compendium. The five chapters are: 1) Perception of Thresholds of Low Frequency Noise; 2) Effect of Low Frequency Noise on People in terms of Annoyance and Sleep Deprivation; 3) Physiological Effects of Low Frequency Noise; 4) Perception Thresholds of Low Frequency Vibration and the Effect of Low Frequency Vibration on People in terms of Comfort and Annoyance; and 5) Physiological and Health Effects of Low Frequency Vibration.

The four papers of Chapter 1 discuss the level of unacceptable noise levels in the infrasound range. These papers present results from their test methods, resources applied for the tests and the subjects used for the testing. The four papers show that the perception thresholds of the two major standards, ISO 226 and ISO 389-7, correlate well with the research results.

The sixteen papers of Chapter 2 provide a background for the understanding of the effect of low frequency noise. The main focus of the papers was on sleep deprivation and annoyance. Some of these papers present results of surveys conducted where low frequency noise complaints occurred. The common concerns of low frequency noise sufferers are also discussed in some of the papers. The difference in perception between sufferers and non- sufferers is the primary focus in some of the papers presented in Chapter 2. A key parameter, assessment method is discussed by one paper and it showed that the Danish assessment method is superior to methods sued in other European countries. One of the main complaints about infra sound studies has been the use of ‘dBA’ descriptor and the unsuitability of ‘dBA’ was also discussed in this chapter.

Chapter 3 contains seven papers and is supposed to discuss the physiological effects of low frequency noise. This chapter should have been better organized. Paper 4 is a repeat of paper 10 and Chapter 2. Only a few papers, three to be exact, discuss the broad definition of physiological effects of low frequency noise.

Chapter 4 (two papers) present results of vibration perception thresholds. The chapter introduction states that hearing threshold of low frequency vibration, but must be corrected to state that vibration perception. The first paper discusses the perception threshold for heavy truck drivers above which the levels become uncomfortable. The second paper compares two different test methods, Polish method and ISO 13091-1-2001 and highlights the need for a better standardized assessment method for perception threshold.

The three papers of Chapter 5 present the results of their research on physiological and health effects of low frequency vibration. Vibration induces disease, the impact on the autonomic system as well as whole body vibration effects were addressed by these papers.

The book provides a general introduction for non-researchers working in the areas of infrasound and low frequency vibration. All the basic issues and concerns commonly encountered in low frequency noise and vibration are addressed by these 32 papers. The above edition is thus a valued collection for acousticians who work with noise in the infrasound range.

Prof. Ramani Ramakrishnan
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Canadian Acoustical Association

Minutes of the Board of Directors Meeting
Victoria, British Columbia
12 October 2010

Present: Christian Giguère (chair), Jérémie Voix, Hugues Nélisse, Frank Russo, Tim Kelsall, Clair Wakefield, Stan Dosso, Sean Pecknold, Ramani Ramakrishnan, Bradford Gover, Roberto Racca

Regrets: Rich Peppin, Dalila Giusti

The meeting was called to order at 5:05 p.m. Minutes of previous Board of Directors meeting on 01 May 2010 were approved as published in June 2010 issue of *Canadian Acoustics*. *(Moved by T. Kelsall, seconded R. Racca, carried)*.

President's Report

Christian Giguère reported that there have been no major problems in the affairs of the Association. There has been activity on two current priorities: to enhance the website with online database and payment features (discussed as an extension to the Secretary's report), and to form and activate a new Acoustical Standards Committee (discussed as its own agenda item). Also identified was the activity to develop promotional materials (starting with a new logo) for our Association (discussed under Other Business).

Secretary's Report

Bradford Gover reported that routine processes of the Association are proceeding with few problems. With respect to routine CAA communications:

- Annual filing with Corporations Canada was submitted and acknowledged.
- Invoices from I-INCE and ICA were received and our Treasurer handled payment.

Secretarial operating costs for the fiscal year totaled $965, mainly for mailing costs and postal box rentals. A budget of $1000 is proposed for next fiscal year.

