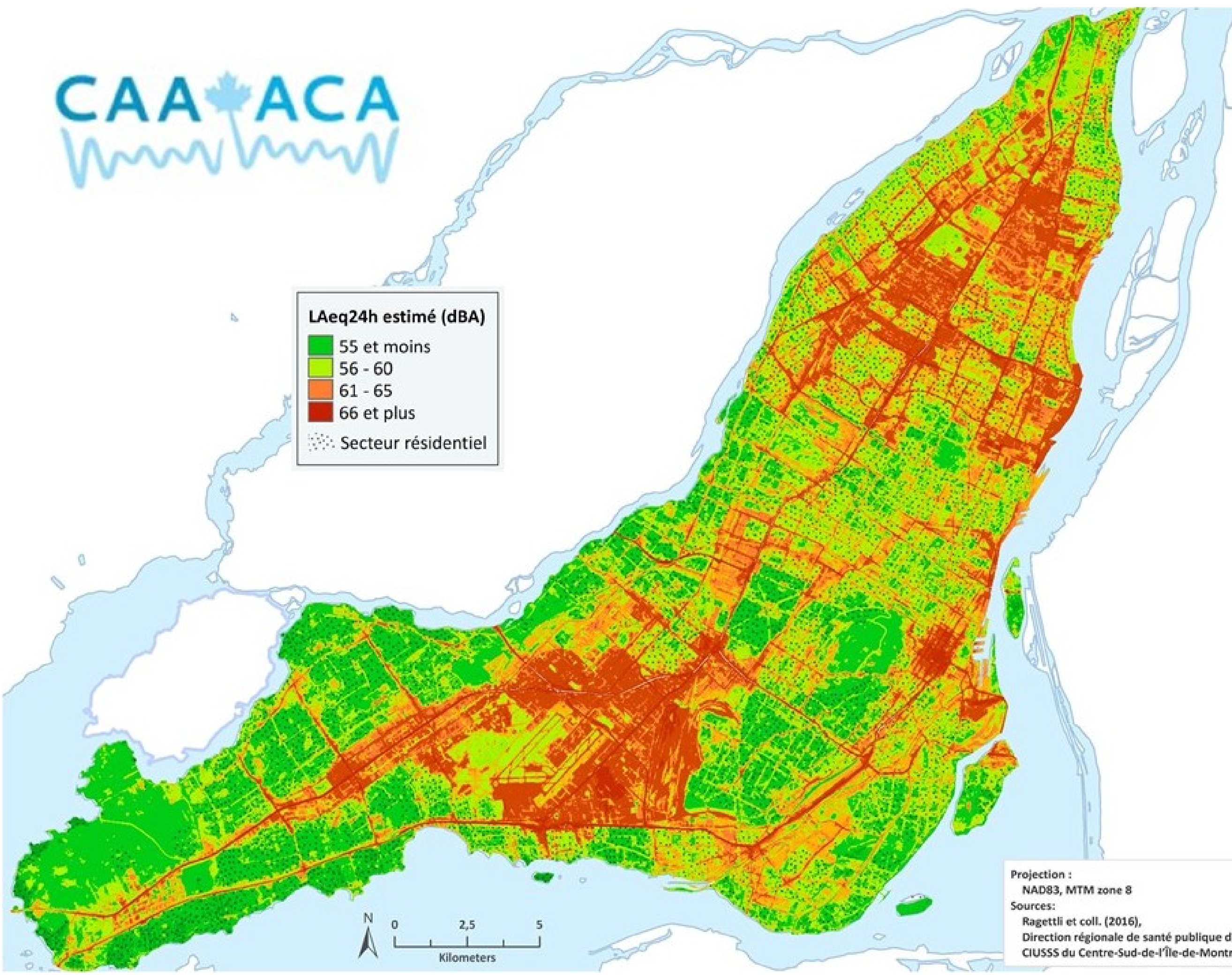
ISSN 0711-6659

canadian acoustics
acoustique canadienne

Journal of the Canadian Acoustical Association - Revue de l'Association canadienne d'acoustique

JUNE 2023 JUIN 2023
Volume 51 - - Number 2 Volume 51 - - Numéro 2



canadian acoustics

acoustique canadienne

Canadian Acoustical Association/Association Canadienne d'Acoustique P.B. 74068 Ottawa, Ontario, K1M 2H9

Association canadienne d'acoustique B.P. 74068 Ottawa, Ontario, K1M 2H9

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 only through the journal online submission system. Complete instructions to authors concerning the required "camera-ready" manuscript are provided within the journal online submission system.

L'**Acoustique Canadienne** publie des articles arbitrés et des informations sur tous les aspects de l'acoustique et des vibrations. Les informations portent sur la recherche, les ouvrages sous forme de revues, les nouvelles, l'emploi, les nouveaux produits, les activités, etc. Des articles concernant des résultats inédits ou des applications ainsi que les articles de synthèse ou d'initiation, en français ou en anglais, sont les bienvenus.

Canadian Acoustics is published four times a year - in March, June, September and December. This quarterly journal is free to individual members of the Canadian Acoustical Association (CAA) and institutional subscribers. **Canadian Acoustics** publishes refereed articles and news items on all aspects of acoustics and vibration. It also includes information on research, reviews, news, employment, new products, activities, discussions, etc. Papers reporting new results and applications, as well as review or tutorial papers and shorter research notes are welcomed, in English or in French. The Canadian Acoustical Association selected **Paypal** as its **preferred system** for the online payment of your subscription fees. Paypal supports a wide range of payment methods (Visa, Mastercard, Amex, Bank account, etc.) and does not requires you to have already an account with them. If you still want to proceed with a manual payment of your subscription fee, please Membership form and send it to the Executive Secretary of the Association (see address above). - - Dr. Roberto Racca - Canadian Acoustical Association/Association Canadienne d'Acoustique c/o JASCO Applied Sciences 2305–4464 Markham Street Victoria, BC V8Z 7X8 - - secretary@caa-aca.ca

Acoustique canadienne est publié quantre fois par an, en mars, juin, septembre et décembre. Cette revue trimestrielle est envoyée gratuitement aux membres individuels de l'Association canadienne d'acoustique (ACA) et aux abonnés institutionnels. L'**Acoustique canadienne** publie des articles arbitrés et des rubriques sur tous les aspects de l'acoustique et des vibrations. Ceci comprend la recherche, les recensions des travaux, les nouvelles, les offres d'emploi, les nouveaux produits, les activités, etc. Les articles concernant les résultats inédits ou les applications de l'acoustique ainsi que les articles de synthèse, les tutoriels et les exposées techniques, en français ou en anglais, sont les bienvenus. L'Association canadienne d'acoustique a sélectionné **Paypal** comme solution pratique pour le paiement en ligne de vos frais d'abonnement. Paypal prend en charge un large éventail de méthodes de paiement (Visa, Mastercard, Amex, compte bancaire, etc) et ne nécessite pas que vous ayez déjà un compte avec eux. Si vous désirez procéder à un paiement par chèque de votre abonnement, merci de remplir le formulaire d'inscription et de l'envoyer au secrétaire exécutif de l'association (voir adresse ci-dessus). - - Dr. Roberto Racca - Canadian Acoustical Association/Association Canadienne d'Acoustique c/o JASCO Applied Sciences 2305–4464 Markham Street Victoria, BC V8Z 7X8 - - secretary@caa-aca.ca

Contents - Table des matières


CONTROL
NOISE
LOWER PROJECT COSTS
IMPROVE SPEECH PRIVACY
BOOST COMFORT
& WELLNESS
QUICK ROI
INCREASE PRODUCTIVITY
FACILITY
FLEXIBILITY
ENHANCE WORKPLACE CULTURE
SUPPORT FOCUS
LogiSon ACOUSTIC NETWORK
SOUND.
THAT WORKS.™
Sound masking is more than a product. It's a service provided by professional technicians who know the effect isn't achieved from the moment they power the system, but by tuning the sound to an independently-proven curve. Designed right, tuned right—that's our motto. And the result is more consistent, comfortable and effective sound masking.
www.logison.com
© 2022 KR MOELLER ASSOCIATES LTD. LOGISON IS A REGISTERED TRADEMARK OF 777388 ONTARIO LIMITED. PHOTO BY VINCENT LIONS.

Editor's note: The Last Dance
Éditorial : La dernière danse

Canadian Acoustics for our society

Dear reader, writing this editorial provides me with the opportunity to thank you for your continuous support over the last seven years, I have been Editor in Chief of Canadian Acoustics journal (JCAA).

At a time when digital information and online meetings are everything, our association, firmly believes of the importance of continuing hard printing and in person conferences, as we are a community of people, passionate professionals who like the "old way" of things.

We are ready to have soon another successful Acoustics Week in Canada in the beautiful Montréal from October 3rd to 6th. Our annual event will be followed by a variety of opportunities in 2024; just to mention a few, from 13 to 17 May 2024, we will co-organize in Ottawa our national conference in conjunction with the 186th Meeting of the Acoustical Society of America. Moreover, we are working to bring other major international events in 2028 (stay tuned!). In brief, despite the challenges of a pandemic and the impact it had on little societies, like ours, we keep the momentum of volunteering to support the growth of the Science of Sound in Canada!

Now, let me present this summer 2023 issue. You will find a numerical paper on metamaterials, a promising field of noise control tool that in this paper are studied performing a finite element analysis of honeycomb membrane-type acoustic metamaterial. Moreover, in the issue, we will present a recent study of the Assessment of noise in the campus of École de Technologie Supérieure in Montréal, still in our engineering acoustics section. You will immediately see the reason for the cover image taken within Ragettli et al, 2016 report Finally, the issue in enriched with a Comparison of Various Algorithms: Research on Piano Audio Signal Feature Identification, so a signal processing paper.

One last personal note. This is probably my last editorial as Editor in Chief of Canadian Acoustics. I want to thank you once more, not only the readers, but also the contributors and all the patient authors who continuously accept the challenge of presenting their work to the attention of colleagues to be judged, in an effort that aims only to push the science and knowledge in our field. Thank you!

Umberto Berardi
Editor in Chief.

Acoustique canadienne pour notre société

Chères lectrices et chers lecteurs, la rédaction de cet éditorial me donne l'occasion de vous remercier pour votre soutien continu au cours des sept dernières années, en tant que rédacteur en chef de la revue Acoustique canadienne (JCAA).

À une époque où l'information numérique et les réunions en ligne sont primordiales, notre association croit fermement à l'importance de poursuivre l'impression papier et les conférences en présentiel, car nous sommes une communauté de personnes, de professionnels passionnés, qui aiment « l'ancienne méthode ».

Nous sommes prêts à recevoir une autre édition de la Semaine canadienne de l'acoustique dans la belle ville de Montréal du 3 au 6 octobre. Notre événement annuel sera suivi d'une variété d'opportunités en 2024. Pour n'en citer que quelques-uns : du 13 au 17 mai 2024 nous coorganiserons à Ottawa notre conférence nationale en partenariat avec la 186e réunion de l'Acoustical Society of America. De plus, nous travaillons pour amener d'autres grands événements internationaux en 2028 (restez à l'écoute !). Bref, malgré les défis de la pandémie et l'impact qu'elle a eu sur de petites sociétés, comme la nôtre, nous maintenons l'élan du bénévolat pour soutenir la croissance de la science du son au Canada !

Maintenant, permettez-moi de vous présenter ce numéro. Vous trouverez un article sur l'utilisation de méthodes numériques appliquées aux métamatériaux, un domaine prometteur pour l'utilisation d'outil de contrôle du bruit qui, dans cet article, est étudié en effectuant une analyse par éléments finis du métamatériau acoustique de type membrane en nid d'abeille. De plus, dans le numéro, nous présenterons une étude récente sur l'Évaluation du bruit au campus de l'École de technologie supérieure de Montréal, toujours dans notre section Génie acoustique. Vous comprendrez instantanément le lien avec l'illustration de couverture tirée de la publication Ragettli et collab., 2016 Enfin, le numéro est enrichi par la « Comparaison de divers algorithmes : recherche sur l'identification des caractéristiques du signal audio du piano », donc un article orienté traitement du signal.

Une dernière note personnelle. Ceci est probablement mon dernier éditorial en tant que rédacteur en chef de l'Acoustique canadienne. Je tiens à vous remercier une fois de plus, non seulement les lecteurs, mais aussi les contributeurs et tous les auteurs patients qui acceptent continuellement le défi de présenter leurs travaux à l'attention de collègues pour être jugés, dans un effort qui ne vise qu'à pousser la science et connaissances dans notre domaine. Merci!

Umberto Berardi
Rédacteur en chef

FINITE ELEMENT ANALYSIS OF HONEYCOMB MEMBRANE-TYPE ACOUSTIC METAMATERIAL

Zacharie Laly *1, 2, Christopher Mechefske †2, Sebastian Ghinet ‡3, Behnam Ashrafi ♦4, and Charly T. Kone #3

[1]CRASH, Centre de Recherche Acoustique-Signal-Humain, Université de Sherbrooke, Québec, Canada.
[2]Department of Mechanical and Materials Engineering, Queen's University, Kingston, Ontario, Canada.
[3]Aerospace, National Research Council Canada, Ottawa, Ontario, Canada.
[4]Aerospace Manufacturing Technology Center, National Research Council Canada, Montreal, Québec, Canada.

Résumé

Dans cet article, un métamatériau acoustique constitué d'une structure en nid d'abeille avec des couches de membrane intégrées est étudié à l'aide de la méthode des éléments finis. Ce matériau léger présente une excellente perte par transmission en basse fréquence. La structure en nid d'abeille avec deux et trois couches de membranes intégrées est analysée numériquement et les effets du nombre de couches de membranes et de l'épaisseur du gap d'air entre les membranes sont illustrés. Les influences des propriétés de matériau et de l'épaisseur de la membrane sur la perte par transmission et l'amplitude et la forme du mode du déplacement à différentes fréquences sont présentées. Il est montré que la perte par transmission augmente sur une large bande fréquentielle lorsque la taille des cellules du nid d'abeille diminue tandis que l'amplitude du déplacement de la membrane est réduite et la forme du mode du déplacement est affectée. Il est observé que la perte par transmission présente de multiples pics de résonance au fur et à mesure que l'épaisseur de la membrane est réduite. Une amélioration de la perte par transmission est observée autour des fréquences anti-résonnantes en augmentant le facteur d'amortissement de la membrane, ce qui provoque une réduction des pics de résonance du déplacement et de la perte par transmission. Le métamatériau étudié peut être utile dans de nombreuses applications d'ingénierie de contrôle du bruit.

Mots clefs : Métamatériau acoustique, structure en nid d'abeille, membrane, perte par transmission, matériau léger.

Abstract

In this paper, a honeycomb membrane-type acoustic metamaterial made of a honeycomb structure with embedded membrane layers is investigated using the finite element method. This lightweight material presents excellent transmission loss (TL) at low frequency. Honeycomb structures with two and three embedded membrane layers are analyzed numerically and the effects of the number of membrane layers and of the thickness of the air gap between membranes are illustrated. Also, the influences of the membrane material properties and thickness on the TL and displacement magnitude and mode shape at different frequencies are presented. It is shown that the TL increases over a large frequency band when the honeycomb cell size decreases while the displacement magnitude of the membrane is reduced and the mode shape is affected. It is observed that the TL presents multiple resonant peaks as the thickness of the membranes is reduced. An improvement of the TL is observed around the anti-resonant frequencies by increasing the damping loss factor of the membrane, which causes a reduction of the resonant displacement magnitude and TL peaks amplitude. The investigated metamaterial can be useful in many noise control engineering applications.

Keywords: Acoustic metamaterial, honeycomb structure, membrane, transmission loss, lightweight material.

1 Introduction

Lightweight materials are often desirable and used in many applications such as in the aerospace and automotive industries. One critical shortcoming of these lightweight materials is their poor sound transmission loss (TL) because of the mass law, which states that the noise transmission trough a material is inversely proportional to the surface mass of the panel and the frequency. This requires a large thickness of lightweight materials to achieve low-frequency acoustic attenuation. An optimal design of lightweight material with a high TL would therefore be interesting. Li et al. [1] investigated experimentally the TL of a lightweight multilayer honeycomb membrane-type acoustic metamaterial (MAM). They illustrated a significant sound insulation performance of the proposed structures by comparison with the traditional honeycomb structure with minimal mass increase. Sui et al. [2] and Lu et al. [3] studied a honeycomb acoustic metamaterial made of a lightweight flexible rubber material layer sandwiched between two layers of honeycomb cell plates. The structure presented negative mass density at frequencies below the first natural frequency, which results in excellent TL with minimum weight-penalty. Nguyen et al. [4] presented a double-

* zacharie.laly@usherbrooke.ca
† chris.mechefske@queensu.ca
‡ sebastian.ghinet@nrc-cnrc.gc.ca
♦ behnam.ashrafi@nrc-cnrc.gc.ca
\# tenoncharly.kone@nrc-cnrc.gc.ca

layer membrane-type acoustic metamaterial (DMAM) using theoretical and numerical analysis. They performed experimental tests on an acoustic panel made of an 8 x 8 DMAM array and showed that the transmission loss ranges from 40 dB to 59 dB in the frequency band 0.45–1.48 kHz, breaking the mass density law. Yang et al. [5] investigated lightweight membrane-type metamaterials made of a circular elastic membrane with a small weight attached at the center with fixed outer boundary. A negative effective mass was obtained at low frequency and the sound insulation was far higher than the one predicted by the mass law. Li et al. [6] proposed a theoretical method to predict the transmission loss of acoustic micro-membranes and showed significant enhancement of the sound insulation at low frequency. The influences on the TL of the membrane geometrical parameters such as the thickness and the surface area are illustrated in their study. Qiu et al. [7] proposed a theoretical method to study an array of membranes mounted in a lattice-like frame. They investigated the influence of the frame vibration and showed how the frames affect the vibration and transmission loss characteristics of the membranes. They derived an analytical expression of the TL by coupling the vibration of the frame and the membranes where the frame and membranes are considered in parallel. It was observed that the global response of the assembly is no longer dominated by the membranes but by the frame as the assembly becomes larger. Finite element method was used by Laly et al. [8] to model a honeycomb structure with embedded membrane layers. An improvement of the transmission loss at low frequency was illustrated. Gao et al. [9] investigated numerically and experimentally the low frequency sound insulation performance of a deformable honeycomb acoustic metamaterial made of honeycomb structures with embedded ethylene-vinyl acetate (EVA) rubber material. They observed that the investigated metamaterial has a better sound insulation than traditional sound insulation structures and its transmission loss can be adjusted by the dislocation, compression, and tensile deformation. Hu et al. [10] studied a type of membrane-sandwich plate metamaterial constituted by sandwich panels with embedded membranes with attached masses. They presented the sound insulation properties for different masses and illustrated the influence of the mass size on the TL. Their experimental analysis showed excellent low frequency sound insulation performance. Ma et al. [11] reported numerical and experimental analysis on a type of 2D multiple cells lumped ultrathin lightweight plate-type acoustic metamaterials structures, which illustrates good TL in the low frequency range. The basic unit cell of the metamaterials was made of an ultrathin stiff nylon plate clamped by two elastic ethylene-vinyl acetate copolymer. Marinova et al. [12] modelled a low frequency noise shield made of a double wall system, which includes two membrane-type acoustic metamaterials panels in the enclosed cavity. The MAM panel comprises a thin elastic mass-loaded membrane and a supporting frame. They validated the numerical simulations with experimental measurements. Zhou et al. [13] presented a method to broaden the low frequency bandwidth of sound insulation by designing a flexible membrane-type acoustic metamaterial sample that is made of a homogenous membrane and a perforated EVA copolymer plate. They created holes of different diameters in the flexible EVA plate and showed that the lower limit, the peak frequency and the upper limit of the TL bandwidth can be regulated by the lumped coupling resonance, anti-resonance and local resonance modes of the relevant areas. They proposed two MAM samples and the results of the TL illustrated improved bandwidths. Mo et al. [14] investigated a structure of acoustic micro membrane metamaterial where each micro membrane is a small square held within a larger lattice frame. Using theoretical and numerical simulations that include the vibration of the frame, they showed how the global TL of the assembly is related to the geometrical parameters of the micro membrane cells and the lattice. They noted that the frame should be strong enough to provide rigid boundaries to the micro membranes in order to improve the TL. Langfeldt et al. [15] presented an analytical method to predict the TL of baffled panels with multiple subwavelength sized membrane-type acoustic metamaterial unit cells using effective surface mass density concept. They studied the influence of flexible MAM unit cell edges and observed that the compliance of the edges has small impact on the TL except in the stiffness-controlled regime.

The development of lightweight materials with good transmission loss at low frequency is a challenge.

The work presented here uses the finite element method to investigate a honeycomb membrane type acoustic metamaterial constituted by a honeycomb structure with embedded membrane layers. This analysis considers a lightweight material design with a high sound attenuation efficiency. The transmission loss of the metamaterial is significantly improved especially at low frequency while the TL of the honeycomb core structure alone is zero over the entire frequency range. The influence of the membrane material properties on the TL is analyzed. By reducing the thickness of the membrane, the displacement magnitude of the membrane and the TL show multiple resonant peaks. The impacts of the honeycomb cell size on the TL and the membrane displacement magnitude and mode shape are studied numerically as well as the effect of the membrane damping loss factor. The honeycomb with two and three integrated membrane layers is studied numerically and the influences of the number of membrane layers as well as the impact of the air gap between the membranes are demonstrated. The numerical analysis is described in section 2. In section 3, the results of different parametric analysis are shown and finally section 4 presents the numerical investigations on honeycomb structure with multiple embedded membrane layers.