Paid new memberships and renewals are down somewhat from last year (301 compared to 374). This is undoubtedly due to the instructions to members to wait for the online system to become available, which has not yet happened. There are an additional 140 members and subscribers who are pending renewal for 2010. If half of these do (which is reasonable to expect), then the total membership is essentially unchanged since last year. As in previous years, the Association's core membership is essentially constant.

Category	Paid 2010	Change From 2009	Pending 2010
Member	184	–61	94
Emeritus	2	+1	0
Student	52	–8	39
Subscribers	63	–5	7
	301	**–73**	**140**

The decision was made to contact the 140 "Pending" members to arrange payment of 2010 dues, rather than wait for the online capability. *(Approval of report moved by R. Ramakrishnan, seconded C. Wakefield, carried)*

Discussion then turned to the proposed new online database and payment system, and particularly to a means to expedite its implementation. The decision was made to contact the service providers (IRM Membee) to identify a way forward that could proceed more rapidly. Christian Giguère will follow up on behalf of the Board.

Treasurer's Report

The Treasurer, Dalila Giusti, submitted a report including a preliminary financial statement for the fiscal year. Most expenses were essentially as budgeted, although journal costs were higher than forecasted. Revenue from membership dues was down (due to the large number of pending renewals), and likewise, revenue from journal advertising was also down. On the other hand, the 2009 Conference (Niagara-on-the-Lake) made a profit of $28,000, so overall, revenue well exceeded expenses. Interest on the capital fund will exceed costs for student awards.

The proposed budget for 2010-2011 was also discussed. At present, the planned budget for 2011 predicts a deficit, due to increased journal and website costs, and to reduced advertising revenues. Clearing the backlog of membership dues should ameliorate the situation somewhat.

The Treasurer's report was accepted. *(Moved S. Pecknold, seconded R. Ramakrishnan, carried)*

The Board agreed in principle to raise the advertising rates in the journal, with the details to be determined after discussion with the Treasurer and the Editor. *(Moved T. Kelsall, seconded R. Racca, carried)*

Editor's Report

The Editor, Ramani Ramakrishnan, gave a brief report on issues related to *Canadian Acoustics*. Highlights included:

- All issues have been published on schedule.
- Preparations for eventual online publication of the journal have progressed modestly.
- Plans for journal issues in 2011 include:
    - March: Will feature (only) invited papers in the area of Architectural Acoustics.
    - June: Proceedings of the 12th International Conference on Hand-Arm Vibration, to be held in Ottawa.
    - September: Proceedings of the annual CAA Conference.

Ramani reminded the Board that, as announced in the minutes of the spring BoD meeting printed in the June 2010 issue, he is planning to not seek election as Editor in Chief in 2013, and would like to use the time until then to assist in the transition of a replacement. Any individuals interested in being considered as the next editor are asked to contact Ramani as soon as possible.

The Board made a unanimous vote of thanks to Ramani for his continuing contributions.

CAA Conferences – Past, Present & Future

2009 (Niagara-on-the Lake): A final report for the conference has been received from conference chair Ramani Ramakrishnan, with the final transfer of funds. The Board thanked the organizers for the high quality of the very successful meeting.

2010 (Victoria): The conference at the Laurel Point Inn, October 13-15, has 110 papers scheduled and 15 exhibitors. Stan Dosso is Chair, Roberto Racca is Technical Chair, and Clair Wakefield organized the exhibition.

2011 (Québec City): The conference will be held in mid-October in Québec City. Conference Chair Christian Giguère will announce details as soon as possible. Watch for announcements in *Canadian Acoustics,* and on the website

Subsequent meetings: Several options for future conferences were discussed. At present, there are no firm plans for 2012 or later.

In June 2013, the International Congress on Acoustics (ICA) will be held in Montréal, and a proposal has been received from John Bradley and John O'Keefe for the CAA to sponsor a satellite International Symposium on Room Acoustics (ISRA). The board discussed and agreed to approve this proposal *(Moved T. Kelsall, seconded R. Ramakrishnan, carried)*

Awards

Frank Russo presented a report summarizing decisions by the coordinators for all CAA awards. There were eligible applications for all awards except the Shaw Postdoctoral Prize, and winners have been selected. Winners were announced on 14 October at the banquet, and in this issue of *Canadian Acoustics*.

It was proposed that two new student awards be created: one in the field of "Architectural Acoustics", and one in the field of "Psychological

Acoustics". A motion was made to approve the creation of these two new awards, in principle, with monetary value to be decided later. *(Moved F. Russo, second S. Pecknold, carried)*

Acoustical Standards Committee

Tim Kelsall reported on the status of the new CAA Acoustical Standards Committee. This committee was formed after the Canadian Standards Association (CSA) disbanded committee Z107, and reorganized their remaining standards related to acoustics. The CAA Acoustical Standards Committee met concurrently with the new CSA standards committee on 13 October. A report is printed in this issue.

CAA Website

Sean Pecknold reported that routine maintenance of and updates to the website have been ongoing. A major revision is on hold until such time as the new online member database and registration capabilities come online.

Other Business

There were several items of other business:

- Alberto Behar attended for a short time as a guest to present the role that the CAA could play in offering distance Education in Acoustics courses for professionals, like the Association of Australian Acoustical Consultants does. A lively discussion ensued. The next step was to identify the university programs in Canada that teach acoustics. Christian Giguère will follow up.
- Following on previous Board of Directors discussions of a new logo for the Association, several proposals were presented for consideration. Favourable discussion ensued, and Christian Giguère will get back to the designer with feedback.
- Jérémie Voix brought up the idea of a "linked in" or "facebook" type online presence for the CAA, which could be specific for a single meeting or event, or more permanent to enable acoustical experts to more easily network, to generate pre-conference momentum, etc. Jérémie agreed to investigate a suitable next step, perhaps in time for next year's meeting in Québec City.
- Stan Dosso led a brief discussion of nominations for the election at the Annual General Meeting (See AGM minutes for details).

Adjournment

Meeting adjourned at 10 p.m. *(Moved S. Dosso, seconded R. Racca, carried)*

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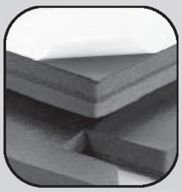

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Canadian Acoustical Association

Minutes of Annual General Meeting

Victoria, British Columbia
14 October 2010

Call to Order

President Christian Giguère called the meeting to order at 5:35 p.m. with 20 members present. (Attendance peaked at 26 during the meeting)

Minutes of the previous Annual General Meeting on 15 October 2009 in Niagara-on-the-Lake were approved as printed in the December 2009 issue of *Canadian Acoustics*. *(Moved by R. Peppin, second T. Kelsall, carried)*

President's Report

Christian Giguère briefly summarized his report to the Board meeting on 12 October. He emphasized that the society has stable membership and is maintaining a balanced budget, and he thanked all those who have made contributions to our activities. He also reported that the key priority for the coming year is shifting our operations to a new web-based system to facilitate routine membership and financial transactions.

Secretary's Report

Bradford Gover gave an overview of membership and operational activity.

- The total of 301 paid renewals and new memberships is down from last year, but this is of course due to the instructions to members to wait for online payment. Those members whose payment is pending will be contacted. The core membership is essentially unchanged from last year.
- An itemized account of the administrative budget of $965 (mainly mailing expenses) was presented to the Board of Directors.
- Steps are ongoing towards shifting the membership database and renewal process to an online system, and promoting a shift towards more email and online transactions, to handle routine processes with less volunteer effort.

Treasurer's Report

In the absence of the Treasurer, Dalila Giusti, Christian Giguère presented an overview of her written report to the Board on CAA finances. CAA is in good financial shape, with total assets of $293,585 at fiscal year-end (before audit). Total assets rose marginally, and interest on our capital investments will cover all prize awards.