2 Honeycomb structure with embedded membrane design

Honeycomb structures used in many applications are lightweight and offer good stiffness. In general, lightweight structures are not efficient in acoustic attenuation especially at low frequencies and this is due to the mass law. The integration of a membrane within honeycomb structures can present both the characteristic of lightness and good acoustic attenuation. In this paper, numerical analysis is performed on honeycomb structures with embedded membranes for transmission loss improvement. This is a honeycomb membrane-type acoustic metamaterial where the impacts of the honeycomb cell size and the membrane properties on the transmission loss are analyzed using finite element method. In Fig. 1, a honeycomb structure with one embedded membrane layer is illustrated. The numerical simulations are conducted using acoustic-solid interaction of COMSOL Multiphysics to illustrate the contribution of the membrane in the transmission loss improvement. The membrane and the two honeycomb structures in Fig. 1 have the same lateral dimensions of 84 mm x 75 mm. An incident fluid and a transmission fluid are connected to the structure and all the air domains within the honeycomb cells are identified. The boundary between the membrane surface and each honeycomb structure layer is considered fixed in the numerical simulations. The geometry is shown in Fig. 1(a) and the mesh is presented in Fig. 1(b) which is a physics-controlled mesh with 144590 domain elements and 39707 boundary elements where the total degrees of freedom is 261100.

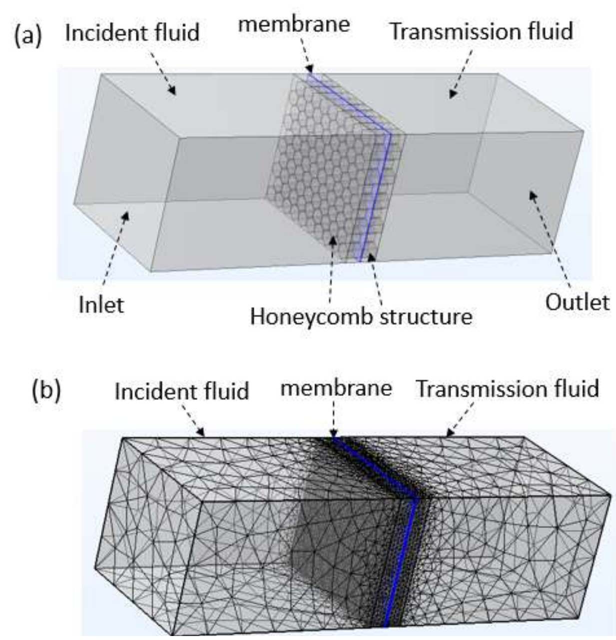


Figure 1: Numerical design of honeycomb structure with embedded membrane: (a) geometry (b) mesh.

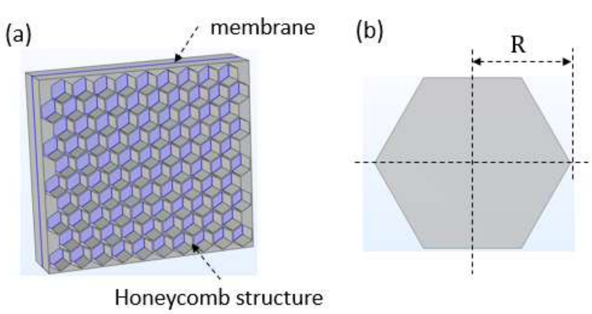


Figure 2: Honeycomb structure with embedded membrane: (a) honeycomb structure geometry (b) one honeycomb cell geometry.

Figure 2(a) shows the membrane that is sandwiched between two honeycomb structure layers and the geometry of one honeycomb cell is presented in Fig. 2(b). The dimension of the cell size is denoted by R and the thickness of each honeycomb structure layer is set to 10 mm.

The membrane is modeled as a linear isotropic elastic material using the solid mechanics module of COMSOL Multiphysics [8, 16, 17]. At the inlet and outlet planes as shown in Fig. 1, the plane wave radiation condition is applied to minimize the reflection of the acoustic waves. The material properties of the honeycomb structure namely the Young's modulus, the density, and the Poisson's ratio are 2.7 GPa, 1100 kg/m^3 and 0.38 respectively. The incident fluid and transmission fluid have the same length of 130 mm. A normal incidence plane wave with pressure amplitude of 1 Pa is applied on the inlet plane and the transmission loss is calculated numerically by the following relation [16-19]

$$TL = 10 \log_{10}\left(\frac{W_{in}}{W_{out}}\right), \tag{1}$$

where W_{in} and W_{out} are respectively the incoming power at the inlet plane and the outgoing power at the outlet plane given by

$$W_{in} = \int_{\partial\Omega} \frac{|p_0|^2}{2\rho c}, \qquad W_{out} = \int_{\partial\Omega} \frac{|p|^2}{2\rho c} \tag{2}$$

where c is the speed of sound in air, ρ is the density of air, and p_0 and p are the pressures at the inlet and outlet planes, respectively.

3 Finite element analysis results of honeycomb structures with embedded membrane

3.1 Numerical analysis of the effects of the membrane material properties

The geometry and mesh presented in Fig. 1 are used for the numerical analysis to evaluate the influence of the membrane material properties on the transmission loss. The honeycomb cell size R is set to 4.5 mm and the thickness of the membrane is 1 mm with a damping loss factor of zero. Different membrane material properties with Young's modulus that are gradually increased from 1.2 MPa to 100 MPa are considered. Table 1 summarizes the material properties of each membrane. Membrane 1 is a butyl rubber while membrane 3 is a silicone elastomer and membranes 2 and 4 are ethylene-vinyl acetate rubbers. Membrane 5 is a rubber material.

Table 1: Material properties of the membrane.

Membranes	Young's modulus (MPa)	Density (kg/m^3)	Poisson's ratio
1	1.2	910	0.4
2	5	660	0.45
3	12	1400	0.48
4	25	850	0.46
5	100	1100	0.49

The boundary between each membrane and each honeycomb structure layer is fixed. The transmission loss obtained using Eq. (1) for each membrane material in Table 1 is shown in Fig. 3. For the TL without a membrane within the honey-

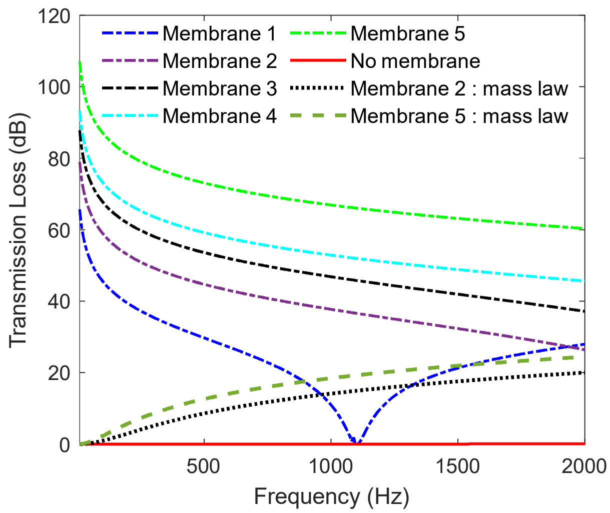


Figure 3: Effect of the membrane material properties on the transmission loss.

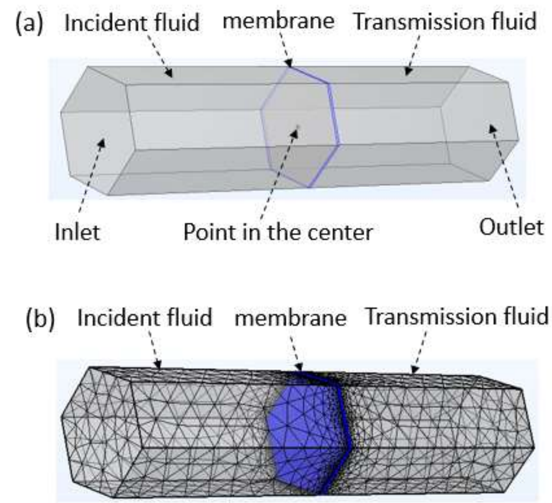


Figure 4: Honeycomb cell used for numerical analysis: (a) geometry (b) mesh.

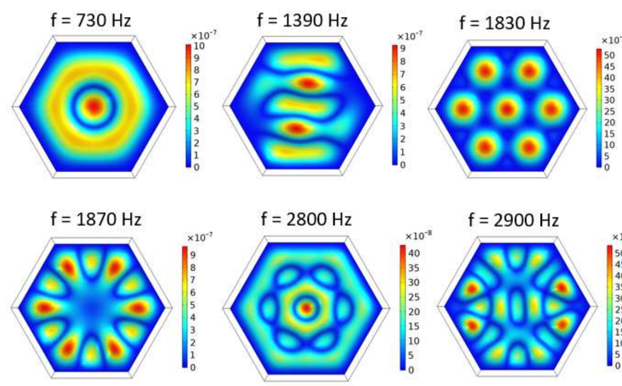


Figure 5: Displacement mode shapes of the membrane 2 for a thickness of 0.1 mm.

comb structures (Fig. 3), the cell size is 4.5 mm and the thickness of the honeycomb is set to 20 mm. The TL predicted by the mass law is illustrated for membrane 2 and 5.

For the honeycomb structure alone without membrane, the transmission loss in Fig. 3 is zero over the entire frequency range. It does not attenuate any noise even though its stiffness and lightness can be interesting. The transmission loss with membrane 1 in Fig. 3 shows a drop at the frequency f=1110 Hz where the TL value is zero and below 500 Hz the TL is greater than 30 dB. With membrane 2, the TL is greater than 35 dB for frequencies lower than 1000 Hz while with membrane 5, the TL is over 65 dB for f<1000 Hz. When one considers a membrane with an increased Young's modulus, the TL increases over the entire frequency range. From membranes 2 to 5, the TL in Fig. 3 presents a large frequency band and is higher than the TL obtained by the mass law.

3.2 Effect of the membrane thickness

The influence of the membrane thickness t is investigated in the following where the numerical analysis was conducted using one honeycomb cell as illustrated in Fig. 4 with membrane fixed boundary conditions. The membrane is connected to an incident and transmission fluid. A plane wave radiation condition is applied on the inlet and outlet plans and a pressure of 1 Pa is applied on the inlet plane and the transmission loss is calculated using Eq. (1). The lateral boundary of the incident and transmission fluid are considered rigid and the point in the center of the membrane on the incident fluid side is located to extract numerically its displacement magnitude. The membrane is modeled using a solid mechanics module while the incident and transmission fluid domains are modeled with a pressure acoustics module of COMSOL and the boundaries between these two physics are the two membrane surfaces.

The membrane 2 of Table 1 is used and the honeycomb cell size R is set to 6 mm. The length of the incident and transmission fluid are both equal to 24 mm. The impact of the membrane thickness on its displacement mode shapes is presented in Figs. 5 and 6 for some frequencies. In Fig. 5, the thickness of the membrane is set to 0.1 mm while in Fig. 6, the thickness is 0.25 mm. The displacement magnitude of the point in the center of the membrane (Fig. 4(a)) for different membrane thicknesses is illustrated in Fig. 7.

For each frequency, the membrane displacement mode shape changes in Figs. 5-6 when the thickness changes. With a thickness of 0.1 mm, one observes that the mode shapes in Fig. 5 are more complex than those in Fig. 6. The displacement magnitude of the point in the center of the membrane presented in Fig. 7 shows two peaks for a thickness of 0.1 mm at 555 Hz and 1880 Hz where the peak values are respectively 13.4 µm and 11.4 µm. For a thickness of 0.25 mm, the displacement values are 1.55 µm and 0.18 µm at 555 Hz and 1880 Hz respectively. With a thickness of 1 mm, the displacement tends towards zero. Thus, when the thickness of the membrane increases, the displacement magnitude decreases.

The impact of the membrane thickness on the transmission loss is shown in Fig. 8.

For a thickness of 1 mm, the transmission loss in Fig. 8 is greater than 25 dB with a wide attenuation band for frequencies below 1000 Hz and decreases when the thickness decreases. For t = 0.5 mm, the TL presents a drop around

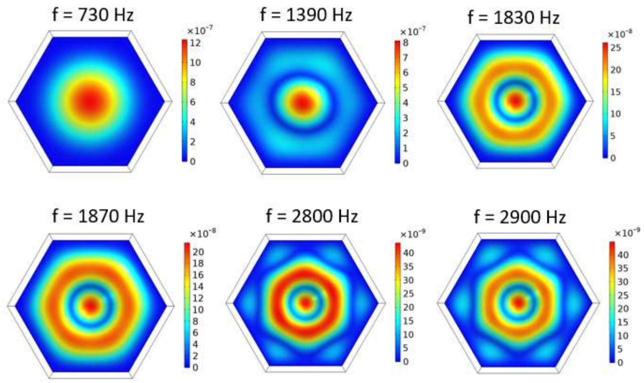


Figure 6: Displacement mode shapes of the membrane 2 for a thickness of 0.25 mm.

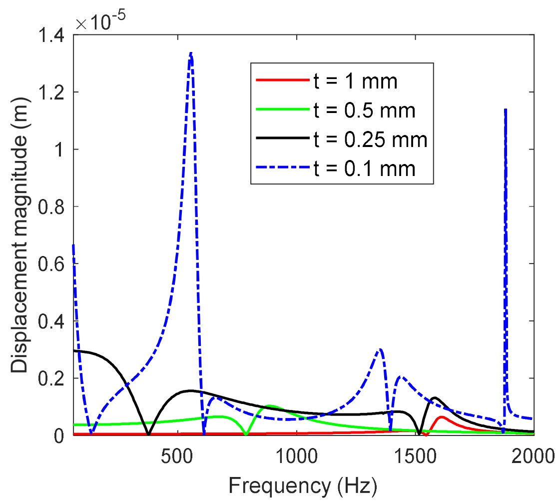


Figure 7: Effect of the membrane thickness on the displacement magnitude.

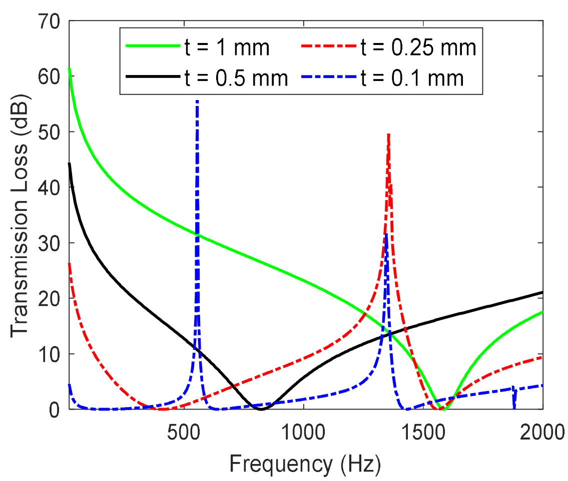


Figure 8: Effect of the membrane thickness on the transmission loss.

825 Hz where its value is zero. With a thickness of 0.25 mm, one observes a resonance peak at 1355 Hz where the TL value is 49 dB. Two TL resonance peaks are observed for a thickness of 0.1 mm at 555 Hz and 1345 Hz where the TL values are 55 dB and 32 dB respectively. Apart from these two resonant frequencies, the TL for t=0.1 mm is low and close to zero for f<450 Hz. Thus, when the thickness of the membrane increases, the TL is improved at low frequency with an increased attenuation frequency band.

3.3 Finite element analysis of the influence of the honeycomb cell size

The effect of the honeycomb cell size is analyzed numerically in this section using the geometry that is illustrated in Fig. 4. Membrane 2 of Table 1 with a thickness of 0.5 mm is considered with fixed boundary conditions and the cell size R is varied. The incident and transmission fluid have the same length, which is equal to 4R. Figure 9 shows the transmission loss for different honeycomb cell sizes. In Fig. 10, the displacement magnitude of the point in the center of the membrane (Fig. 4) is illustrated.

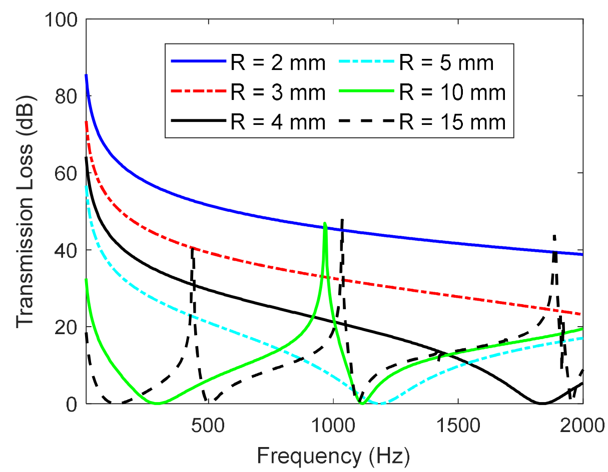


Figure 9: Effect of the cell size on the transmission loss for the membrane thickness of 0.5 mm.

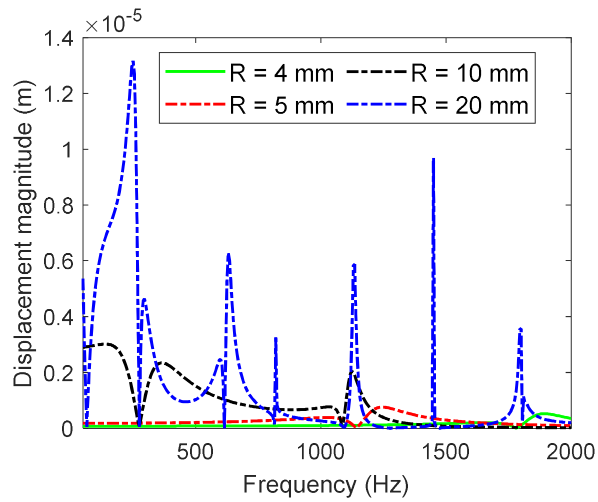


Figure 10: Effect of the cell size on the displacement magnitude for the membrane thickness of 0.5 mm.

The transmission loss in Fig. 9 is greater than 38 dB over the entire frequency range for R=2 mm and for R=3 mm, it is greater than 23 dB and increases when the frequency decreases. The TL drop for R=4 mm and R=5 mm is observed around 1845 Hz and 1190 Hz respectively and for R=10 mm, the TL presents a resonant peak at 970 Hz with value of 46 dB. With a cell dimension of 15 mm, the TL exhibits 3 resonance peaks at 435 Hz, 1035 Hz and 1885 Hz where the TL values are 40 dB, 48 dB and 44 dB respectively. As the honeycomb cell size is reduced, the TL increases over the entire frequency range. The displacement magnitude in Fig. 10 of the point at the center of the membrane presents several resonance peaks for R=20 mm. When the honeycomb cell size decreases, the resonant peaks disappear and the amplitude of the displacement tends towards zero. The reduction of the honeycomb cell size leads to a reduction in the displacement amplitude, but the TL increases over the entire frequency band.

The influence of the honeycomb cell size on the displacement mode shapes of the membrane is illustrated in Figs. 11-13. These show the displacement mode shapes for honeycomb cell size R of 10 mm 15 mm and 20 mm respectively. The thickness of the membrane is set to 0.25 mm with fixed boundary conditions.

As the dimension of the honeycomb cell increases, the displacement mode shapes in Figs. 11-13 change at each frequency while the displacement amplitude presents more peaks in Fig. 10 as well as the TL in Fig. 9.

3.4 Analysis of the membrane damping loss factor

The impact of the membrane damping loss factor on the displacement and transmission loss were also investigated. One honeycomb cell as shown in Fig. 4 is considered with size R of 8 mm. The membrane 4 of Table 1 is used with a thickness of 1 mm and its boundary conditions are fixed. The isotropic loss factor η of the membrane is varied from zero to 40% and the transmission loss is presented in Fig. 14. Figure 15 shows the average surface displacement magnitude of the membrane for different loss factors.

In Fig. 14, one observes a drop in the TL for η=0 at 1850 Hz where the TL value is zero and the displacement at this frequency in Fig. 15 presents a peak with a value of 0.21 μm. When the loss factor η increases, the transmission loss around 1850 Hz increases and the peak of the displacement magnitude in Fig. 15 decreases. For η=0.1, the TL value at 1850 Hz is 10.56 dB in Fig. 14 and the displacement amplitude value is 0.063 μm. For η =0.4, the TL value at 1850 Hz is 20 dB while the peak of the displacement is 0.02 μm. Between 1655 Hz and 2060 Hz, the TL is improved when the loss factor η increases while the peak of the displacement magnitude is reduced.

One considers now the membrane 2 of Table 1 with fixed boundary conditions and the dimension of the honeycomb cell size R is set to 8 mm. In Figs. 16-17, the transmission loss is presented for different damping loss factors. In Fig. 16, the membrane thickness is set to 1 mm while in Fig. 17, the thickness is 0.25 mm.

The influence of the damping loss factor on the displacement magnitude and displacement mode shape at different frequencies is illustrated in Figs. 18-20 for a membrane thickness of 0.25 mm.