In 2011, a budget deficit is predicted due to increased journal and website costs, and reduced advertising revenues. The Board agreed to raise advertising rates in the journal. Additionally, collecting the dues from members whose renewal is pending should ameliorate the situation.

(Acceptance of Treasurer's report moved by S. Dosso, second D. Quirt, carried.)

The Board is recommending leaving the membership dues for 2011 unchanged. *(Moved by S. Pecknold, second H. Forester, carried)*

Editor's Report

Ramani Ramakrishnan gave the Editor's report. *Canadian Acoustics* production has proceeded smoothly throughout the year, and content for issues is largely set through Dec 2011.

The cost of mailing paper copies of the journal (particularly abroad) has dramatically increased in recent years, and a move to an online journal is desirable.

Ramani announced that he does not intend to seek re-election as Editor in Chief after 2012. He would like to work closely with a replacement so that the transition. Anyone who might like to seek election to the position of Editor, is asked to contact Ramani as soon as possible.

Award Coordinator's Report

Frank Russo acknowledged the continuing hard work by award coordinators, and reported that this year CAA is awarding all prizes with the exception of the Shaw Postdoctoral Prize. In addition, we have sponsors for the three student paper awards for presentations at the conference. (For names of award recipients, see the separate announcement in this issue.)

Two new prizes have been authorized by the board, one in Architectural Acoustics, and one in Psychological Acoustics. Monetary value of these awards is to be determined.

Past and Future Meetings

Reports were presented on the past, present and future annual meetings:

2009 (Niagara-on-the Lake): A final report for the conference has been received from conference chair Ramani Ramakrishnan, with the final transfer of funds. The organizers were thanked for the high quality of the very successful meeting.

2010 (Victoria): The conference at the Laurel Point Inn, October 13-15, has 110 papers scheduled and 15 exhibitors. Stan Dosso is Chair, Roberto Racca is Technical Chair, and Clair Wakefield organized the exhibition. The organizers were complimented for the excellent job on the meeting.

2011 (Québec City): The conference will be held in mid-October in Québec City. Conference Chair Christian Giguère will announce details as soon as possible. Watch for announcements in *Canadian Acoustics,* and on the website

Subsequent meetings: Several options for future conferences were discussed. At present, there are no firm plans for 2012 or later.

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Acoustical Standards Committee

Tim Kelsall reported on the status of the new CAA Acoustical Standards Committee. This committee was formed after the Canadian Standards Association (CSA) disbanded committee Z107, and reorganized their remaining standards related to acoustics. The CAA Acoustical Standards Committee met concurrently with the new CSA standards committee on 13 October. A report is printed in this issue.

Nominations and Election

CAA corporate bylaws require that we elect the Executive and Directors each year. The Past President, Stan Dosso, presented nominations and managed the election process.

For the election process, Stan read the name(s) of the nominees, and then asked if there were other nominees from the floor.

- First, he presented names of proposed continuing Directors (Rich Peppin, Roberto Racca, Tim Kelsall, Clair Wakefield, Frank Russo, Sean Pecknold, Jérémie Voix, and Hugues Nelisse).
- Then, he presented nominees for executive positions (Christian Giguère for President, Dalila Giusti for Treasurer, Ramani Ramakrishnan for Editor, Bradford Gover for Executive Secretary)

In each case, there were no other nominations from the floor, so these nominees were declared elected by acclamation.

Adjournment

Adjournment was proposed by Rich Peppin and seconded by Roberto Racca. Carried. Meeting adjourned at 6:35 p.m.