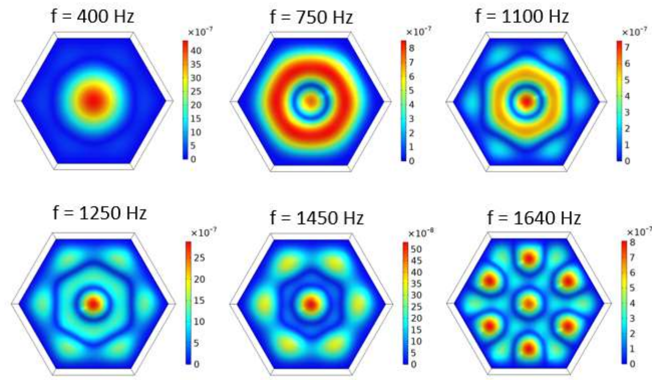


Figure 11: Displacement mode shape of the membrane for a cell size of 10 mm.

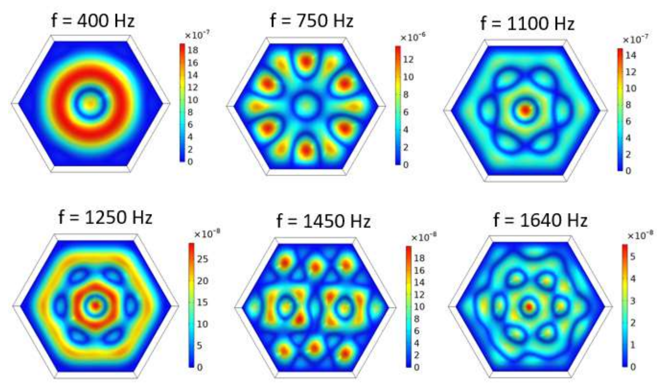


Figure 12: Displacement mode shape of the membrane for a cell size of 15 mm.

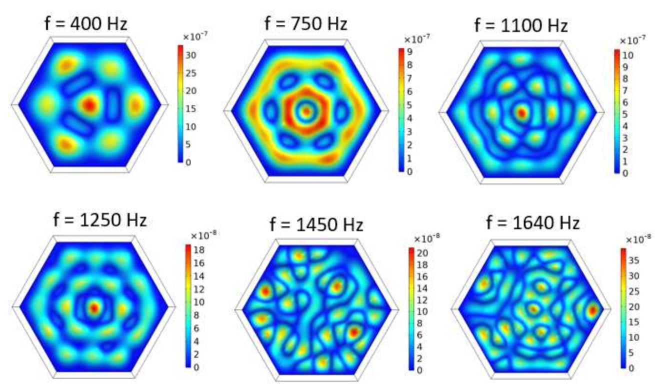


Figure 13: Displacement mode shape of the membrane for a cell size of 20 mm.

The displacement magnitude peaks in Fig. 18 decrease as the damping factor increases. In Fig. 16, the TL is improved around the anti-resonance frequencies of 930 Hz and 3225 Hz when the damping loss factor increases, on the other hand the resonance TL peak at 2760 Hz decreases. In Fig. 17, one observes also that the resonance TL peaks are reduced when the damping loss factor increases and the displacement modes in Figs. 19-20 are influenced.

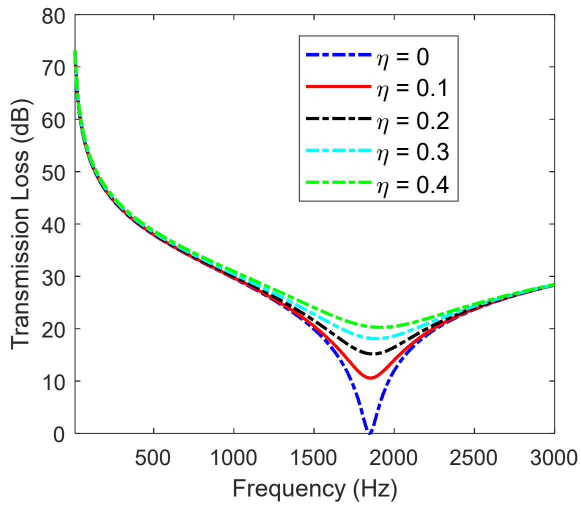


Figure 14: Effect of the membrane damping loss factor on the transmission loss.

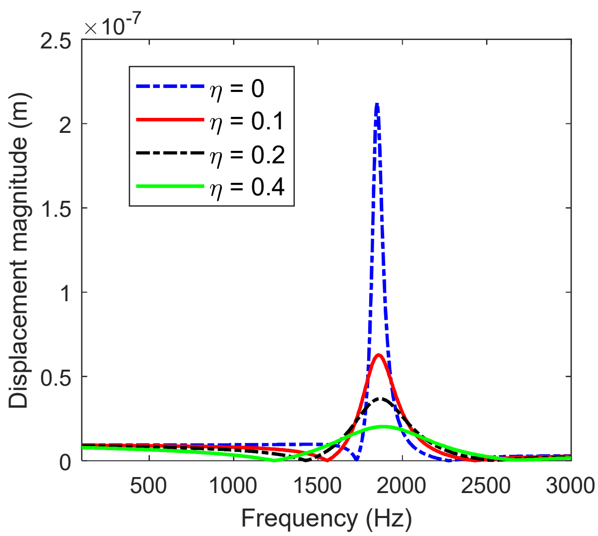


Figure 15: Effect of the membrane damping loss factor on the displacement magnitude.

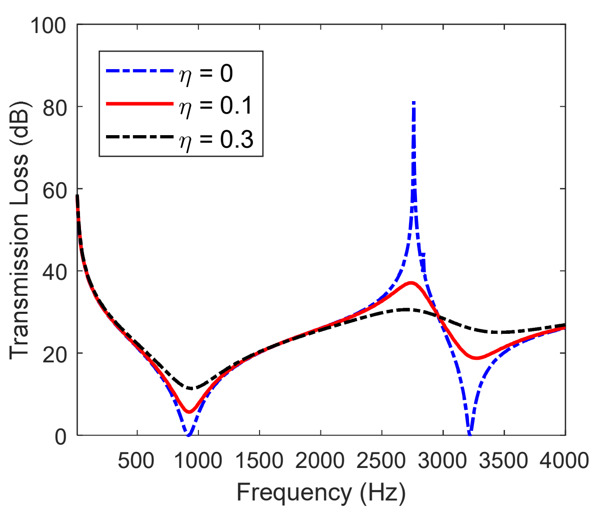


Figure 16: Effect of the membrane damping loss factor on the transmission loss for a membrane thickness of 1 mm.

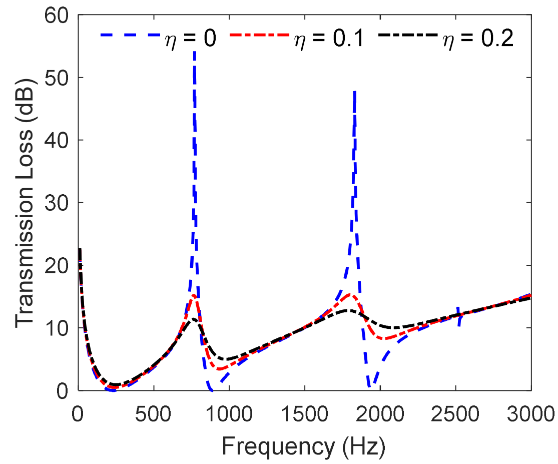


Figure 17: Effect of the membrane damping loss factor on the transmission loss for a membrane thickness of 0.25 mm.

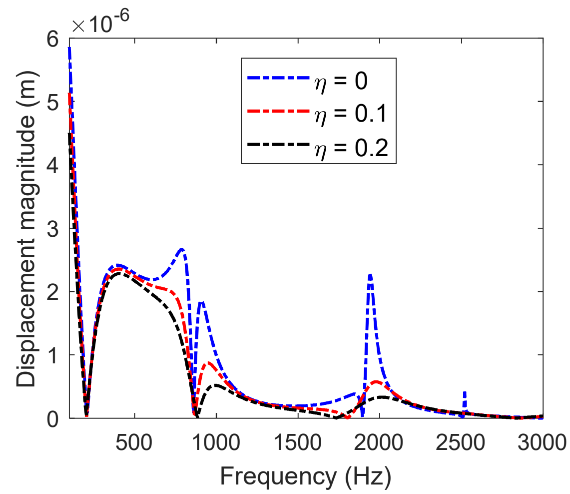


Figure 18: Effect of the membrane damping loss factor on displacement magnitude for a membrane thickness of 0.25 mm.

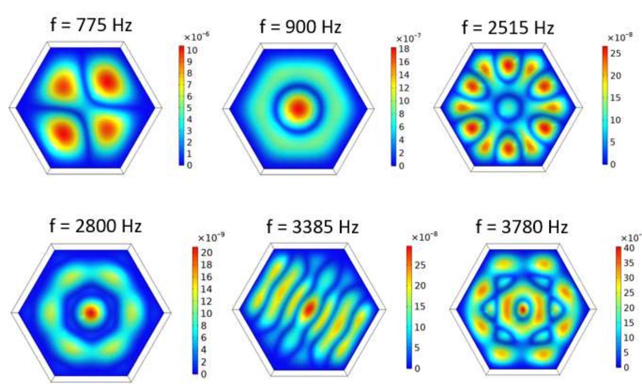


Figure 19: Displacement mode shape of the membrane for a damping loss factor of zero.

4 Honeycomb structure with multiple embedded membranes

In this section, numerical studies are carried out on the honeycomb structure with multiple embedded membrane layers.

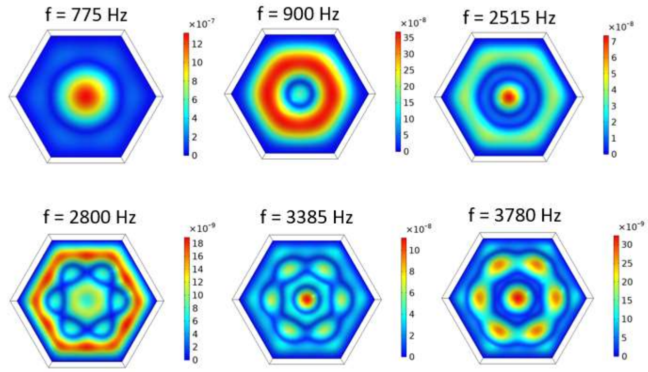


Figure 20: Displacement mode shape of the membrane for a damping loss factor of 0.2.

4.1 Honeycomb structure with two embedded membranes

Figure 21 shows a honeycomb cell with two integrated membrane layers. D represents the thickness of the air gap between the two membranes. Fixed boundary conditions are considered for each membrane with the same thickness t. The geometry is shown in Fig. 21(a) where the incident and transmission fluid have the same length. The mesh is illustrated in Fig. 21(b). Each membrane is characterized as a linear isotropic elastic material using the solid mechanics module of COMSOL and the material properties of each membrane are identical to membrane 2 of Table 1 with zero damping loss factor. An incident pressure of 1 Pa is applied on the inlet plan and the transmission loss is calculated using Eq. (1).

In Fig. 22, the influence of the thickness t of each membrane on the transmission loss is illustrated. The thickness of the air gap D is set to 10 mm and the cell size R is 5 mm.

With a thickness of 0.25 mm, the TL in Fig. 22 presents a peak at 2000 Hz with a value of 114 dB and drops to zero at the frequencies around 610 Hz, 1425 Hz, 2325 Hz and 2530 Hz. For a thickness of 0.5 mm, the TL presents a drop in the frequency range 1000-1600 Hz while outside this band, the TL is greater than 20 dB for f<1000 Hz and f>1600 Hz. With a thickness of 1 mm, the TL is greater than 30 dB for f<2000 Hz and increases when the frequency decreases.

Figure 23 shows the influence of the air gap thickness D on the transmission loss where the honeycomb cell size is set to 5 mm. Each membrane has a thickness of 0.5 mm. The TL in Fig. 23 for each value of D has two frequencies where the TL is zero. The first frequency drop of the TL around 1180 Hz is the same for all values of D. For air gap thicknesses of 3 mm, 5 mm, 10 mm, 15 mm and 20 mm, the second frequency drop is observed at 2150 Hz, 1855 Hz, 1570 Hz, 1440 Hz and 1375 Hz respectively. As D increases, the second frequency drop decreases. For D=20 mm, the frequency band where the TL drops is reduced and outside this frequency band, the TL is higher than the others. By increasing the thickness of the air gap, the TL in Fig. 23 is improved.

Figure 24 shows the impact of the cell size R on the transmission loss of the honeycomb structure with two integrated membranes layers. The thickness of each membrane is 0.5 mm with an air gap thickness D of 10 mm.

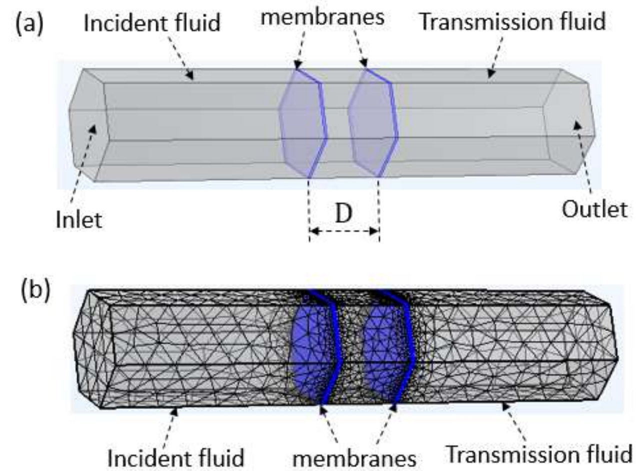


Figure 21: Honeycomb structure with two embedded membrane layers: (a) geometry (b) mesh.

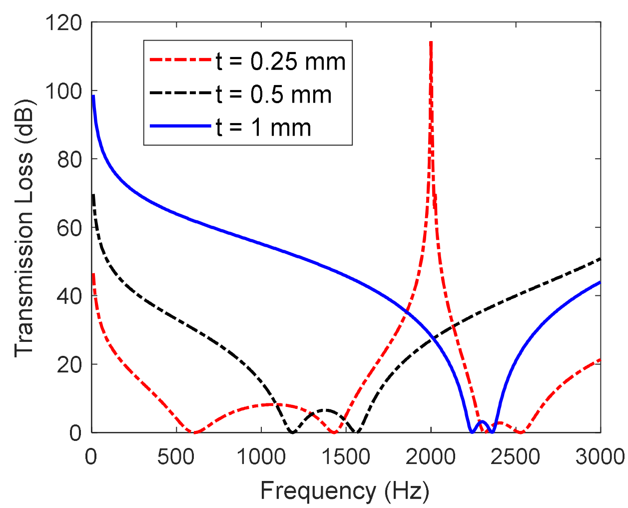


Figure 22: Effect of the membranes thickness on the transmission loss of honeycomb cell with two embedded membrane layers.

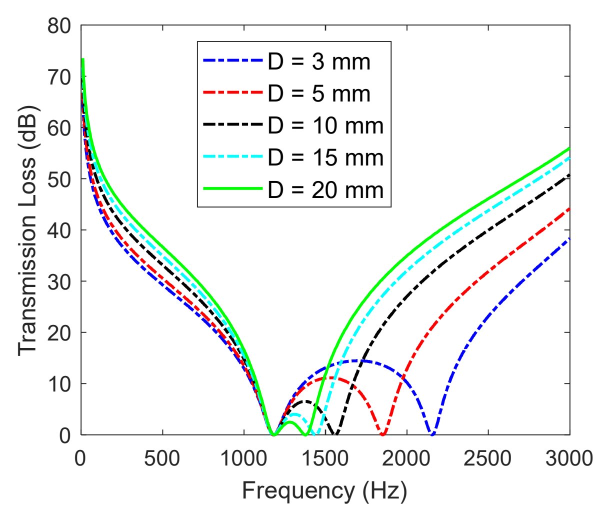


Figure 23: Effect of the air gap thickness on the transmission loss.

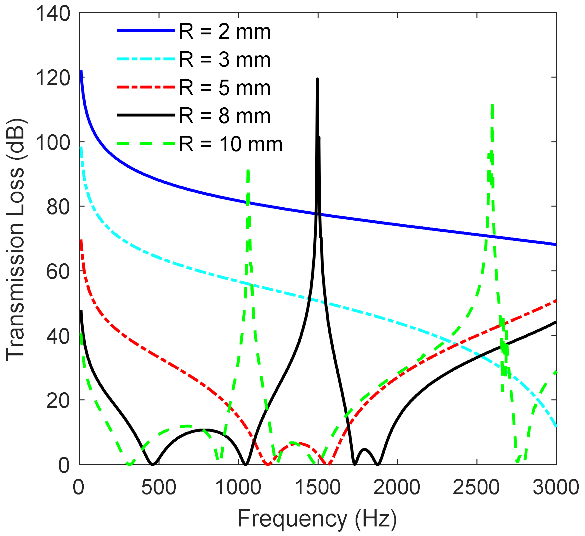


Figure 24: Effect of cell size on the transmission loss of honeycomb cell with two embedded membrane layers.

For cell dimensions of 2 mm and 3 mm, the TL in Fig. 24 exhibits a wide attenuation frequency band. For R = 2 mm, the TL is greater than 68 dB over the entire frequency range and with R=3 mm, the TL is greater than 43 dB for f<2000 Hz. With a dimension R of 5 mm, the TL presents a drop in the frequency range 1120-1640 Hz. When R is equal to 8 mm, there is a resonant frequency of 1495 Hz where the TL value reaches 119.5 dB and for R=10 mm, one observes two resonance frequencies at 1060 Hz and 2595 Hz where the TL peak values reach 92 dB and 113 dB respectively. Thus, by reducing the dimension of the honeycomb cell size, the TL increases over a wide frequency range.

4.2 Honeycomb structure with three embedded membranes

A honeycomb structure with three embedded membrane layers is shown in Fig. 25. The thickness of each membrane is set to 0.5 mm with fixed boundary conditions. The thickness of the air gap between the membranes is denoted by D_1 and D_2 and the material properties of membrane 2 in Table 1 are used for the three membranes with negligible (zero) loss factor.

The effect of the cell size R on the TL of honeycomb structure with three embedded membrane layers is shown in Fig. 26. The thickness of each air gap is set to 5 mm. For a cell dimension R of 3 mm, it is observed that the TL in Fig. 26 exhibits a wide attenuation frequency band. Three anti-resonance frequencies at 1190 Hz, 1570 Hz and 2090 Hz where the TL is zero appear for R=5 mm while there is a resonance TL peak at 1515 Hz for R=8 mm. For R=10 mm, the TL presents two resonant frequencies at 975 Hz and 2275 Hz where the TL peak values are 93 dB and 122 dB respectively. The TL frequency band is improved by reducing the cell size.

Figure 27 shows the TL for different air gap thicknesses by setting the honeycomb cell size to 5 mm.
The TL in Fig. 27 increases for f<1175 Hz when the air gap thickness increases. The three anti-resonance frequencies decrease respectively with the increase of D_1 and D_2 and the

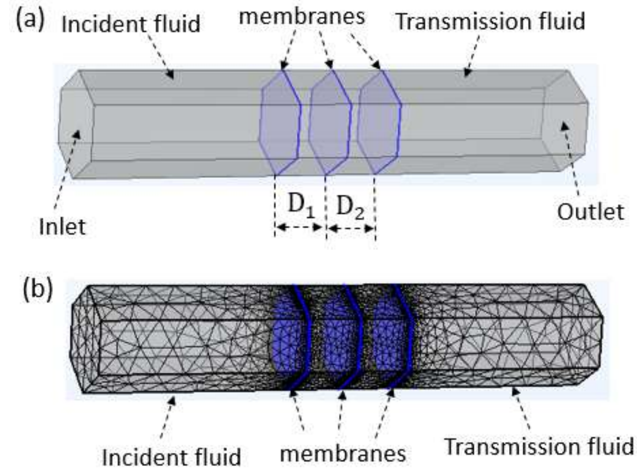


Figure 25: Honeycomb structure with three embedded membrane layers: (a) geometry (c) mesh.

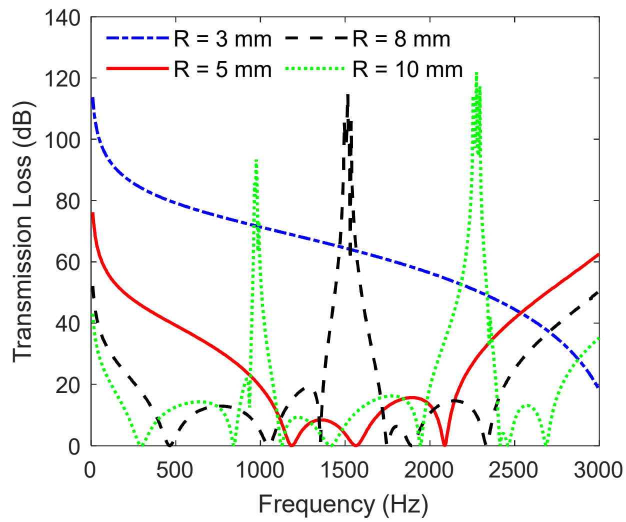


Figure 26: Effect of the cell size on the transmission loss of honeycomb cell with three embedded membrane layers.

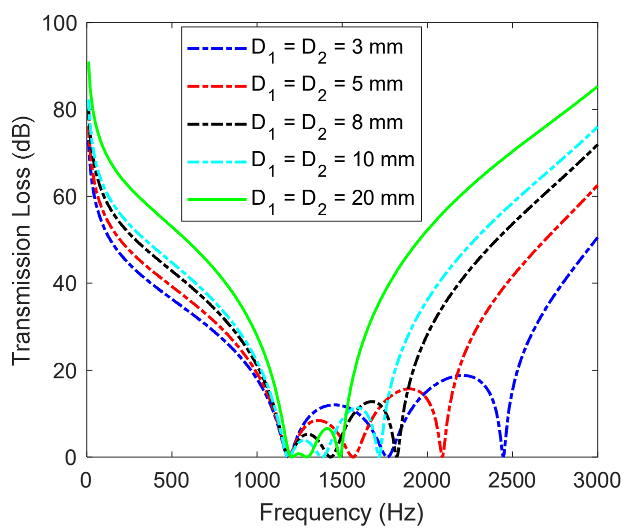


Figure 27: Effect of the air gap thickness on the transmission loss of honeycomb cell with three embedded membrane layers.

TL frequency drop band decreases.

Figure 28 illustrates the impact of the number of membrane layers within the honeycomb structure where the cell size is set to 5 mm. Membrane 2 of Table 1 with fixed boundary conditions is used with a thickness of 1 mm. For the two-membrane layer case, the thickness of the air gap D is set to 10 mm and for the three-membrane layer case; the thickness of each air gap D_1 and D_2 is set to 5 mm.

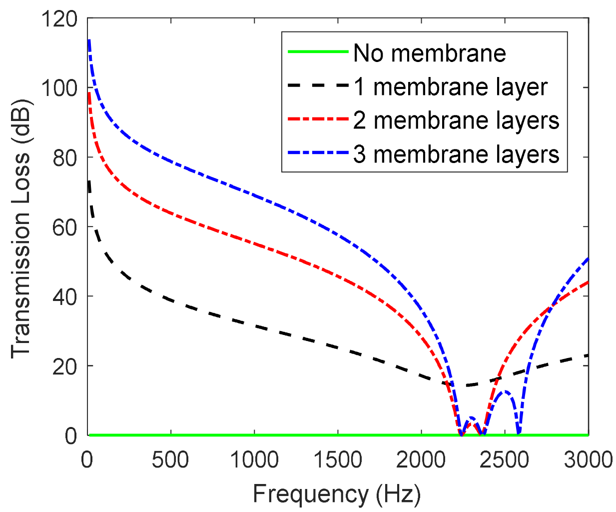


Figure 28: Effect of the number of membrane layers on the transmission loss.

The TL of the honeycomb structure alone without membrane in Fig. 28 is zero over the entire frequency range. When the number of membrane layers within the honeycomb structure increases, the TL increases for frequencies lower than 2000 Hz.

The honeycomb membrane-type acoustic metamaterial that is studied in this paper can provide engineering solutions in applications where the reduction of low frequency noise with a minimum added weight while offering structural stiffness is required. This type of metamaterial can be useful in many noise control engineering applications such as aerospace, manufacturing and process industries, and land transportations.

5 Conclusion

A honeycomb membrane-type acoustic metamaterial made of a honeycomb structure with embedded membrane layers was studied using the finite element method. The transmission loss of the metamaterial showed significant improvement especially at low frequency while the TL of the honeycomb core alone is zero over the entire frequency range. It is observed that the TL increases over a large frequency band as the honeycomb cell size is reduced and the displacement magnitude of the membrane decreases. The impact of the membrane thickness on the TL and displacement magnitude and mode shape at different frequencies were demonstrated. The TL and displacement magnitude show multiple resonant peaks as the thickness of the membrane decreases. The influence of the membrane damping loss factor was investigated. Honeycomb structures with two and three embedded membrane layers were studied numerically and the effects of the number of membrane layers and of the thickness of the air gap between membranes on the TL were presented. The TL increases over a large frequency band when the number of membrane layers within the honeycomb structure increases. The investigated metamaterial can help in low frequency noise attenuation in many engineering applications including aerospace, land transportation, and various industries.

Acknowledgments

The authors would like to acknowledge the National Research Council Integrated Aerial Mobility Program for the financial support.

References

[1] Y. Li, Y. Zhang, and S. Xie. A lightweight multilayer honeycomb membrane-type acoustic metamaterial. Applied Acoustics, 168: 107427, 2020.

[2] N. Sui, X. Yan, T.-Y. Huang, J. Xu, F.-G. Yuan, and Y. Jing. A lightweight yet sound-proof honeycomb acoustic metamaterial. Appl. Phys. Lett. 106: 171905, 2015.

[3] K. Lu, J.H. Wu, D. Guan, N. Gao, and L. Jing. A lightweight low-frequency sound insulation membrane-type acoustic metamaterial. AIP Advances 6, 025116, 2016.

[4] H. Nguyen, Q. Wu, J. Chen, Y. Yu, H. Chen, S. Tracy, and G. Huang. A broadband acoustic panel based on double-layer membrane-type metamaterials. Appl. Phys. Lett. 118: 184101, 2021.

[5] Z. Yang, J. Mei, M. Yang, N.H. Chan, and P. Sheng. Membrane-type acoustic metamaterial with negative dynamic mass. Phys. Rev. Lett., 101:1–4, 2008.

[6] S. Li, D. Mao, S. Huang, and X. Wang. Enhanced transmission loss in acoustic materials with micro-membranes. Applied Acoustics, 130: 92–98, 2018.

[7] S. Qiu, S. Li, X. Wang, and D. Mao. Enhanced transmission loss through lattice-supported micro-membranes. Applied Acoustics, 153, 127–131, 2019.

[8] Z. Laly, C. Mechefske, S. Ghinet, B. Ashrafi, and C.T. Kone. Numerically analysis of honeycomb structure with embedded membrane for transmission loss improvement. Canadian Acoustics Proceedings, 50(3), 22-23, 2022.

[9] Nansha Gao, Hong Hou, and Jiu Hui Wu. A composite and deformable honeycomb acoustic metamaterial. International Journal of Modern Physics B 32(20), 1850204, 2018.

[10] Haoming Hu, Shuai Zhu, and Hongyu Cui. Sound insulation performance of Mis-haped mass membrane-honeycomb structure acoustic metamaterials. Proc. SPIE 12164, International Conference on Optoelectronic Materials and Devices, ICOMD 2021, Guangzhou, China.

[11] Fuyin Ma, Meng Huang, and Jiu Hui Wu. Ultrathin lightweight plate-type acoustic metamaterials with positive lumped coupling resonant. JOURNAL OF APPLIED PHYSICS 121, 015102, 2017.

[12] Polina Marinova, Stephan Lippert, and Otto von Estorff. On the numerical investigation of sound transmission through double-walled structures with membrane-type acoustic metamaterials. J. Acoust. Soc. Am. 142 (4), 2400–2406, 2017.

[13] Guojian Zhou, Jiu Hui Wu, Kuan Lu, Xiujie Tian, Xiao Liang, Wei Huang and Keda Zhu. An approach to broaden the low-fre-

quency bandwidth of sound insulation by regulating dynamic effective parameters of acoustic metamaterials. J. Phys. D: Appl. Phys. 52 215102, 2019.

[14] Juan Mo, Zongren Peng and Xu Wang. Achieving Enhanced Sound Insulation through Micromembranes-Type Acoustic Metamaterials. Appl. Sci. 12, 1950, 2022.

[15] F. Langfeldt, W. Gleine and O. von Estorff. An efficient analytical model for baffled, multi-celled membrane-type acoustic metamaterial panels. Journal of Sound and Vibration 417, 359–375, 2018.

[16] Z. Laly, C. Mechefske, S. Ghinet, B. Ashrafi, and C.T. Kone. Finite element design of acoustic metamaterial based on parallel Helmholtz resonators with embedded membranes. Canadian Acoustics Proceedings, 50(3), 20-21, 2022.

[17] Z. Laly, C. Mechefske, S. Ghinet, and C.T. Kone. Numerical modelling of acoustic metamaterial made of periodic Helmholtz resonator containing a damping material in the cavity. In Proceedings of 2022 Inter-national Congress on Noise Control Engineering, INTER-NOISE 2022, 21-24 August 2022, Glasgow, UK.

[18] Z. Laly, C. Mechefske, S. Ghinet, C.T. Kone and N. Atalla. Modelling of acoustic metamaterial sound insulator using a transfer matrix method for aircraft cabin applications. In Proceedings of 2022 Inter-national Congress on Noise Control Engineering, INTER-NOISE 2022, 21-24 August 2022, Glasgow, UK.

[19] Z. Laly, C. Mechefske, S. Ghinet, and C.T. Kone. Numerical design of acoustic metamaterial based on parallel Helmholtz resonators for multi-tonal noise control. In Proceedings of 2022 International Congress on Noise Control Engineering, INTERNOISE 2022, 21-24 August 2022, Glasgow, UK.

Évaluation du Bruit dans le Campus de l'École de Technologie Supérieure de Montréal et Proposition d'un Métamatériau Acoustique pour la Réduction du Bruit de Transformateur Électrique

Olivier Doutres [*,1], Maël Lopez [†,1], Kévin Rouard [‡,1], Louis-Philippe Campagna [♦,1], Titouan Cougoulic [♯,1], Anthony Jutras [♠,1], David Lauzon [♣,1], Pierre-Luc Pépin-Pagé [⸸,1] et Alexis Purson [§,1]

[1]Department of Mechanical Engineering, École de technologie supérieure (ÉTS), Montréal, Québec, Canada.

Résumé

Les campus universitaires situés au cœur des villes peuvent être exposés à des niveaux de bruit importants qui peuvent nuire à l'apprentissage des étudiants et étudiantes, à la performance de la communauté enseignante et des membres du personnel mais aussi à la qualité de vie des résidents et résidentes du quartier. Cet article présente les résultats d'un projet d'étude réalisé par des étudiants du cours d'acoustique industrielle de l'École de technologie supérieure (ÉTS) et qui avait pour principal objectif de quantifier et évaluer le bruit extérieur et intérieur du campus universitaire de l'ÉTS. Les étudiants devaient aussi localiser les principales sources de bruit intérieures à l'aide d'une caméra acoustique et modéliser un métamatériau acoustique qui permettra de réduire le bruit tonal émis par les transformateurs en basses fréquences.

Mots clés : bruit environnemental, acoustique, université, campus, bruit intérieur, métamatériau acoustique

Abstract

University campuses located in the heart of cities can be exposed to significant noise levels that can hinder the learning of students, the performance of the teaching community and staff, as well as the quality of life for residents in the neighborhood. This article presents the results of a study project conducted by students of the "Industrial Acoustics" course at École de technologie supérieure (ÉTS), with the main objective of quantifying and assessing the outdoor and indoor noise levels on the university campus. The students were also tasked with identifying the main sources of indoor noise using an acoustic camera and modeling an acoustic metamaterial that would help reduce the tonal noise emitted by transformers at low frequencies.

Keywords: environmental noise, acoustics, university, campus, indoor noise, acoustic metamaterial

1 Introduction

Le bruit peut avoir des effets délétères sur la santé des individus [1] : perte auditive, perturbation du sommeil, difficulté à communiquer ; effets cardiovasculaires et psychophysiologiques ; réduction des performances ; gêne ; et effets sur le comportement social. En milieu scolaire, le bruit peut aussi avoir un effet sur l'apprentissage, notamment sur la compréhension de la lecture, sur la mémoire et sur l'intelligibilité de la parole [2]. Ainsi, plusieurs études se sont intéressées au bruit de campus universitaires [3-5] et concluent généralement que les niveaux de bruit sont trop élevés pour ce genre d'environnement dédié à l'ap-prentissage. Le campus de l'École de technologie supérieure (ÉTS) est situé en plein cœur de la ville de Montréal (Canada) et est, sans surprise, exposé à un niveau de bruit assez élevé comme le montre la cartographie des niveaux de bruit réalisée en 2014 par Ragettli et coll. [6, 7] et présentée à la Figure 1(a). Selon cette carte, les niveaux de bruit dans ce quartier y sont bien supérieurs au niveau maximum de 55 dB(A) recommandé par l'Organisation mondiale pour la santé (OMS) [1] (recommandation pour les espaces extérieurs des écoles). Les nuisances sonores de ce quartier central de Montréal sont en effet bien réelles et le bruit du campus a été mentionné à plusieurs reprises lors d'une consultation sur le développement urbanistique du campus, réalisée en 2018 [8]. Voici, par exemple, quelques suggestions issues de ces consultations : « *Créer des ambiances sonores reposantes* », « *Concevoir des murs verts pour contrer la pollution et le bruit* », « *Construire des havres de paix (atténuer les nuisances sonores) ouverts au public, mais destinés aux employés et aux étudiants de l'ÉTS et conserver ainsi un équilibre entre les besoins des membres de la communauté de l'ÉTS et les résidents du quartier* ». Néanmoins, les niveaux de bruit dans les différentes parties du campus ÉTS ne sont pas connus avec précision. Une cartographie plus détaillée permettrait d'identifier (i) les zones les plus calmes qui seraient ainsi les zones les plus adaptées à un repos en extérieur, comme souhaité par la communauté et (ii) les zones les plus bruyantes qui nécessiteraient des correctifs acoustiques afin d'améliorer le confort des résidents et résidentes du quartier et de la communauté ÉTS.

[*] olivier.doutres@etsmtl.ca
[†] mael.lopez.1@ens.etsmtl.ca
[‡] kevin.rouard.1@ens.etsmtl.ca
[♦] louisphilippe48@hotmail.com
[♯] cougoulic.titouan@gmail.com
[♠] anthonyjutras@hotmail.com
[♣] david_lauzon@hotmail.com
[⸸] p-l.pp@hotmail.com
[§] alexis.purson@hotmail.fr

Les environnements acoustiques intérieurs des différents pavillons sont tout aussi importants. Ils doivent être adaptés au contexte d'apprentissage, mais aussi favorables au travail de bureau pour tous les membres du personnel du campus. Tout comme pour le bruit extérieur, les niveaux de bruit intérieur au campus ÉTS ne sont pas connus et doivent être mesurés.

Une classe du cours « Acoustique industrielle » de l'ÉTS [10] (sigle MEC636) a eu pour mission, dans le cadre de leur projet de session, de réaliser une étude de bruit dans le campus de l'ÉTS pour contribuer à améliorer les environnements acoustiques et ainsi la qualité de vie de la communauté. La première étape du projet a consisté à étudier le bruit extérieur dans le campus ÉTS. La seconde étape a consisté à caractériser les environnements sonores de plusieurs locaux des différents pavillons de l'ÉTS (ex., auditorium, salles de classe, cafétéria, bureaux, bibliothèque) et évaluer la qualité acoustique de ces environnements. Les sources de bruit d'intérêt dans ce projet étaient des sources stationnaires associées au fonctionnement des bâtiments comme la ventilation, les systèmes mécaniques et électriques, les serveurs informatiques. Enfin, les étudiants devaient localiser les principales sources de bruit internes aux pavillons, dont celles des salles électrique et mécanique. Ils devaient aussi proposer un concept de métamatériau acoustique dédié à l'encoffrement des transformateurs électriques afin de réduire leur éventuelle nuisance dans les locaux adjacents.

Ce papier a pour objectif de présenter l'étude de bruit du campus ÉTS réalisée par les étudiants du cours d'acoustique industrielle de l'ÉTS dans le cadre de leur projet de session. Le contexte pédagogique de ce projet étudiant est tout d'abord présenté à la section 2. La section 3 présente ensuite le matériel de mesure utilisé, les environnements extérieurs et intérieurs évalués ainsi que les indicateurs utilisés pour caractériser leur qualité acoustique. La section 3 se termine par la présentation du modèle utilisé pour simuler le comportement acoustique du métamatériau destiné à la réduction du bruit des transformateurs. La section 4 présente et discute les résultats de l'étude. La section 5 rappelle les principales conclusions et décrit les perspectives de ce projet.

2 Contexte pédagogique du projet

Le cours « Acoustique industrielle » MEC636 est un cours de spécialisation de fin de baccalauréat en génie mécanique de l'ÉTS. Il vise à rendre les étudiants et étudiantes aptes à mesurer et à réduire le bruit en s'appuyant sur les bases théoriques de l'acoustique industrielle et les techniques expérimentales associées. Ce cours est principalement basé sur trois éléments pédagogiques non conventionnels [11] : (i) une méthode pédagogique active basée sur la pédagogie de la coopération, (ii) l'utilisation intensive de l'outil informatique par l'intermédiaire de séances de travaux pratiques et d'examens en laboratoire informatique et (iii) un projet de session en équipe.

La session compte 13 semaines d'enseignement. Le projet de session, qui fait ici l'objet de cet article, était composé de trois séances de laboratoire et d'une séance de travaux pratiques. Le projet débutait par les trois laboratoires aux semaines 8, 9 et 10. Le premier laboratoire avait pour objectif la réalisation des cartographies des niveaux de bruit extérieurs dans le campus ÉTS. Les deux laboratoires suivants avaient pour objectif la caractérisation de multiples environnements acoustiques intérieurs des principaux pavillons du campus (pavillons A, B, D et E, voir carte de la Figure 1(b)).

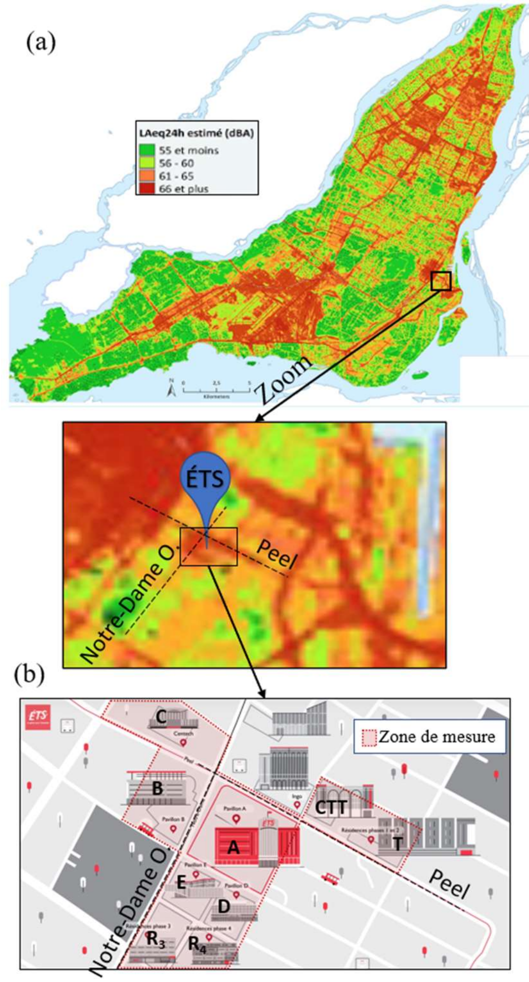


Figure 1 : (a) Carte des niveaux de bruit de l'île de Montréal (tirée de [6, 7]); (b) Campus ÉTS et zone de mesure du bruit extérieur.

Le projet de session présenté dans ce papier diffère un peu du projet des années précédentes qui avait pour objectif la réduction de bruit d'un petit équipement domestique (ex., mélangeur de cuisine, souffleuse à feuille, sèche-cheveux) [11]. Toutefois, ces deux types de projets permettent aux étudiants et étudiantes de mettre en application toutes les connaissances théoriques et expérimentales apprises pendant le cours.

Préalablement au premier laboratoire de projet, les étudiants étaient formés au diagnostic acoustique des environnements et des sources de bruit et donc à l'utilisation d'instruments de mesure des niveaux de pression acoustique (niveau global), mais aussi à la représentation des signaux dans le domaine fréquentiel (ex., bandes d'octaves, bandes fines). Les étudiants avaient déjà effectué des mesures de bruit, analysé et interprété les résultats dans le but d'évaluer des plaintes de bruit (en milieu de travail principalement). Après la 7^{ième} semaine de cours, les bases théoriques de la propagation des ondes acoustiques dans les fluides dissipatifs et non-dissipatifs ainsi que la méthode des matrices de transfert [10, 12] leur avaient été enseignées. Cette dernière est utilisée dans le cadre du cours MEC636 afin de simuler le comportement acoustique en absorption et en transmission de divers systèmes de réduction du bruit tels que les parois simples et multiples ainsi que les silencieux réactifs et dissipatifs. La séance de travaux pratiques du projet de session (à la semaine 13 de la session) permettait d'appliquer ces connaissances. L'objectif de la séance était de concevoir un matériau acoustique composé d'un pavage de résonateurs « quart d'onde » et de Helmholtz, aussi appelé métamatériau, afin d'absorber l'énergie acoustique à des fréquences ciblées et problématiques de sources de bruit identifiées lors des campagnes de mesures intérieures dans le campus.