Canadian Acoustical Association
Association canadienne d'acoustique

2010 PRIZE WINNERS / RÉCIPIENDAIRES 2010

BELL GRADUATE STUDENT PRIZE IN SPEECH COMMUNICATION AND HEARING /
PRIX ÉTUDIANT BELL EN COMMUNICATION VERBALE ET AUDITION

Nicolas Ellaham (University of Ottawa)
"Prediction of binaural speech intelligibility when using non-linear hearing aids"

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

Ben Biffard (University of Victoria)
"Acoustic Seabed Classification and Characterization by Single-Beam Echo Sounder"

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

Shira Daltrop (University of British Columbia)
"Factors affecting the acoustical performance of highway noise barriers"

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

Jill Lowther, Kayla Hack, Daniel McDonald, Kelly Sowden & Leigh Vanderloo (University of Western Ontario)
"Sorry, can you repeat that?: A health promotion campaign addressing noise-induced hearing problems among senior health sciences students"

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

Cody Shaw (Fort Richmond Collegiate, Ste. Agathe, Manitoba)
"Phononic Crystals: Revisited"

DIRECTORS' AWARDS / PRIX DES DIRECTEURS

Non-student / Non-étudiant:

Michael D. Hall (James Madison University)
"Clarifying spectral and temporal dimensions of musical instrument timbre"

Student / Étudiant:

Joana da Rocha, University of Victoria
"Prediction of flow-induced noise in transport vehicles:
Development and validation of acoupled structural-acoustic analytical framework"

STUDENT PRESENTATION AWARDS / PRIX POUR COMMUNICATIONS ÉTUDIANTES
NIAGARA-ON-THE-LAKE (ON), OCTOBER 14-16, 2009

Kostas Zolotas (University of Victoria)
"Lingual ultrasound of articulations made with the didgeridoo"
Sponsored by Scantek

Marianne Pelletier (University of Toronto at Mississauga)
"Effect of age on lexical decision speed when sentence context is acoustically distorted"
Sponsored by Kinetics Noise Control

Emma Murowinski (Defense Research and Development - Atlantic)
"Measurements and modelling of atmospheric acoustic propagation over water"
Sponsored by Xscala Sound & Vibration

CONGRATULATIONS / FÉLICITATIONS

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ALEXANDER G. BELL GRADUATE STUDENT PRIZE IN SPEECH COMMUNICATION AND HEARING
ECKEL GRADUATE STUDENT PRIZE IN NOISE CONTROL
FESSENDEN GRADUATE STUDENT PRIZE IN UNDERWATER ACOUSTICS
RAYMOND HETU UNDERGRADUATE STUDENT PRIZE IN ACOUSTICS

Prix

PRIX POST-DOCTORAL EDGAR ET MILLICENT SHAW EN ACOUSTIQUE
PRIX ETUDIANT ALEXANDER G. BELL EN COMMUNICATION ORALE ET AUDITION (2^{E} OU 3^{E} CYCLE)
PRIX ETUDIANT ECKEL EN CONTROLE DU BRUIT (2^{E} OU 3^{E} CYCLE)
PRIX ETUDIANT FESSENDEN EN ACOUSTIQUE SOUS-MARINE (2^{E} OU 3^{E} CYCLE)
PRIX ETUDIANT RAYMOND HETU EN ACOUSTIQUE (1^{ER} CYCLE)

Deadline for Applications:
April 30^{th} 2011

Date limite de soumission des demandes:
30 Avril 2011

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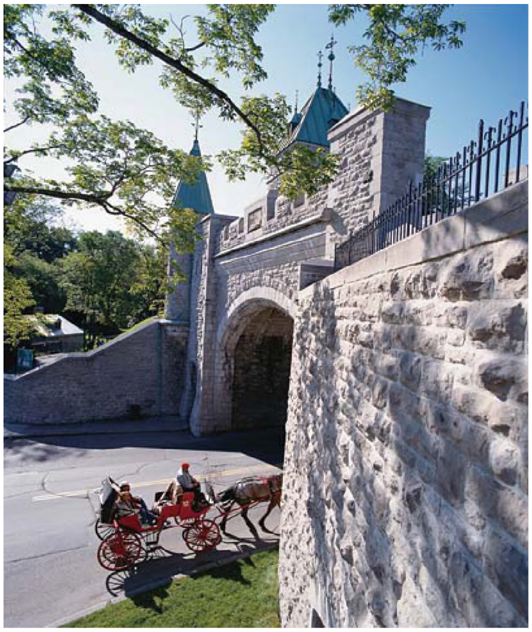