3 Matériel et méthode

3.1 Matériel

Bruit extérieur

Les mesures extérieures ont été réalisées avec l'application NoiseCapture [13, 14] installée sur les téléphones cellulaires des étudiants (i.e., modèles Galaxy A23 5G, A52 5G, S20 FE 5G (Samsung, Séoul, Corée du Sud) et Pixel 3A (Google, Mountain View, CA, USA)) (voir Figure 2(a)). Cette application permet de réaliser des mesures de niveau de bruit et de les combiner avec leur trace GPS afin de pouvoir les afficher sur une carte interactive au sein de l'application. Les appareils ont été calibrés manuellement juste avant la séance de mesure à l'aide d'une procédure d'étalonnage manuelle dirigée par le chargé de laboratoire. Cette procédure consiste à corriger le niveau sonore obtenu par l'application par comparaison avec une mesure simultanée avec un sonomètre calibré.

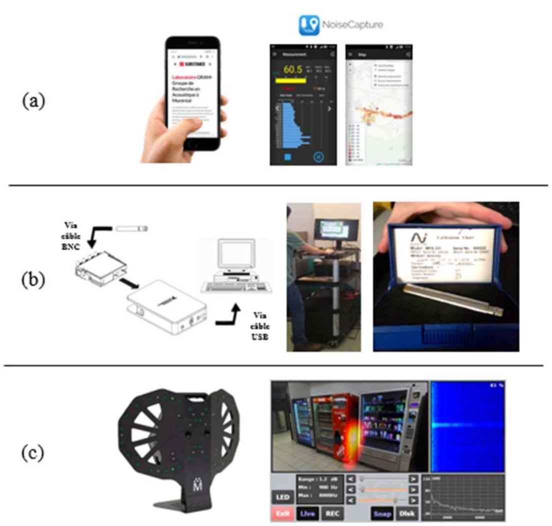


Figure 2 : (a) Téléphone cellulaire et application « NoiseCapture » [13, 14] pour les mesures en extérieur; (b) instrumentation du cours MEC636 pour les mesures en intérieur; (c) caméra acoustique LF-ANT (Mecanum, Sherbrooke QC, Canada) pour l'imagerie acoustique [15].

Bruit intérieur

Les mesures de bruit dans les environnements intérieurs ont été réalisées à l'aide de microphones demi-pouce champ-libre (type MPA231) de classe 1 de la marque BSWA (Beijing, Chine) ainsi que de cartes d'acquisition National Instruments (Austin, TX, USA) cDAQ-9171 (voir Figure 2(b)). Les chaînes de mesure ont été calibrées à l'aide d'un calibreur Larson Davis (Depew, NY, USA) CAL200. Le logiciel « MEC636-V4 », développé à l'ÉTS sous environnement Labview (National Instruments, Austin, TX, USA), a été utilisé pour l'acquisition et le post-traitement des mesures. Une caméra acoustique LF-ANT (Mecanum, Sherbrooke, QC, Canada) [15, 16] (voir Figure 2(c)) a aussi été utilisée pour réaliser des images acoustiques des environnements et localiser les principales sources de bruit. À l'instar d'une caméra thermique qui indique les points chauds de température, une caméra acoustique indique les zones où le bruit est le plus élevé et permet donc de voir le bruit. Cet équipement a été acheté pour compléter les outils du cours MEC636 destinés au diagnostic acoustique des environnements et des sources de bruit. Dans le projet de session, cette caméra a été utilisée dans la perspective d'améliorer le confort acoustique des espaces d'apprentissage et de travail du campus en traitant les principales sources de bruit des pavillons.

3.2 Environnements

Bruit extérieur

La campagne de mesures du bruit extérieur a été réalisée dans la zone du campus de l'ÉTS représentée par la surface rouge sur la Figure 1(b). L'intersection principale du campus est située au coin des rues Notre-Dame Ouest et Peel. Les différentes zones du campus indiquées sur la carte sont : (i) les pavillons A, B, D et E (qui incluent des salles de classe, des bureaux, une bibliothèque, des auditoriums et salles de conférence, des cafétérias, un centre sportif et un centre de la petite enfance), (ii) le Centech C qui est un incubateur technologique, (iii) les résidences étudiantes T, R_3 et R_4 et (iv) le centre de technologie thermique CTT.

Deux périodes de mesures ont été réalisées le 22 février 2023 en après-midi: (1) une première, de 14h30 à 15h30, appelée « heure creuse », (2) une seconde, de 16h00 à 17h00, appelée « heure de pointe ». Ces deux périodes ont été choisies, car l'ÉTS est située sur des axes routiers importants de Montréal et d'importantes différences de niveaux de bruit étaient attendues entre les deux périodes, avec un niveau plus élevé pendant les heures de pointe.

Bruit intérieur

Les deux campagnes de mesures des environnements intérieurs de l'ÉTS ont été réalisées les 23 février et 9 mars 2023. Les lieux mesurés ont été séparés en trois catégories. Les deux premières catégories correspondent aux « **espaces d'apprentissage centraux** » et « **espaces d'apprentissage auxiliaires** », comme cela est défini dans la norme ANSI/ASA S12.60 [17]. La première catégorie inclut les locaux d'enseignement et d'apprentissage à aire ouverte ou fermée et pour lesquels la communication orale est essentielle à la réussite académique des étudiants et étudiantes. Cette catégorie englobe donc en partie les salles de classe, la bibliothèque et les auditoriums. Les mesures ont majoritairement été réalisées lorsque les locaux étaient inoccupés et/ou avec des personnes silencieuses. Les sources de bruit dans ces locaux étaient principalement celles du système de ventilation et de climatisation. La seconde catégorie regroupe les espaces d'apprentissage pour lesquels la communication est essentielle pour l'étudiant ou l'étudiante, mais dont les fonctions premières ne sont pas l'apprentissage formel. Elles incluent plutôt l'apprentissage informel, les interactions sociales et autres activités similaires. Ces espaces incluent en partie les aires communes (ex. atriums), cafétérias, salles de sports et les locaux pour la vie étudiante tels que les clubs. La troisième catégorie correspond aux « **salles électrique et mécanique** » de différents pavillons du campus et inclut aussi les locaux adjacents qui peuvent être impactés par les sources de bruit de ces salles.

3.3 Indicateurs et valeurs maximales recommandées

Cette section présente les différents indicateurs utilisés pour caractériser les environnements acoustiques extérieurs et intérieurs ainsi que les valeurs maximales recommandées pour les environnements mesurés et tirés des documents de références (ex., OMS [1, 9], norme ANSI/ASA S.12.60 [17], manuel ASHRAE [18]).

Bruit extérieur

L'application « NoiseCapture » permet de mesurer le niveau de bruit équivalent pondéré A à chaque seconde ($L_{Aeq,1s}$) pendant que l'enregistrement est lancé et que l'étudiant se déplace dans le campus. Le niveau de pression acoustique mesuré avec une pondération A permet d'approcher la façon dont l'oreille humaine perçoit les différentes composantes fréquentielles des sons à des niveaux d'écoute typiques de la parole. À la fin de chaque campagne de mesures (« heure creuse » et « heure de pointe »), l'application pave l'espace en hexagones de 15 mètres de rayon équivalent. Pour chaque campagne de mesures, l'application concatène toutes les mesures réalisées dans chacun des hexagones et fournit un niveau de bruit équivalent L_{Aeq} par hexagone [13, 14]. La durée des mesures réalisées pendant les deux campagnes était comprise entre 30 secondes et 5 minutes. Le temps de mesure cumulé de tous les étudiants était de 1h57 pour la campagne « heure creuse » et 1h55 pour la campagne « heure de pointe ».

Les membres de la communauté de l'ÉTS qui circulent à l'extérieur des bâtiments du campus sont majoritairement exposés au bruit routier. La valeur d'exposition (sur 24h, $L_{Aeq,24h}$) maximale recommandée par l'OMS afin de prévenir les effets du bruit pour les sources liées à la circulation routière (c.-à-d., maladies ischémiques cardiovasculaires; diabète de type 2; dérangement, perturbations du sommeil, difficulté de lecture et de compréhension orale) est de 50 dB(A) [7, 9]. Même si en pratique, les mesures ont été réalisées pendant des durées beaucoup plus courtes que 24h (pour des raisons de praticité évidentes), elles pourront tout de même être comparées à une valeur seuil définie sur 24h [6, 19]. Une autre valeur limite de l'OMS, plus permissive, de 55 dB(A) était recommandée pour les environnements extérieurs des écoles [1]. Cette valeur maximum est considérée dans ce travail, car elle a souvent été utilisée dans les études similaires réalisées dans des campus universitaires [3, 4].

Bruit intérieur

Deux indicateurs sont majoritairement utilisés pour caractériser la qualité acoustique des espaces d'apprentissage (centraux et auxiliaires) [17] : le niveau de bruit de fond (niveau équivalent pondéré A, L_{Aeq}) et le temps de réverbération (TR). Ces deux indicateurs se mesurent lorsque les locaux sont inoccupés. La mesure du bruit de fond dans un local permet de caractériser l'ampleur des contributions des sources de bruit extérieures (ex., trafic routier, trafic aérien, usines, activité dans les cours d'école) et intérieures (ex., bruit de ventilation, bruit dans les locaux voisins). Le TR mesure l'ampleur de la réverbération dans un local et est égal au temps nécessaire pour que le niveau d'un son continu diminue de 60 dB après avoir été éteint. Ce temps dépend du volume du local, de l'absorption des matériaux sur les surfaces et de la fréquence. Dans ce projet, le TR a été mesuré dans les salles de classe (catégorie des espaces d'apprentissage centraux) et les mesures de niveaux de bruit ont été réalisées sur des durées de 10 à 15 secondes (à cause des contraintes de temps associées à la durée limitée des laboratoires d'enseignement dédiés au projet).

Un bruit de fond et/ou une réverbération excessifs dans ces espaces interfèrent avec la communication orale et constituent alors un obstacle « acoustique » à l'apprentissage [17]. Ainsi des valeurs maximums sont préconisées dans des ouvrages de référence [17, 18] et sont rappelées dans le Tableau 1 ci-dessous.

Les valeurs maximales dépendent évidemment de l'utilisation des locaux. La qualité acoustique d'un espace d'apprentissage central doit être supérieure à celle d'un espace d'apprentissage auxiliaire et ainsi les valeurs maximums conseillées pour ce premier sont plus basses. La littérature est tout particulièrement fournie pour les salles de classe puisque cet endroit est de la plus haute importante pour la communication orale et l'apprentissage de la communauté étudiante. Un article de revue de la littérature sur le sujet [20] conclut que, pour des salles de classe de petit et moyen volume, un TR compris entre 0,6 et 0,7 est adéquat pour les étudiants et étudiantes quel que soit leur âge et que le niveau de bruit de fond ne doit pas dépasser 40 dB(A) pour les étudiants et étudiantes de plus de 12 ans.

3.4 Modélisation du métamatériau acoustique

Dans le but de réduire le bruit basses fréquences des salles électrique et mécanique, perceptible dans les locaux voisins (voir section 4.2), un métamatériau acoustique a été proposé. Ce matériau servira de base d'encoffrement pour la principale source du local identifié à l'aide de la caméra acoustique. Le métamatériau est constitué d'un pavage d'une cellule unitaire

Tableau 1 : Valeurs maximales recommandées pour les espaces d'apprentissage centraux et auxiliaires.

Catégorie	Type d'espace	L_{Aeq} max (dB(A))	TR (s) dans les bandes d'octave 500, 1000 et 2000 Hz
Espaces d'apprentissage centraux	Salles de classe, bibliothèque, bureaux privés, salles de conférence, salle de pratique de musique.	35 (volume $\leq$ 566 m^3) [17] 40 (volume > 566 m^3) [17]	0,6 (volume < 283 m^3) [17] 0,7 (283 m^3 < volume $\leq$ 566 m^3) [17}
	Salles de classe (100 m^3 < volume $\leq$ 290 m^3)	40 pour des élèves de 12 ans et plus [20]	0,6<TR<0,7 [20]
Espaces d'apprentissage auxiliaires	Cafétéria	40 [17]	
	Gymnase	40 [17] 50 [18]	
	Bureaux à aire ouverte	45 [18]	
	Espaces à grande capacité d'accueil avec amplification de la parole	55 [18]	

absorbante constituée d'un multi-résonateur de Helmholtz (acronyme utilisé HR, pour « ***Helmholtz Resonator*** » en anglais) (à deux degrés de liberté) et d'un résonateur quart d'onde (acronyme utilisé QR, pour « ***Quarter-wavelength Resonator*** » en anglais) comme le montre la Figure 3(a). Le comportement en absorption du matériau a été modélisé par la méthode des matrices de transfert en considérant une excitation sous ondes planes d'incidence normale à la surface du matériau. Dans ce cas, une seule cellule est suffisante pour sa modélisation (voir Figure 3(b)). Les résonateurs sont conçus pour absorber l'énergie acoustique à quatre fréquences identifiées comme étant les plus problématiques.

Le coefficient d'absorption de la cellule unitaire est déterminé à partir de l'impédance acoustique d'entrée Z et de l'impédance caractéristique de l'air Z_0:

$$\alpha = 1 - \left|\frac{Z - Z_0}{Z + Z_0}\right|^2 . \quad (1)$$

L'impédance d'entrée de la surface de la cellule unitaire du métamatériau Z se calcule à partir des impédances acoustiques d'entrée du résonateur quart d'onde Z_{QR} et du multi-résonateur de Helmholtz Z_{HR} selon la méthode des sommes des admittances [21, 22]:

$$Z = \left(\frac{S_{\mathrm{t}}}{S_{\mathrm{QR}}}\frac{1}{Z_{\mathrm{QR}}} + \frac{S_{\mathrm{t}}}{S_{\mathrm{HR}}}\frac{1}{Z_{\mathrm{HR}}}\right)^{-1}, \quad (2)$$

avec S_{t} la surface totale de la cellule unitaire, S_{QR} la surface d'entrée résonateur quart d'onde et S_{HR} la surface d'entrée du multi-résonateur de Helmholtz. La modélisation par matrice de transfert des impédances acoustiques des deux résonateurs (Z_{QR} et Z_{HR}) est présentée en annexe.

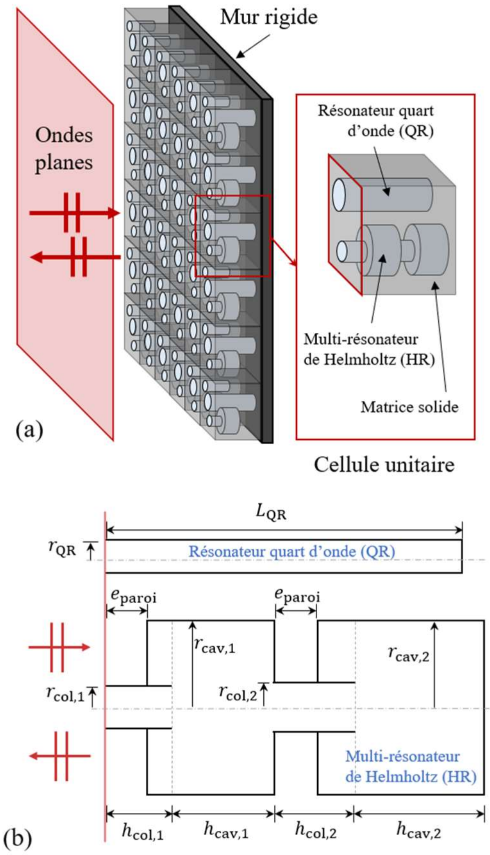


Figure 3 : Métamatériau acoustique ; (a) schématisation tridimensionnelle, (b) vue en coupe d'une cellule unitaire.

4 Résultats et discussion

4.1 Bruit extérieur

Les cartes de bruit des deux campagnes de mesures extérieures « heure creuse » et « heure de pointe » sont présentées respectivement sur les Figures 4(a) et 4(b). Globalement, pour les deux périodes de mesures, les tendances présentées sur la carte des niveaux de bruit de la Figure 1(a) (voir zoom) sont retrouvées : (i) les zones les plus exposées au bruit sont les rues Notre-Dame Ouest et Peel (ainsi que la zone autour du bâtiment C) et (ii) le niveau de bruit de ces zones est généralement supérieur à 65 dB(A). Une grande partie du campus de l'ÉTS est donc exposée à des niveaux bien supérieurs à ceux recommandés par l'OMS.

La source principale de bruit dans le campus ÉTS est le trafic routier. Dans le cas de la zone autour du bâtiment C, les hauts niveaux de bruit pourraient être induits par les départs de bus d'une gare située un peu plus au nord (non visible sur la figure), mais qui empruntent la rue au nord de la Zone C pour accéder, entre autres, à une autoroute (voir pictogramme sur la Figure 4). Néanmoins, la Figure 4 montre que, contre-intuitivement, les niveaux de bruit semblent plus élevés pendant la période d'heure creuse que pendant l'heure de pointe. Cela peut s'expliquer par le nombre important de travaux de construction dans le quartier, générant du bruit de construction, mais aussi du bruit routier (les bruits de camions à benne étant particulièrement élevés [23]), et ce principalement avant 16h. Par exemple, un niveau de bruit élevé est observé

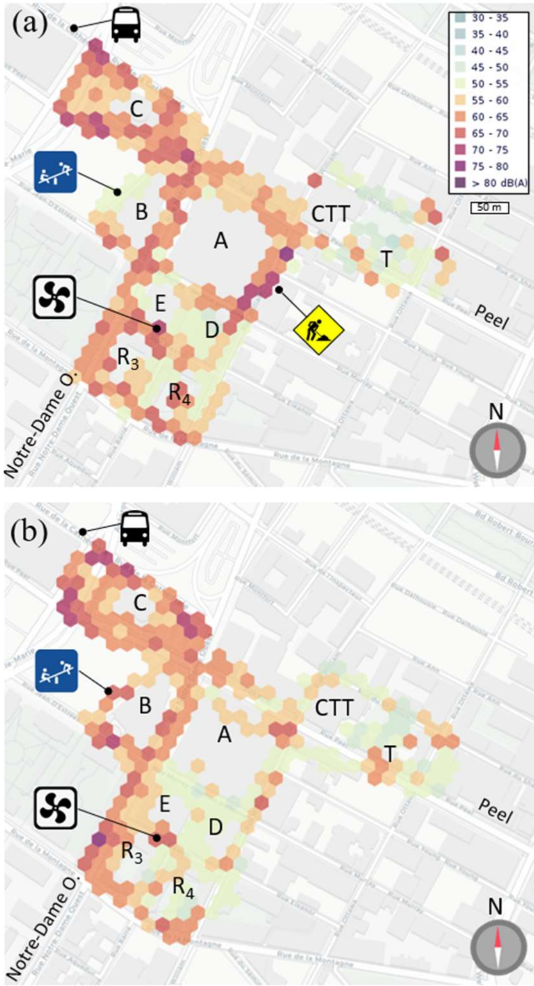


Figure 4 : Cartographie du bruit extérieur dans le campus ÉTS pendant les périodes : (a) heure creuse, (b) heure de pointe.

uniquement pendant la période d'heure creuse dans la rue longeant la face sud-est du pavillon A et qui est causé par des travaux de construction d'un nouveau pavillon de l'ÉTS (voir pictogramme jaune sur la Figure 4(a)).

Quatre zones plus « calmes » (L_{Aeq} inférieur à 55 dB(A)) peuvent être identifiées sur les deux cartes de bruit du campus ÉTS et correspondent aux cours extérieures : (i) entre les pavillons E et D, (ii) entre les résidences R_3 et R_4, (iii) entre le CTT et les résidences T et (iv) au nord-ouest du pavillon B, qui est une aire de jeu du centre de la petite enfance de l'ÉTS (voir pictogramme bleu sur la Figure 4). Pour cette dernière zone, le bruit est plus élevé pendant la mesure « heure de pointe », car les enfants étaient en train de jouer dans la cour. Il est tout de même intéressant de noter que cet espace est raisonnablement protégé du bruit du trafic routier (voir Figure 4(a)). Les trois premières zones « calmes » listées ci-dessus seraient à prioriser pour passer un temps de repos (ex. dîner) à l'extérieur dans le campus ÉTS. Malheureusement, ce n'est pas le cas du parc entourant le bâtiment C qui se trouve très impacté par le bruit routier. Ce dernier pourrait faire l'objet de correctifs acoustiques (ex. écrans verts) pour améliorer le confort acoustique des usagers.

Finalement, une source de bruit notable apparaît sur les deux cartes de la Figure 4. Il s'agit d'une sortie de ventilation au sud du pavillon E et identifiée par un pictogramme sur la Figure 4. Ce bruit peut être perçu dans la zone comprise entre les pavillons E et D, ce qui réduit l'espace « calme » entre ces deux pavillons.

4.2 Bruit intérieur

Cette section présente les résultats des études de bruit en intérieur pour les trois types d'espace présentés à la section 3.2.

Espaces d'apprentissage centraux

Au total, 15 locaux appartenant à la catégorie des espaces d'apprentissage centraux ont été mesurés. La Figure 5(a) présente la distribution des niveaux de bruit pour les 15 locaux et montre que 80 % des locaux présentent un niveau de bruit inférieur à 40 dB(A) et 40 % un niveau inférieur ou égal à 35 dB(A). Parmi ces 15 locaux, 5 sont des salles de classe (de volume moyen ≈ 290 m^3) et 80 % de ces salles de classe ont un niveau de bruit inférieur à 40 dB(A) (voir Figure 5(b)) ce qui est tout à fait adapté à l'apprentissage selon [20]. De plus, les TR mesurés aux différentes bandes d'octaves (c.-à-d., 500 Hz, 1000 Hz et 2000 Hz) dans ces salles de classe étaient tous inférieurs à 0,7 seconde, ce qui encore une fois est jugé tout à fait adéquat selon [20] (même si légèrement supérieur aux recommandations de la norme ANSI/ASA [17]). La seule salle de classe mesurée qui dépasse les 40 dB(A) est située sous le 6ième étage du pavillon D où se situe la principale salle mécanique du pavillon et est adjacente à un puits mécanique. Malgré cela, le niveau mesuré est de 41 dB(A) ce qui reste très proche de la valeur limite proposée dans [20]. La Figure 5(c) montre aussi que le pavillon D possède plus de locaux sous le seuil de 40 dB(A) par rapport au pavillon A qui est plus ancien.

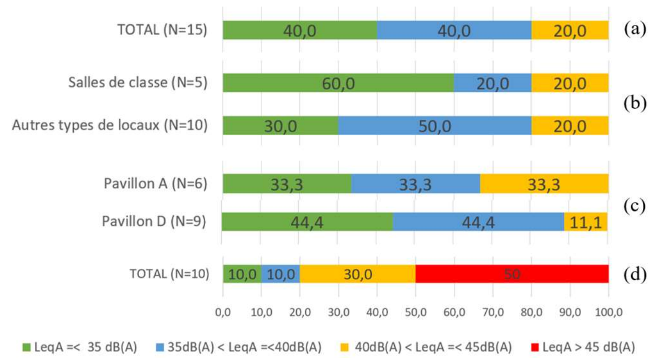


Figure 5 : Distribution des niveaux de bruit, en dB(A), dans les espaces d'apprentissage centraux : (a) tous les locaux mesurés, (b) salles de classe VS les autres types de locaux, (c) locaux du pavillon A VS ceux du pavillon C plus récent. (d) Niveau de bruit, en dB(A), dans les espaces d'apprentissage auxiliaires.

Espaces d'apprentissage auxiliaires

Dix locaux appartenant à la catégorie des espaces d'apprentissage auxiliaires ont été mesurés pendant les séances de laboratoire dédiées aux mesures intérieures : l'espace cafétéria, le gymnase, des locaux des clubs étudiants (2 fois), des espaces ouverts collaboratifs (3 fois), l'atrium du pavillon E dans lequel se tient par moment des prestations amplifiées (2 fois) et un espace (fermé) à grande capacité d'accueil avec amplification de la parole. La Figure 5(d) montre la distribution des niveaux de pression pour ces espaces d'apprentissage auxiliaires et la Figure 6 présente des images acoustiques prises dans certains de ces espaces. La moitié des espaces mesurés présentent un niveau de bruit inférieur ou égal à 45 dB(A). Les espaces qui dépassent cette valeur sont : (i) la cafétéria du pavillon A avec 50 dB(A), à priori à cause des nombreux équipements de refroidissement qui s'y trouvent (voir Figure 6(a)), (ii) le gymnase avec 57 dB(A), à cause de la ventilation (voir Figure 6(b)), (iii) un espace collaboratif du pavillon D avec 56 dB(A), qui se trouve proche d'une cafétéria et (iv) l'atrium du pavillon E avec 48 dB(A), à cause de l'escalier mécanique.

Salles électrique et mécanique

Quatre salles électriques, trois salles mécaniques et une salle des serveurs ont été mesurées pendant les séances de laboratoire du projet. Le niveau de bruit dans ces locaux (voir Tableau 2) est évidemment plus élevé que celui des espaces d'apprentissage, mais n'est pas très élevé au regard de la réglementation québécoise sur le bruit au travail [24] dont la limite d'exposition quotidienne au bruit ($L_{ex,8h}$) est fixée à 85 dB(A).

Tableau 2 : Niveau de bruit en dB(A) dans les salles électrique, mécanique et des serveurs.

Type de local	Nombre de locaux	L_{Aeq} (dB(A))
Salle électrique	4	58 < L_{Aeq} < 73
Salle mécanique	3	67 < L_{Aeq} < 69
Salle des serveurs	1	57

Néanmoins, les bruits générés dans ces locaux ont une signature spectrale caractéristique avec de l'énergie concentrée à certaines fréquences, comme les montre la Figure 7 (voir les courbes bleues des Figures 7(a), 7(b) et 7(c)). Ces bruits sont perçus dans les locaux voisins (voir les courbes rouges et jaunes des Figures 7(a), 7(b) et 7(c)), et même s'ils sont de faible amplitude, peuvent gêner les personnes qui y travaillent. Ces bruits sont principalement des « bourdonnements » électriques (« ***electric hum*** » en anglais) générés par le noyau de transformateurs et caractérisés par une énergie acoustique importante au double de la fréquence du réseau (2×60 = 120 Hz) et à ses harmoniques. Un transformateur a d'ailleurs été identifié avec la caméra acoustique comme une des sources principales de bruit d'un local électrique comme le montre la Figure 6(c). Les spectres de bruit de la Figure 7 présentent aussi une composante importante à 60 Hz (et ses harmoniques, dont 180 Hz) et qui peut être perçue dans les locaux voisins.

Une solution pour réduire le bruit de ces équipements est l'encoffrement acoustique [25]. Un encoffrement permet d'isoler l'équipement bruyant de l'environnement acoustique externe et doit avoir des parois internes absorbantes afin de réduire aussi l'énergie acoustique dans la cavité interne formée par l'encoffrement. La section suivante présente un métamatériau acoustique destiné à être utilisé comme matériau

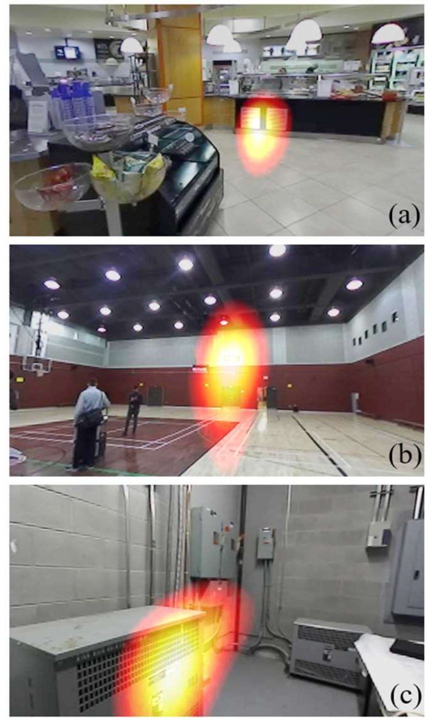


Figure 6 : Images acoustiques de trois locaux : (a) cafétéria du pavillon A, (b) gymnase du pavillon B et (c) salle électrique. Le centre de la tâche colorée indique la position de la source acoustique dominante dans le local.

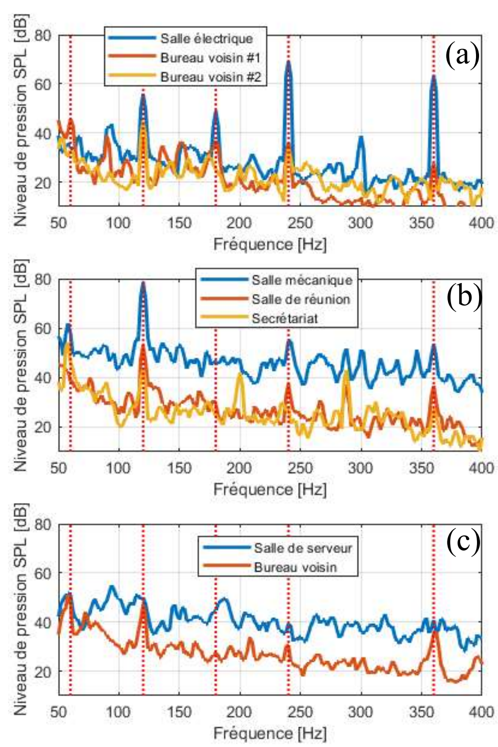


Figure 7 : Spectre des niveaux de pression en dB dans (a) une salle électrique et deux bureaux voisins, (b) une salle mécanique et deux locaux voisins (une salle de réunion et un secrétariat), (c) une salle des serveurs et un bureau voisin. Les traits rouges discontinus verticaux sont placés aux fréquences 60 Hz, 120 Hz, 180 Hz, 240 Hz et 360 Hz.

constitutif de l'encoffrement des transformateurs. Une solution de type métamatériau est privilégiée car les matériaux acoustiques conventionnels sont inefficaces à absorber l'énergie en si basses fréquences, ici $f < 400$ Hz.

4.3 Métamatériau acoustique pour la réduction du bruit des transformateurs électriques

La section précédente a permis de déterminer plusieurs fréquences pour lesquelles l'énergie des sources de bruit dans les locaux électrique et mécanique est importante et nécessite d'être traitées. Particulièrement les fréquences visées dans ce projet sont : 60 Hz, 120 Hz, 180 Hz et 360 Hz. Le QR a été accordé afin d'absorber l'énergie acoustique à la fréquence de 120 Hz. Pour cela, sa longueur est de $L_{\mathrm{QR}} = 702{,}3$ mm (voir Eqs. (A3) et (A4) de l'annexe). Ce résonateur étant efficace à sa fréquence fondamentale et aux multiples impairs de celle-ci, il permet d'absorber l'énergie à la fréquence de 360 Hz (3×120 Hz). Le diamètre permettant la meilleure absorption à ces deux fréquences a été déterminé par essai-erreur à $r_{QR} = 15$ mm.

L'absorption de l'énergie acoustique aux autres fréquences de 60 et 180 Hz a été réalisée par le HR. Les dimensions du HR ont été déterminées par un processus d'optimisation avec un algorithme génétique dont la fonction coût est la suivante :

$$\varepsilon = |f_{pic,1} - 60| - |f_{pic,i+1} - 180|, \qquad (3)$$

où $f_{pic,i}$ est la fréquence du $i$ème pic d'absorption. Une contrainte sur la valeur du coefficient d'absorption des pics a été fixée à 0,8. Les valeurs des caractéristiques géométriques du multi-résonateur de la Figure 3(b) sont indiquées dans le Tableau 3 ci-dessous. La seule valeur imposée était celle de l'épaisseur des parois : $e_{\mathrm{paroi}} = 5$ mm.

Tableau 3 : Paramètres de la géométrie (en mm) du multi-résonateur de Helmholtz.

$r_{\mathrm{col},1}$	$r_{\mathrm{col},2}$	$h_{\mathrm{col},1}$	$h_{\mathrm{col},2}$	$r_{\mathrm{cav},1}$	$r_{\mathrm{cav},2}$	$h_{\mathrm{cav},1}$	$h_{\mathrm{cav},2}$
6	7	7	45	83	72	40	40

Un modèle numérique de la géométrie d'une cellule unitaire du métamatériau ainsi optimisé est présenté à la Figure 8. Le QR est enroulé en spirale afin de minimiser l'épaisseur totale du matériau [26]. La disposition des deux résonateurs (c.-à-d., QR enroulé autour du HR) a été pensée pour être imprimée en 3D. Cette conception permettrait de réduire le gaspillage de matériau et la complexité d'usinage.

La Figure 9 représente le coefficient d'absorption en fonction de la fréquence de la cellule unitaire du métamatériau proposé. Les quatre pics d'absorption se situent aux fréquences cibles et leur amplitude est supérieure à 0,95.

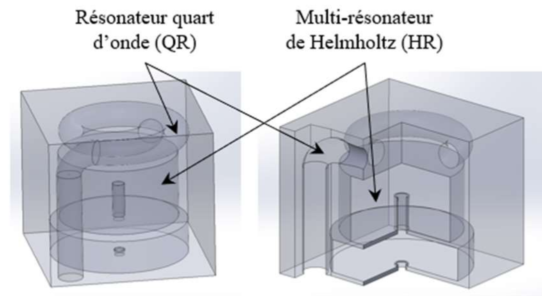


Figure 8 : Cellule unitaire du métamatériau : **(a)** vue isométrique, **(b)** vue en coupe.

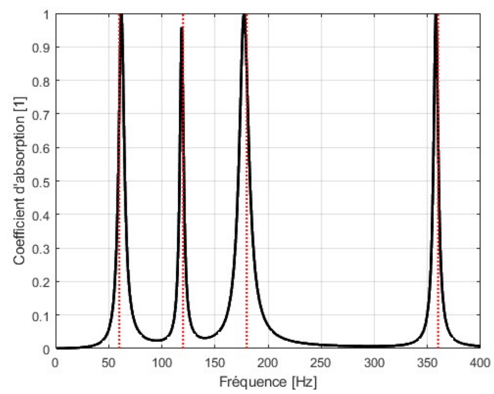


Figure 9 : Coefficient d'absorption du métamatériau excité sous onde plane d'incidence normale. Les traits rouges discontinus verticaux sont placés aux fréquences 60, 120, 180 et 360 Hz.

Conclusion

Cet article avait pour principal objectif de quantifier et évaluer le bruit extérieur et intérieur du campus universitaire de l'ÉTS à Montréal. Tous les travaux ont été réalisés par les étudiants du cours « Acoustique industrielle » de l'ÉTS (MEC636) dans le cadre de leur projet de session. L'objectif pour eux étaient double : (i) appliquer les connaissances théoriques et expérimentales apprises pendant le cours et (ii) les rendre utiles à la communauté de l'ÉTS et aux résidents et résidentes du quartier. La première étape du projet a permis de réaliser des cartes de bruit du campus et d'identifier les zones les plus calmes et les plus bruyantes. Sans surprise, ce campus de centre-ville est assez bruyant avec des niveaux de bruit dépassant les recommandations de l'OMS (supérieur à 55 dB(A)) sur la majorité de sa surface. Les niveaux de bruit sont plus élevés en journée (avant 16h) à cause des nombreux travaux de construction dans le quartier. Néanmoins, certains secteurs plus calmes ont pu être identifiés et se trouvent dans les cours piétonnes extérieures entre les pavillons et résidences universitaires. La seconde étape du projet a permis d'évaluer la qualité acoustique de plusieurs locaux intérieurs des différents pavillons de l'ÉTS (ex., salles de classe, bureaux, bibliothèque). De manière générale, les salles de classe mesurées sont tout à fait adéquates pour l'apprentissage (c.-à-d., bruit de fond inférieur à 40 dB(A) et temps de réverbération inférieurs à 0,7 s). En revanche, 50 % des espaces d'apprentissages auxiliaires dépassent les 45 dB(A) (ex., gymnase, cafétéria, atrium, espace collaboratif) et mériteraient des correctifs acoustiques pour améliorer le confort acoustique de la communauté étudiante. La troisième et dernière étape du projet s'est concentrée sur le bruit des salles électrique et mécanique des pavillons du campus. Ces locaux émettent des bruits qui ont une signature tonale caractéristique et qui peuvent être perçus dans les locaux adjacents et déranger le personnel qui y travaille. Une solution est d'encoffrer les sources de bruit de ces locaux. Les étudiants du cours ont alors localisé les principales sources à l'aide d'une caméra acoustique et conçu un métamatériau acoustique dédié à l'encoffrement de ces dernières. Une simulation du comportement acoustique du métamatériau proposé a permis de montrer que le matériau serait capable d'absorber l'énergie acoustique à quatre fréquences identifiées comme problématiques dans les spectres de bruit mesurés dans ces salles et bureaux voisins.

Ce travail de projet de session du cours MEC636 comporte bien évidemment plusieurs limitations, qui ouvriront sur les perspectives à réaliser dans les années à venir avec d'autres groupes d'étudiants et étudiantes du même cours. Concernant les mesures extérieures, il faudrait les répéter pour les deux périodes étudiées, et ce, à plusieurs moments de l'année (seules des mesures en hiver ont été réalisées pour le moment) afin d'obtenir des niveaux de bruit moyens plus représentatifs. L'application « NoiseCapture »[13] est tout à fait adaptée pour cela puisqu'elle se base sur une approche collaborative pour la production des données. De plus, cet outil permet d'intégrer la mesure de bruit au téléphone cellulaire dans le cadre du cours et donc de pouvoir insister sur les différents mécanismes nécessaires à l'obtention de mesures de qualité avec ce type d'appareil (ex., calibration). Le nombre de mesures est aussi une limitation dans la campagne de mesure du bruit intérieur présentée dans ce papier. Des mesures d'une durée plus longue et pour plus de locaux devront être réalisées dans le futur. D'autres indicateurs pourraient aussi être calculés et utilisés pour analyser la qualité acoustique des environnements intérieurs [20, 27]. Des mesures subjectives, au travers de questionnaires, pourraient aussi venir compléter les mesures objectives réalisées avec des microphones en se basant sur les avancées de la recherche sur la caractérisation et l'analyse des paysages sonores (« ***soundscape*** » en anglais) [28-30]. Ces mesures permettraient de caractériser la manière dont les environnements (extérieurs et intérieurs) du campus sont perçus par les gens et de mieux guider la recherche de solutions pour offrir des environnements acoustiques plus confortables. En ce qui concerne le métamatériau, il resterait à fabriquer une cellule afin de valider expérimentalement le concept puis de concevoir un encoffrement réalisé à l'aide du pavage de plusieurs cellules et de vérifier son efficacité *in situ*.

Remerciements

Les auteurs souhaiteraient remercier les responsables du projet Noise-Planet pour leurs conseils et la création des deux « NoiseCapture party » qui ont permis de réaliser les cartes des bruits des deux périodes de mesure. De plus, nous aimerions remercier la direction de l'ÉTS pour avoir soutenu ce projet mais aussi les différents membres du personnel qui nous ont ouvert les portes de leur lieu de travail. Nous souhaitons tout particulièrement remercier Magdalena Stanescu, ingénieure en mécanique et électricité du bâtiment de l'ÉTS, pour son temps et son accompagnement pendant les laboratoires du projet ainsi que Nathalie Desgagné, directrice adjointe au Bureau de la prévention et de la sécurité, pour les fructueuses discussions.

Annexe : Modélisation des résonateurs du métamatériau par la méthode des matrices de transfert.

Propagation dans les couches d'air, prise en compte des dissipations visco-thermiques et rayonnement aux embouchures.

La propagation des ondes acoustiques dans une couche d'air d'épaisseur L est prise en compte à l'aide de la matrice de transfert suivante :

$$T_{air} = \begin{bmatrix} \cos(\tilde{k}_0 L) & j\tilde{Z}_0 \sin(\tilde{k}_0 L) \\ \dfrac{j\sin(\tilde{k}_0 L)}{\tilde{Z}_0} & \cos(\tilde{k}_0 L) \end{bmatrix}, \quad \text{(A1)}$$

avec $\tilde{Z}_0$ l'impédance spécifique de l'air (Pa.s.m^{-1}), $\tilde{k}_0$ le nombre d'onde dans l'air. $\tilde{Z}_0$ et $\tilde{k}_0$ sont complexes et dépendent de la fréquence afin de tenir compte les dissipations visco-thermiques dans les cols et cavités. Pour cela, le modèle de Qunli est utilisé [31] en calculant au préalable la résistivité au passage à l'air des différentes portions de conduit (ex., col, cavité) à l'aide de leur rayon r selon [12]:

$$\sigma_0 = \frac{8\eta}{r^2}, \quad \text{(A2)}$$

où η la viscosité dynamique de l'air (Pa.s).

Le rayonnement aux embouchures (i.e., cols du HR et entrée du QR) a été considéré en appliquant une correction sur longueur géométrique de la couche d'air qui rayonne dans un espace d'air de plus grande surface. La correction de longueur est donnée par :

$$L' = L + n \cdot 0{,}82 \cdot r\,, \quad \text{(A3)}$$

où n=1 si la couche n'a qu'une seule de ses deux extrémités qui rayonne (i.e., QR), ou n=2 si ses deux extrémités rayonnent (i.e., cols du HR).

Résonateur quart d'onde (QR)

Le résonateur quart d'onde a été modélisé à l'aide d'une couche d'air, $T_{\mathrm{QR}} = T_{\mathrm{air}}$, dont la longueur corrigée est fixée par la fréquence visée f :

$$L'_{\mathrm{QR}} = \frac{c_0}{4f}, \quad \text{(A4)}$$

où $f = 120$ Hz et c_0 est la célérité du son dans l'air (m.s^{-1}).

À partir de la matrice de transfert T_{QR}, l'impédance d'entrée du résonateur quart d'onde est déterminée par :

$$Z_{\mathrm{QR}} = \frac{T_{\mathrm{QR},11}}{T_{\mathrm{QR},21}}. \quad \text{(A5)}$$

Multi-résonateur de Helmholtz

Les changements de section (CS) du multi-résonateur de Helmholtz sont pris en compte à l'aide de la matrice suivante :

$$T_{\mathrm{CS}} = \begin{bmatrix} 1 & 0 \\ 0 & \frac{S_{\mathrm{s}}}{S_{\mathrm{e}}} \end{bmatrix}, \qquad \text{(A6)}$$

avec S_{e}, la surface de la section en amont du changement de section et S_{s}, la surface de la section en aval du changement de section.