Saint-Louis Gate (Photo: Yves Tessier, Tessima)

— FIRST ANNOUNCEMENT —

ACOUSTICS WEEK IN CANADA

Quebec City, October 12-14, 2011

Acoustics Week in Canada 2011, the annual conference of the Canadian Acoustical Association, will be held in Quebec City from October 12 to 14. This premier Canadian symposium in acoustics and vibration will take place in beautiful Old Quebec, a UNESCO world heritage treasure with European appeal. You surely will not want to miss this event. The conference will include three days of plenary lectures and technical sessions on all areas of acoustics, a meeting of the Acoustical Standards Committee, the CAA Annual General Meeting, an Exhibition of acoustical equipment, materials and services, the Conference Banquet, an Award ceremony and other social events.

Plenary Lectures and Technical Sessions – Three plenary lectures are planned in areas of broad and relevant appeal to the acoustical community, highlighting the regional expertise and distinctiveness. Technical sessions will be organized in all major areas of acoustics, including the following topics:

- Architectural Acoustics
- Physical Acoustics and Ultrasound
- Psycho- and Physio-Acoustics
- Hearing and Speech Sciences
- Underwater Acoustics
- Bio-Acoustics and Biomedical Acoustics
- Engineering Acoustics and Noise Control
- Musical Acoustics and Electro acoustics
- Shock and Vibration
- Hearing Loss Prevention
- Signal Processing and Numerical Methods
- Acoustical Standards

If you would like to propose and/or organize a special session on a specific topic, you are invited to contact the Technical Co-Chairs as soon as possible.

Venue and Accommodation – Details will be provided on the second announcement.

Exhibition and Sponsorship – The conference will show case an exhibition of acoustical equipment, products and services on Thursday October 13, 2011. If you or your company are interested in participating in the Exhibition or in sponsoring conference social events, technical sessions, coffee breaks or student prizes, all of which being excellent promotional opportunities, please contact the Exhibition Coordinator.

Courses/Workshops – If you would like to offer a course/seminar in association with Acoustics Week in Canada, please contact the Conference Chair. Assistance can be provided in accommodating such an event, but it must be financially independent of the conference.

Student Participation – Student participation is strongly encouraged. Travel subsidies and reduced registration fees will be available. Student presenters are eligible to win prizes for the best presentations.

Paper Submission – The abstract deadline is June 15, 2011. The two-page summaries for publication in the proceedings issue of *Canadian Acoustics* are due by August 1st, 2011. Details will be given on the conference website.

Local Organizing Committee

Conference Chair:	Christian Giguère cgiguere@uottawa.ca
Technical Co-Chairs:	Jérémie Voix jeremie.voix@etsmtl.ca
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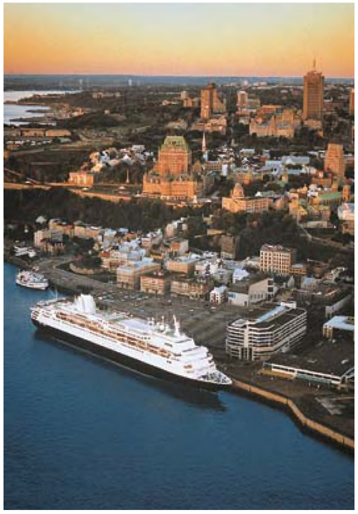

Vue aérienne du Vieux-Québec
(Photo: Yves Tessier, Tessima)