Chaque col du multi-résonateur est prolongé dans la cavité. Chaque cavité du multi-résonateur a alors été séparée en deux parties, délimitées par le trait gris discontinu sur la Figure 3(b). Les parties situées avant le col peuvent être prises en compte à l'aire de résonateurs quart d'onde de longueur $\left(h_{\mathrm{col},i} - e_{\mathrm{paroi}}\right)$ (avec $i = 1{,}2$ pour les cavités 1 et 2). Ces résonateurs, situés en dérivation par rapport à la direction de propagation dans l'épaisseur du HR, ont été modélisés à l'aide la matrice suivante :

$$T_{\mathrm{res},i} = \begin{bmatrix} 1 & 0 \\ \frac{1}{Z_{\mathrm{res},i}} & 1 \end{bmatrix}, \qquad \text{(A7)}$$

avec $Z_{\mathrm{res},i}$ l'impédance d'entrée acoustique de chaque résonateur quart d'onde des cavités $i = 1{,}2$ et qui est donné par :

$$Z_{\mathrm{res},i} = \frac{Z_0}{j \tan\left(k_0\left(h_{\mathrm{col},i} - e_{\mathrm{paroi}}\right)\right)}. \qquad \text{(A8)}$$

La matrice de transfert totale du multi-résonateur de Helmholtz est :

$$T_{\mathrm{multi}} = T_{\mathrm{air},1} \cdot T_{\mathrm{CS},1} \cdot T_{\mathrm{res},1} \cdot T_{\mathrm{CS},2} \cdot T_{\mathrm{air},2} \cdot T_{\mathrm{CS},3} \cdot T_{\mathrm{air},3} \cdot T_{\mathrm{CS},4} \cdot T_{\mathrm{res},2} \cdot T_{\mathrm{CS},5} \cdot T_{\mathrm{air},4}\,. \qquad \text{(A9)}$$

Le calcul de la matrice du multi-résonateur de Helmholtz T_{multi} permet de trouver son impédance d'entrée à partir de ses composantes T_{11} et T_{21}selon :

$$Z_{\mathrm{HR}} = \frac{T_{11-\mathrm{multi}}}{T_{21-\mathrm{multi}}}. \qquad \text{(A10)}$$

Références

[1] World Health Organization - WHO. Guidelines for Community Noise. Geneva, 2019.

[2] R. Martin, P. Deshaies, M. Poulin. Avis sur une politique québécoise de lutte au bruit environnemental : pour des environnements sonores sains. Québec : INSPQ, 2016.

[3] T. B. de Souza, K. C. Alberto, S. A. Barbosa. Evaluation of noise pollution related to human perception in a university campus in Brazil. Applied Acoustics157,107023, 2020. https://doi.org/10.1016/j.apacoust.2019.107023.

[4] P. H. T. Zannin, M. S. Engel, P. E. K. Fiedler, F. Bunn. Characterization of environmental noise based on noise measurements, noise mapping and interviews: A case study at a university campus in Brazil. Cities 31,317–327, 2013. https://doi.org/10.1016/j.cities.2012.09.008.

[5] D. Çolakkadıoğlu, M. Yücel, B. Kahveci, Ö . Aydınol. Determination of noise pollution on university campuses: a case study at Çukurova University campus in Turkey. Environ Monit Assess 190,203, 2018. https://doi.org/10.1007/s10661-018-6568-8.

[6] M. S. Ragettli, S. Goudreau, C. Plante, M. Fournier, M. Hatzopoulou, S. Perron, A. Smargiassi. Statistical modeling of the spatial variability of environmental noise levels in Montreal, Canada, using noise measurements and land use characteristics. Journal of Exposure Science and Environmental Epidemiology 26, 597–605, 2016.

[7] Vivre en ville. Gestion intégrée du bruit environnemental : Trousse d'outils pour un climat sonore agréable., Collection Passer à l'Action, 26 p. vivreenville.org, 2020.

[8] Comité Campus ÉTS. Rapport final : Consultations pour le développement urbanistique du Campus ÉTS. https://www.etsmtl.ca/ETS/A-propos/developpement-campus-ets, 2018.

[9] R. Martin. Vision et orientations gouvernementales en matière de lutte contre le bruit environnemental au Québec. Québec, Canada: Ministère de la Santé et des Services sociaux, 2019.

[10] F. Laville, O. Doutres, J. Voix. MEC636 : Acoustique industrielle », Notes de cours, École de technologie supérieure, 2023.

[11] F. A. Laville. Une expérience d'enseignement de l'acoustique industrielle intégrant pédagogie de la coopération, projet de session et laboratoire informatique [A teaching experience in the industrial acoustics integrating pedagogy of cooperation, computer laboratory and session project]. Canadian Acoustics.;40(2),11–14, 2012.

[12] J.-F. Allard, N. Atalla. Chapter 11: Modelling multilayered systems with porous materials using the transfer matrix method. Propagation of sound in porous media: modelling sound absorbing materials (2nd ed., pp. 243-281). Chichester: John Wiley & Sons, 2009.

[13] J. Picaut , N. Fortin, E. Bocher, G. Petit, P. Aumond, G. Guillaume. An open-science crowdsourcing approach for producing community noise maps using smartphones. Building and Environment. 148,20–33, 2019. https://doi.org/10.1016/j.buildenv.2018.10.049.

[14] G.Guillaume, P. Aumond , E. Bocher, Can A., Écotière D., Fortin N., et al. NoiseCapture smartphone application as pedagogical support for education and public awareness. The Journal of the Acoustical Society of America. 151(5), 3255–3265, 2022. https://doi.org/10.1121/10.0010531

[15] Mecanum. (n.d.). LF-ANT | Caméra acoustique 3D. https://mecanum.com/fr/measuring-instruments/acoustic-camera/ (accessed May 1, 2023).

[16] J. Frechette-Viens, N. Quaegebeur, N. Bastien. A low-latency acoustic camera for transient noise source localization. Berlin Beamforming Conference.Berlin, 2020.

[17] American National Standard. Acoustical Performance Criteria, Design Requirements, and Guidelines for Schools, Part 1: Permanent Schools. ANSI/ASA S12.60-2010/Part 1 (R2015). Melville, NY : Acoustical Society of America, 2015.

[18] ASHRAE handbook, chap. 48 « Noise and vibration control», 2011.

[19] J. Romeu, M. Genescà, T. Pàmies, S. Jiménez. Street categorization for the estimation of day levels using short-term measurements, Applied Acoustics, 72(8), 569-577, 2011. https://doi.org/10.1016/j.apacoust.2010.09.012.

[20] G. Minelli, G. E. Puglisi, A. Astolfi. Acoustical parameters for learning in classroom: A review. Building and Environment.208,108582, 2022. https://doi.org/10.1016/j.buildenv.2021.108582.

[21] K. Verdière, R. Panneton, S. Elkoun, T. Dupont, P. Leclaire. Comparison between parallel transfer matrix method and admittance sum method. The Journal of the Acoustical Society of America136(2) : EL90–EL95, 2014. https://doi.org/10.1121/1.4885481.

[22] N. Sharafkhani. An ultra-thin multi-layered metamaterial for power transformer noise absorption. Building Acoustics.29,53-62, 2022. https://doi.org/10.1177/1351010X211041704.

[23] A. Gilchrist, E. N. Allouche, D. Cowan. Prediction and mitigation of construction noise in an urban environment. Canadian Journal of Civil Engineering. 30,659–672, 2003. https://doi.org/.1139/l03-019.

[24] Gouvernement du Québec. Loi sur la santé et la sécurité du travail. Décret 781-2021. Gazette officielle du Québec, 153(24), 2722-2733, 2021.

[25] D. K. Meinke, E. H. Berger, D. P. Driscoll, R. L. Neitzel, K. Bright. Chapter 10 : Noise Control Engineering. In The noise manual. (6th edition). Falls Church, VA: American Industrial Hygiene Association, 2022.

[26] G. Catapane, G. Petrone, O. Robin, K. Verdière. Coiled quarter wavelength resonators for low-frequency sound absorption under plane wave and diffuse acoustic field excitations. Applied Acoustics, 2023. 209,109402. https://doi.org/10.1016/j.apacoust.2023.109402.

[27] Astolfi A., Minelli G., Puglisi G. E. A basic protocol for the acoustic characterization of small and medium-sized classrooms. The Journal of the Acoustical Society of America. 152(3),1646–1659, 2022.

[28] International Organization for Standardization. Acoustics — Soundscape — Part 2: Data collection and reporting requirements. ISO/TS 12913-2:2018, 2013. https://www.iso.org/standard/75267.html

[29] F. Aletta, T. Oberman, Ö .Axelsson, H. Xie, Z. Yuan, S. K. Lau, et al. Soundscape assessment: towards a validated translation of perceptual attributes in different languages. In: INTER-NOISE and NOISE-CON Congress and Conference Proceedings. Seoul; p. 3137–3146, 2020.

[30] S. Mancini, A. Mascolo, G. Graziuso, C. Guarnaccia. Soundwalk, Questionnaires and Noise Measurements in a University Campus: A Soundscape Study. Sustainability. 13(2),841, 2001.

[31] W. Qunli. Empirical relations between acoustical properties and flow resistivity of porous plastic open-cell foam. Applied Acoustics.25,141–148, 1998. https://doi.org/10.1016/0003-682X(88)900 90-4.

Assessment of Noise in the Campus of École de Technologie Supérieure in Montréal and Proposal of an Acoustic Metamaterial for the Reduction of Electrical Transformer Noise

Olivier Doutres *1, Maël Lopez †1, Kévin Rouard ‡1, Louis-Philippe Campagna ♦1, Titouan Cougoulic ♯1, Anthony Jutras ♠1, David Lauzon ♣1, Pierre-Luc Pépin-Pagé ⸸1 et Alexis Purson §1
[1]Department of Mechanical Engineering, École de technologie supérieure (ÉTS), Montréal, Québec, Canada.

Résumé

Les campus universitaires situés au cœur des villes peuvent être exposés à des niveaux de bruit importants qui peuvent nuire à l'apprentissage des étudiants et étudiantes, à la performance de la communauté enseignante et des membres du personnel mais aussi à la qualité de vie des résidents et résidentes du quartier. Cet article présente les résultats d'un projet d'étude réalisé par des étudiants du cours d'acoustique industrielle de l'École de technologie supérieure (ÉTS) et qui avait pour principal objectif de quantifier et évaluer le bruit extérieur et intérieur du campus universitaire de l'ÉTS. Les étudiants devaient aussi localiser les principales sources de bruit intérieures à l'aide d'une caméra acoustique et modéliser un métamatériau acoustique qui permettra de réduire le bruit tonal émis par les transformateurs en basses fréquences.

Mots clés : bruit environnemental, acoustique, université, campus, bruit intérieur, métamatériau acoustique

Abstract

University campuses located in the heart of cities can be exposed to significant noise levels that can hinder the learning of students, the performance of the teaching community and staff, as well as the quality of life for residents in the neighborhood. This article presents the results of a study project conducted by students of the "Industrial Acoustics" course at École de technologie supérieure (ÉTS), with the main objective of quantifying and assessing the outdoor and indoor noise levels on the university campus. The students were also tasked with identifying the main sources of indoor noise using an acoustic camera and modeling an acoustic metamaterial that would help reduce the tonal noise emitted by transformers at low frequencies.

Keywords: environmental noise, acoustics, university, campus, indoor noise, acoustic metamaterial

1 Introduction

Noise can have detrimental effects on individuals' health [1]: hearing loss, sleep disruption, difficulty in communication, cardiovascular and psychophysiological effects, reduced performance, discomfort, and impacts on social behavior. In educational settings, noise can also affect learning, particularly reading comprehension, memory, and speech intelligibility [2]. As a result, several studies have focused on noise in university campuses[3-5], generally concluding that noise levels are too high for an environment dedicated to learning. The campus of the École de technologie supérieure (ÉTS) is located in the heart of Montreal, Canada, and is, unsurprisingly, exposed to high noise levels as shown in the noise level mapping conducted in 2014 by Ragettli et al. [6, 7] and presented in Figure 1(a). According to this map, noise levels in this area exceed the maximum recommended level of 55 dB(A) by the World Health Organization (WHO) [1] (recommendation for outdoor spaces in schools). The noise pollution in this central area of Montreal is indeed a real issue, and the campus noise has been mentioned multiple times during a consultation on the campus urban development conducted in 2018 [8]. For example, some suggestions arising from these consultations include "*Creating relaxing soundscapes*," "*Designing green walls to counteract pollution and noise*," "*Building havens of peace (mitigate noise pollution) open to the public but intended for ÉTS employees and students, and maintaining a balance between the needs of the ÉTS community and the neighborhood residents*." However, the precise noise levels in different parts of the ÉTS campus are not known. A more detailed mapping would help identify (i) the quietest areas that would be most suitable for outdoor rest, as desired by the community, and (ii) the noisiest areas that would require acoustic improvements to enhance the comfort of the neighborhood residents and the ÉTS community.

The indoor acoustic environments of the different buildings are equally important. They need to be adapted to the learning context but also conducive to office work for all campus staff members. Just like outdoor noise, the indoor noise levels at the ÉTS campus are not known and need to be measured.

* olivier.doutres@etsmtl.ca
† mael.lopez.1@ens.etsmtl.ca
‡ kevin.rouard.1@ens.etsmtl.ca
♦ louisphilippe48@hotmail.com
♯ cougoulic.titouan@gmail.com
♠ anthonyjutras@hotmail.com
♣ david_lauzon@hotmail.com
⸸ p-l.pp@hotmail.com
§ alexis.purson@hotmail.fr

As part of their semester project for the "Industrial Acoustics" course at ÉTS [10] (course code MEC636), a class had the mission to conduct a noise study on the ÉTS campus to contribute to improving the acoustic environments and thus the quality of life for the community. The first step of the project involved studying the outdoor noise on the ÉTS campus. The second step involved characterizing the sound environments of several rooms in different buildings at ÉTS (e.g., auditorium, classrooms, cafeteria, offices, library) and evaluating the acoustic quality of these environments. The noise sources of interest in this project were stationary sources associated with the operation of the buildings, such as ventilation, mechanical and electrical systems, and computer servers. Lastly, the students were required to locate the main sources of noise inside the buildings, including those from electrical and mechanical rooms. They also had to propose a concept for an acoustic metamaterial dedicated to enclosing electrical transformers to reduce their potential impact on adjacent rooms.

The purpose of this paper is to present the noise study of the ÉTS campus conducted by students of the Industrial Acoustics course at ÉTS as part of their semester project. The educational context of this student project is initially presented in Section 2. Section 3 then describes the measurement equipment used, the evaluated outdoor and indoor environments, and the indicators used to characterize their acoustic quality. Section 3 concludes with the presentation of the model used to simulate the acoustic behavior of the metamaterial intended for reducing transformer noise. Section 4 presents and discusses the results of the study. Section 5 summarizes the main conclusions and outlines the project's future prospects.

2 Pedagogical context of the project

The course "Industrial acoustics" (MEC636) is an advanced specialization course in the final year of the mechanical engineering bachelor's program at ÉTS. It aims to equip students with the skills to measure and reduce noise based on the theoretical foundations of industrial acoustics and associated experimental techniques. This course is primarily based on three unconventional pedagogical elements [11]: (i) an active pedagogical method based on cooperative learning, (ii) intensive use of computer tools through practical sessions and computer-based exams, and (iii) a team-based semester project.

The course spans 13 weeks of instruction. The semester project, which is the subject of this paper, consisted of three laboratory sessions and one practical session. The project started with the three laboratories in weeks 8, 9, and 10. The first laboratory aimed to conduct noise level mapping of the outdoor areas on the ÉTS campus. The following two laboratories focused on characterizing multiple indoor acoustic environments in the main buildings of the campus (Buildings A, B, D, and E, as shown on the map in Figure 1(b)). The semester project presented in this paper differs slightly from the projects of previous years, which focused on reducing the noise of small household equipment (e.g., kitchen blender, leaf blower, hairdryer) [11]. However, both types of projects

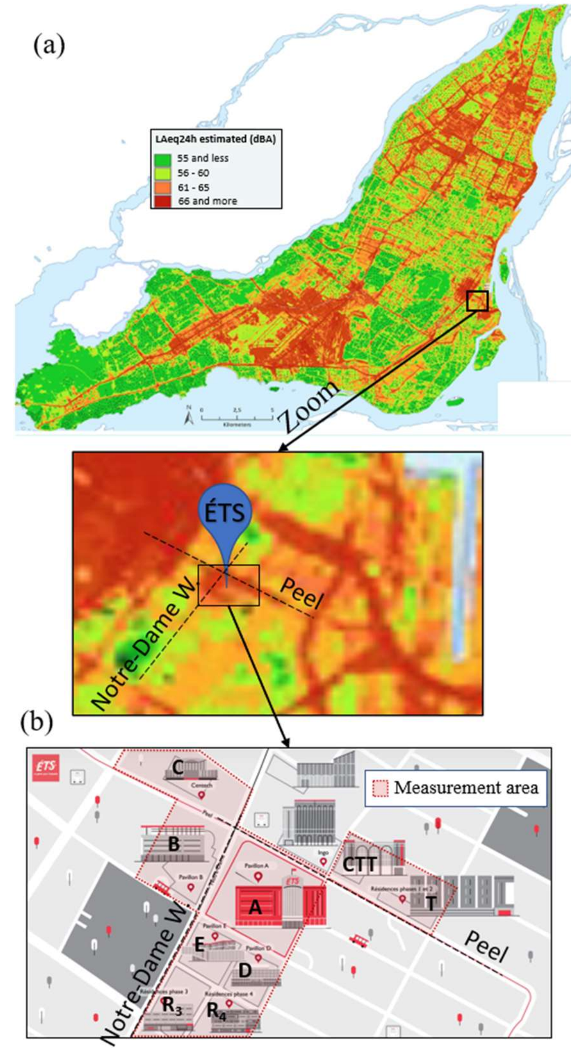


Figure 1: (a) Map of noise levels in the island of Montreal (adapted from [6, 7]); (b) ÉTS campus and outdoor noise measurement area.

allow students to apply the theoretical and experimental knowledge acquired during the course.

Prior to the first project laboratory, students were trained in acoustic diagnostics of environments and noise sources, including the use of instruments to measure sound pressure levels (overall noise level) and the representation of signals in the frequency domain (e.g., octave bands, narrow bands). Students had already conducted noise measurements, analyzed and interpreted the results in order to assess noise complaints (primarily in the workplace). After the 7th week of the course, they were taught the theoretical foundations of wave propagation in dissipative and non-dissipative fluids, as well as the transfer matrix method [10, 12]. The transfer matrix method is used in the MEC636 course to simulate the acoustic behavior in absorption and transmission of various noise reduction systems, such as single and multiple walls, as well as reactive and dissipative mufflers. The practical session of the semester project (in week 13 of the semester) allowed students to apply this knowledge. The objective of the session was to design an acoustic material composed of a paving of quarter-wavelength and Helmholtz resonators, also known as metamaterial, to absorb acoustic energy at targeted and problematic frequencies from noise sources identified during the indoor measurement campaigns on the campus.

3 Material and method

3.1 Material

Outdoor noise

The outdoor measurements were conducted using the NoiseCapture application [13, 14] installed on the students' mobile phones (i.e., Galaxy A23 5G, A52 5G, S20 FE 5G by Samsung from Seoul, South Korea, and Pixel 3A by Google from Mountain View, CA, USA) (see Figure 2(a)). This application allows for noise level measurements to be taken and combined with GPS data to display them on an interactive map within the application. The devices were manually calibrated just before the measurement session using a manual calibration procedure guided by the lab instructor. This procedure involved correcting the noise level obtained by the application through comparison with a simultaneous measurement using a calibrated sound level meter.

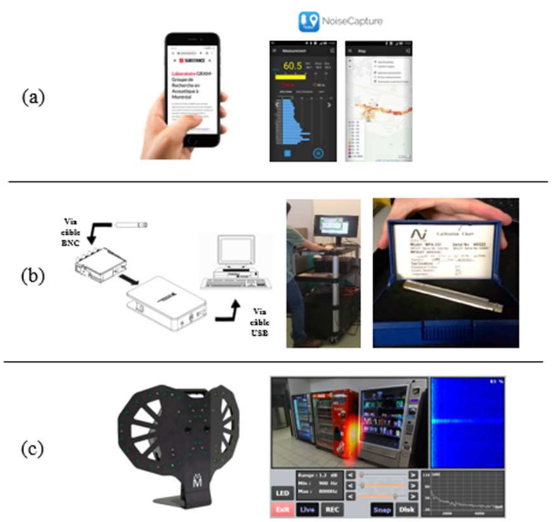


Figure 2 : (a) Mobile phone and "NoiseCapture" application [13, 14] for outdoor measurements; (b) instrumentation of the MEC636 course for indoor measurements; (c) LF-ANT acoustic camera (Mecanum, Sherbrooke QC, Canada) for acoustic imaging [15].

Indoor noise

The noise measurements in indoor environments were conducted using 1/2-inch free-field microphones (MPA231) of Class 1 from BSWA (Beijing, China), along with National Instruments (Austin, TX, USA) cDAQ-9171 data acquisition cards (see Figure 2(b)). The measurement chains were calibrated using a Larson Davis calibrator (Depew, NY, USA) CAL200. The "MEC636-V4" software, developed at ÉTS using LabVIEW (National Instruments, Austin, TX, USA), was used for data acquisition and post-processing. An LF-ANT acoustic camera (Mecanum, Sherbrooke, QC, Canada) [15, 16] (see Figure 2(c)) was also used to capture acoustic images of the environments and locate the main sources of noise. Similar to a thermal camera that shows hot spots of temperature, an acoustic camera reveals areas with the highest noise levels, allowing for visualizing sound. This equipment was purchased to complement the tools in the MEC636 course for the acoustic diagnosis of environments and noise sources. In the session project, this camera was used with the aim of improving the acoustic comfort of learning and working spaces on the campus by addressing the main noise sources in the buildings.