— PREMIÈRE ANNONCE —

SEMAINE CANADIENNE D'ACOUSTIQUE

Québec, 12 au 14 octobre 2011

La Semaine canadienne d'acoustique 2011, le congrès annuel de l'Association canadienne d'acoustique, se tiendra à Québec du 12 au 14 octobre prochain. Cet événement de premier plan dans le domaine de l'acoustique et des vibrations, tenu au cœur d'une ville si pittoresque et joyau du patrimoine mondial de l'UNESCO, en fera encore cette année un colloque à ne pas manquer. Il comprendra trois jours de séances plénières et sessions scientifiques, une réunion du Comité de normalisation en acoustique, l'Assemblée générale annuelle de l'ACA, une exposition d'équipement, produits et services en acoustique, un banquet, la remise annuelle des prix et d'autres activités sociales.

Séances plénières et sessions scientifiques – Trois présentations plénières dans des domaines d'intérêt général en acoustique sont prévues, mettant en évidence l'expertise régionale. Des sessions scientifiques seront organisées dans tous les domaines principaux de l'acoustique et des vibrations, dont les thèmes suivants:

- Acoustique architecturale
- Physique acoustique et Ultrasons
- Physio et Psychoacoustique
- Sciences de la parole et Audition
- Acoustique sous-marine
- Bioacoustique et Acoustique biomédicale
- Génie acoustique et Contrôle du bruit
- Acoustique musicale et Électroacoustique
- Chocs et Vibrations
- Prévention de la perte audition
- Traitement des signaux et Méthodes numériques
- Normalisation

Si vous désirez suggérer ou organiser une session spéciale, svp contactez le comité scientifique dès maintenant.

Lieu du congrès et Hébergement – Renseignements disponibles lors de la deuxième annonce.

Exposition technique et Commandite – Le congrès comprendra une exposition d'équipement, produits et services en acoustique le jeudi 13 octobre 2011. Si vous ou votre entreprise êtes intéressés à réserver un table pour cette exposition technique ou commanditer des événements sociaux, sessions scientifiques, pauses-cafés ou prix étudiants, lesquels présenteront tous d'excellentes occasions promotionnelles, veuillez communiquer avec le coordinateur de l'exposition technique.

Cours/Ateliers – Si vous souhaitez offrir un cours ou atelier dans le cadre de la Semaine canadienne d'acoustique, veuillez contacter le Président du congrès. Le comité de congrès vous prêtera assistance pour organiser votre événement, mais il doit être financièrement indépendant du congrès.

Participation étudiante – La participation étudiante est fortement encouragée. Des subventions de voyages et des frais d'inscription réduits seront offerts. Des prix seront décernés pour les meilleures présentations étudiantes lors du congrès.

Soumissions – La date d'échéance pour la soumission des résumés de présentation est le 15 juin 2011. Les articles de deux pages pour publication dans le numéro spécial des actes de congrès dans l'*Acoustique canadienne* sont dus le 1 août 2011. Plus de renseignements suivront sur le site internet de la conférence.

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The Canadian Acoustical Association / l'Association Canadienne d'Acoustique

MEMBERSHIP DIRECTORY 2010 / ANNUAIRE DES MEMBRES 2010

The number that follows each entry refers to the areas of interest as coded below.
Le nombre juxtaposé à chaque inscription réfère aux champs d'intérêt tels que condifés ci-dessous

Areas of interest		Champs d'intérêt
Architectural Acoustics	1	Acoustique architecturale
Engineering Acoustics / Noise Control	2	Génie acoustique / Contrôle du bruit
Physical Acoustics / Ultrasonics	3	Acoustique physique / Ultrasons
Musical Acoustics / Electro-acoustics	4	Acoustique musicale / Electroacoustique
Psycho- and Physio-acoustics	5	Psycho- et physio-acoustique
Shock and Vibration	6	Chocs et vibrations
Hearing Sciences	7	Audition
Speech Sciences	8	Parole
Underwater Acoustics	9	Acoustique sous-marine
Signal Processing / Numerical Methods	10	Traitement des signaux / Méthodes numériques
Other	11	Autre

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