3.2 Environnements

Outdoor noise

The outdoor noise measurements were conducted in the designated area of the ÉTS campus highlighted in red on Figure 1(b). The main intersection of the campus is located at the corner of Notre-Dame West and Peel streets. The different zones within the campus indicated on the map are: (i) buildings A, B, D, and E, which include classrooms, offices, a library, auditoriums, conference rooms, cafeterias, a sports center, and a daycare center; (ii) Centech C, which is a technology incubator; (iii) student residences T, R_3 and R_4; and (iv) the thermal technology center (CTT).

Two measurement periods were conducted on the afternoon of February 22, 2023: (1) a first period from 2:30 PM to 3:30 PM, referred to as the "off-peak hour," and (2) a second period from 4:00 PM to 5:00 PM, referred to as the "peak hour." These two periods were chosen because the ÉTS campus is located near major roadways in Montreal, and significant differences in noise levels were expected between the two periods, with higher levels during peak hours.

Indoor noise

The indoor environmental measurements at ÉTS were conducted on February 23 and March 9, 2023. The measured locations were divided into three categories. The first two categories correspond to "**core learning spaces**" and "**ancillary learning spaces**" as defined in the ANSI/ASA S12.60 standard [17]. The first category includes open or enclosed teaching and learning spaces where oral communication is essential for students' academic achievement. This category partially encompasses classrooms, the library and auditoriums. The measurements were predominantly taken when the rooms were unoccupied and/or with quiet individuals present. The main sources of noise in these spaces were typically the ventilation and air conditioning systems. The second category comprises learning spaces where communication is crucial for the student but their primary function is not formal learning. Instead, they involve informal learning, social interactions, and similar activities. These spaces include common areas (e.g., atriums), cafeterias, sports facilities, and student life areas such as clubs. The third category corresponds to the "**electrical and mechanical rooms**" in various campus buildings, as well as adjacent rooms that may be impacted by the noise sources from these rooms.

3.3 Indicators and recommended maximum values

This section presents the different indicators used to characterize outdoor and indoor acoustic environments, as well as the recommended maximum values for the measured environments, taken from reference documents (e.g., WHO[1, 9], ANSI/ASA S.12.60 standard [17], ASHRAE handbook [18]).

Outdoor noise

The "NoiseCapture" application allows measuring the equivalent A-weighted sound level every second ($L_{Aeq,1s}$) while the recording is active and the student moves around the campus. The A-weighted sound pressure level approximates how the human ear perceives the different frequency components of sounds at typical speech listening levels. At the end of each measurement campaign ("off-peak hour" and "peak hour"), the application divides the space into hexagons with an equivalent radius of 15 meters. For each measurement campaign, the application combines all the measurements taken in each hexagon and provides an equivalent noise level L_{Aeq} per hexagon [13, 14]. The duration of the measurements taken during both campaigns ranged from 30 seconds to 5 minutes. The cumulative measurement time for all students was 1 hour and 57 minutes for the "off-peak hour" campaign and 1 hour and 55 minutes for the "peak hour" campaign.

Members of the ÉTS community who move outside the campus buildings are mostly exposed to road traffic noise. The maximum recommended exposure value (over 24 hours, $L_{Aeq,24h}$) by the WHO to prevent the effects of noise for sources related to road traffic (i.e., cardiovascular ischemic diseases; type 2 diabetes; annoyance, sleep disturbances, difficulty reading and oral comprehension) is 50 dB(A) [7, 9]. Although in practice, the measurements were taken for much shorter durations than 24 hours (for practical reasons), they can still be compared to a threshold value defined over 24 hours[6, 19]. Another more permissive limit value from the WHO of 55 dB(A) was recommended for outdoor environments of schools [1]. This maximum value is considered in this study as it has often been used in similar studies conducted on university campuses [3, 4].

Indoor noise

Two indicators are predominantly used to characterize the acoustic quality of learning spaces (core and ancillary) [17]: background noise level (A-weighted equivalent level, L_{Aeq}) and reverberation time (TR). Both indicators are measured when the rooms are unoccupied. Measuring the background noise level in a room allows assessing the magnitude of contributions from external noise sources (e.g., road traffic, air traffic, factories, activity in schoolyards) and internal noise sources (e.g., ventilation noise, noise from neighboring rooms). The reverberation time measures the extent of reverberation in a room and represents the time required for a continuous sound level to decay by 60 dB after being switched off. This time depends on the volume of the room, the absorption properties of the materials on the surfaces and the frequency. In this project, the reverberation time was measured in classrooms (category of core learning spaces), and noise level measurements were performed for durations of 10 to 15 seconds (due to time constraints associated with the limited duration of project-specific teaching laboratories).

Excessive background noise and/or reverberation in these spaces interfere with oral communication and constitute an "acoustic" barrier to learning [17]. Therefore, maximum recommended values are provided in reference works [17, 18] and are summarized in Table 1 below.

Table 1: Maximum recommended values for core and ancillary learning spaces.

Category	Type of space	L_{Aeq} max (dB(A))	TR (s) in octave bands 500, 1000 and 2000 Hz
Core learning spaces	Classrooms, library, private offices, conference rooms, music practice rooms.	35 (volume ≤ 566 m^3) [17] 40 (volume > 566 m^3) [17]	0,6 (volume < 283 m^3) [17] 0,7 (283 m^3 <volume ≤ 566 m^3) [17]
	Classrooms (100 m^3 < volume ≤ 290 m^3)	40 for students aged 12 and older [20]	0,6<TR<0,7 [20]
Ancillary learning spaces	Cafeteria	40 [17]	
	Gymnasium	40 [17] 50 [18]	
	Open-plan offices	45 [18]	
	Large capacity spaces with speech amplification	55 [18]	

The maximum recommended values depend on the use of the rooms. The acoustic quality of a core learning space should be higher than that of an ancillary learning space, and therefore the recommended maximum values for the former are lower. There is a wealth of literature available specifically for classrooms, as this space is of utmost importance for oral communication and student learning. A literature review on this topic [20] concludes that, for small and medium-sized classrooms, a reverberation time (TR) between 0.6 and 0.7 is adequate for students of all ages, and the background noise level should not exceed 40 dB(A) for students aged 12 and older.

3.4 Acoustic metamaterial modeling

In order to reduce the low-frequency noise from electrical and mechanical rooms that can be perceived in neighboring spaces (see Section 4.2), an acoustic metamaterial has been proposed. This material will serve as an acoustic enclosure for the main noise source in the room, identified using the acoustic camera. The metamaterial consists of a tiling pattern of an absorptive unit cell composed of a two-degree-of-freedom Helmholtz Resonator (HR) and a quarter-wavelength resonator (QR), as shown in Figure 3(a). The absorption behavior of the material has been modeled using the transfer matrix method, considering normal incidence plane wave excitation at the material surface. In this case, a single cell is sufficient for modeling (see Figure 3(b)). The resonators are designed to absorb acoustic energy at four identified problem frequencies.

The absorption coefficient of the unit cell is determined based on the input acoustic impedance Z and the characteristic impedance of air Z_0:

$$\alpha = 1 - \left|\frac{Z - Z_0}{Z + Z_0}\right|^2 . \quad (1)$$

The input impedance of the surface of the metamaterial unit cell, Z, is calculated based on the input acoustic impedances of the quarter-wavelength resonator, Z_{QR}, and the Helmholtz resonator, Z_{HR}, using the admittance sum method [21, 22]:

$$Z = \left(\frac{S_{\mathrm{t}}}{S_{\mathrm{QR}}}\frac{1}{Z_{\mathrm{QR}}} + \frac{S_{\mathrm{t}}}{S_{\mathrm{HR}}}\frac{1}{Z_{\mathrm{HR}}}\right)^{-1}, \quad (2)$$

with S_{t} the total surface area of the unit cell, S_{QR} the input surface area of the quarter-wavelength resonator, and S_{HR} the input surface area of the Helmholtz resonator. The transfer matrix modeling of the acoustic impedances of the two resonators (Z_{QR} and Z_{HR}) is presented in the appendix.

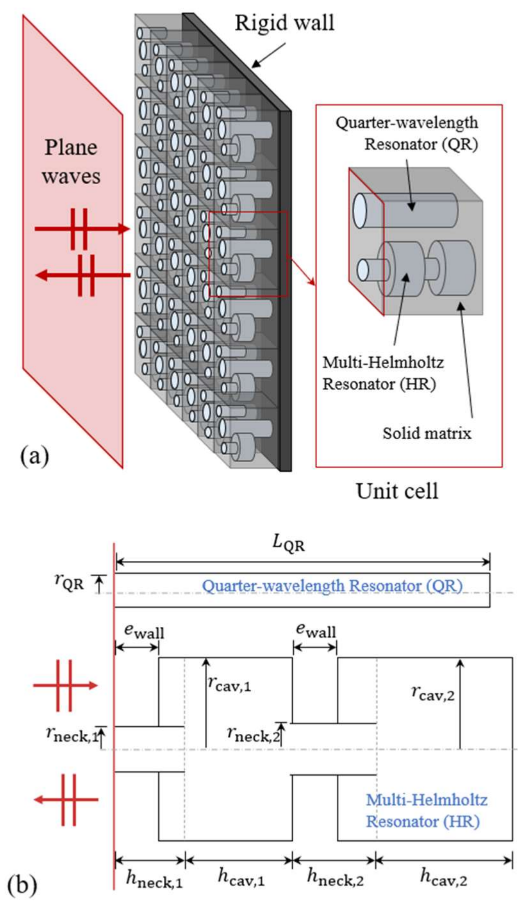


Figure 3: Acoustic metamaterial; (a) three-dimensional schematic, (b) cross-sectional view of a unit cell.

4 Results and discussion

4.1 Outdoor noise

The noise maps of the two outdoor measurement campaigns, "off-peak hour" and "peak hour," are presented in Figures 4(a) and 4(b), respectively. Overall, for both measurement periods, the trends shown on the noise level map in Figure 1(a) (see zoom) are observed: (i) the areas most exposed to noise are Notre-Dame West Street and Peel Street (as well as the area around Building C), and (ii) the noise levels in these areas are generally above 65 dB(A). A large part of the ÉTS campus is therefore exposed to levels well above those recommended by the WHO.

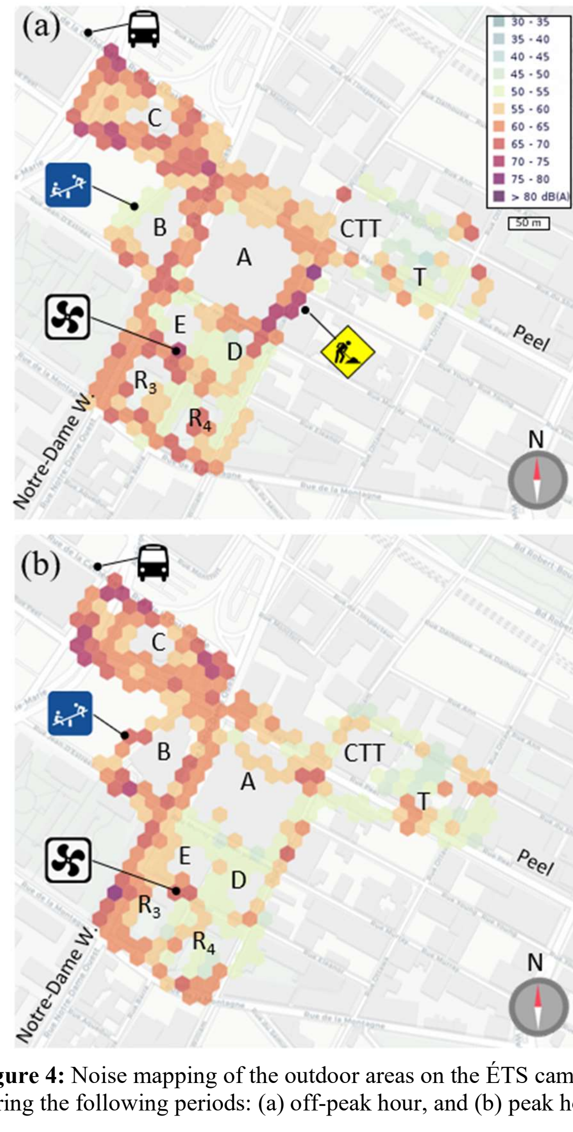


Figure 4: Noise mapping of the outdoor areas on the ÉTS campus during the following periods: (a) off-peak hour, and (b) peak hour.

The main source of noise on the ÉTS campus is road traffic. In the case of the area around Building C, the high noise levels could be attributed to the departure of buses from a station located slightly to the north (not visible in the figure), which use the street north of Zone C to access, among other things, a highway (see pictogram on Figure 4). However, Figure 4 shows that, counterintuitively, noise levels seem to be higher during off-peak hours than during peak hours. This can be explained by the significant number of constructions works in the area, generating both construction noise and road noise (dump trucks' noise being particularly high [23]), mainly before 4:00 PM. For example, a high noise level is observed only during off-peak hours on the street along the southeast side of Building A, caused by construction works for a new ÉTS building (see yellow pictogram on Figure 4(a)).

Four “quieter” areas (L_{Aeq} below 55 dB(A)) can be identified on both noise maps of the ÉTS campus, corresponding to outdoor courtyards: (i) between buildings E and D, (ii) between residences R_3 and R_4, (iii) between the CTT and residences T, and (iv) northwest of building B, which is a playground for the ÉTS daycare (indicated by a blue pictogram on Figure 4). For the latter zone, the noise level is higher during the "peak hour" measurement because children were playing in the courtyard. It is still interesting to note that this space is reasonably protected from traffic noise (see Figure 4(a)). The first three listed "quiet" zones would be prioritized for taking breaks (e.g., lunch) outdoors on the ÉTS campus. Unfortunately, this is not the case for the park surrounding building C, which is heavily impacted by traffic noise. This area could benefit from acoustic improvements (e.g., green screens) to enhance the acoustic comfort for users.

Lastly, a notable source of noise appears on both maps in Figure 4. It is a ventilation exhaust located south of building E, indicated by a pictogram on Figure 4. This noise can be perceived in the area between buildings E and D, reducing the "quiet" space between these two buildings.

4.2 Indoor noise

This section presents the results of the indoor noise studies for the three types of spaces mentioned in section 3.2.

Core learning spaces

A total of 15 rooms belonging to the category of core learning spaces were measured. Figure 5(a) presents the distribution of noise levels for the 15 rooms and shows that 80 % of the rooms have a noise level below 40 dB(A), and 40 % have a level below or equal to 35 dB(A). Among these 15 rooms, 5 are classrooms (with an average volume of approximately 290 m^3), and 80 % of these classrooms have a noise level below 40 dB(A) (see Figure 5(b)), which is considered suitable for learning according to [20]. Furthermore, the measured reverberation times (TR) at different octave bands (i.e., 500 Hz, 1000 Hz, and 2000 Hz) in these classrooms were all below 0.7 seconds, which again is considered adequate according to [20] (although slightly higher than the recommendations of the ANSI/ASA standard [17]). The only classroom measured that exceeds 40 dB(A) is located under the 6th floor of Building D, where the main mechanical room of the building is situated and is adjacent to a mechanical shaft. Despite this, the measured level is 41 dB(A), which is still very close to the proposed limit value in [20]. Figure 5(c) also shows that Building D has more rooms below the 40 dB(A) threshold compared to Building A, which is older.

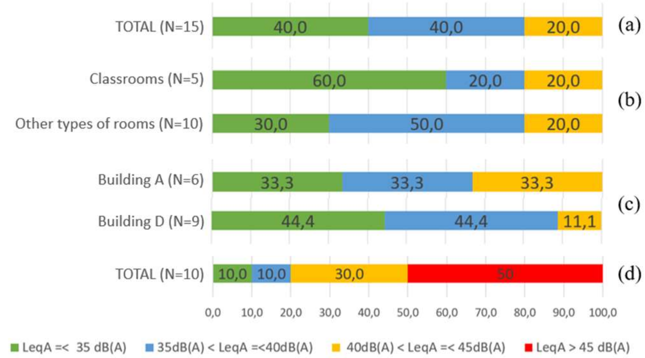


Figure 5: Distribution of noise levels, in dB(A), in core learning spaces: (a) all measured rooms, (b) classrooms vs other types of rooms, (c) rooms in Building A vs rooms in newer Building D. (d) Noise level, in dB(A), in ancillary learning spaces.

Ancillary learning spaces

Ten rooms belonging to the category of ancillary learning spaces were measured during indoor measurement sessions: the cafeteria, the gymnasium, student club rooms (twice), collaborative open spaces (three times), the atrium of Building E where amplified performances occasionally take place (twice), and a large capacity space (enclosed) with speech amplification. Figure 5(d) shows the distribution of sound pressure levels for these ancillary learning spaces, and Figure 6 presents acoustic images taken in some of these spaces. Half of the measured spaces have a noise level below or equal to 45 dB(A). The spaces that exceed this value are: (i) the cafeteria in Building A with 50 dB(A), presumably due to the numerous cooling equipment present (see Figure 6(a)), (ii) the gymnasium with 57 dB(A), due to ventilation (see Figure 6(b)), (iii) a collaborative space in Building D with 56 dB(A), which is located near a cafeteria, and (iv) the atrium of Building E with 48 dB(A), due to the escalator.

Electrical and mechanical rooms

Four electrical rooms, three mechanical rooms, and one server room were measured during the laboratory sessions of the project. The noise level in these rooms (see Table 2) is naturally higher than that in the learning spaces but is not very high according to the Quebec regulations on noise in the workplace [24], where the daily noise exposure limit ($L_{ex,8h}$) is set at 85 dB(A).

Table 2: Noise level in dB(A) in electrical, mechanical, and server rooms.

Type of room	Number of rooms	L_{Aeq} (dB(A))
Electrical room	4	58 < L_{Aeq} < 73
Mechanical room	3	67 < L_{Aeq} < 69
Server room	1	57

However, the noises generated in these rooms have a characteristic spectral signature with energy concentrated at certain frequencies, as shown in Figure 7 (see the blue curves in Figures 7(a), 7(b), and 7(c)). These noises are perceived in neighboring rooms (see the red and yellow curves in Figures 7(a), 7(b), and 7(c)), and although they have low amplitude, they can be bothersome to people working in these areas. These noises are primarily “electric hum” generated by transformer cores, characterized by significant acoustic energy at twice the power frequency (2×60 = 120 Hz) and its harmonics. One transformer has been identified as one of the main sources of noise in an electrical room, as shown in Figure 6(c) captured by the acoustic camera. The noise spectra in Figure 7 also exhibit a significant component at 60 Hz (and its harmonics, including 180 Hz), which can be perceived in neighboring rooms.

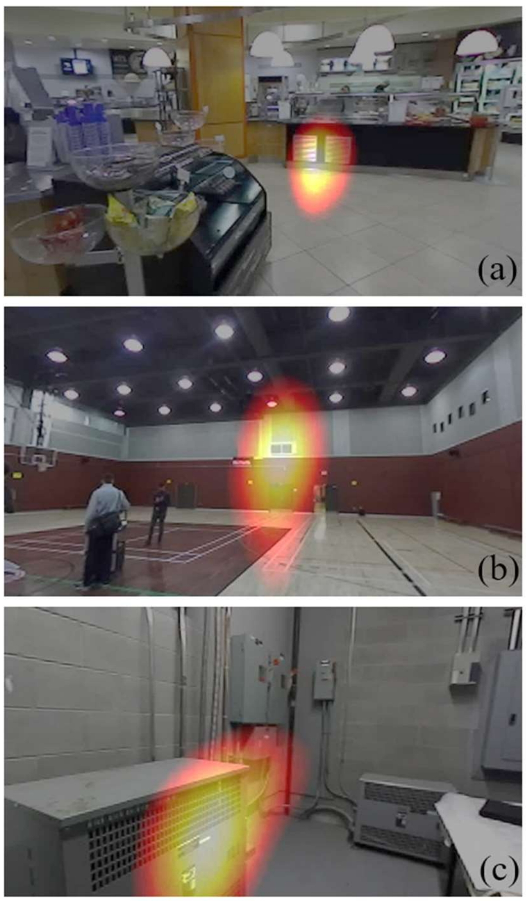


Figure 6: Acoustic images of three rooms: (a) cafeteria of Building A, (b) gymnasium of Building B, and (c) electrical room. The center of the colored spot indicates the position of the dominant acoustic source in the room.

One solution to reduce the noise from these equipments is the use of acoustic enclosures [25]. An enclosure isolates the noisy equipment from the external acoustic environment and should have internally lined absorbent walls to also reduce the acoustic energy within the internal cavity formed by the enclosure. The following section presents an acoustic metamaterial intended to be used as a constituent material for the transformer enclosure. A metamaterial-based solution is preferred because conventional acoustic materials are inefficient in absorbing energy at such low frequencies, here $f <$ 400 Hz.

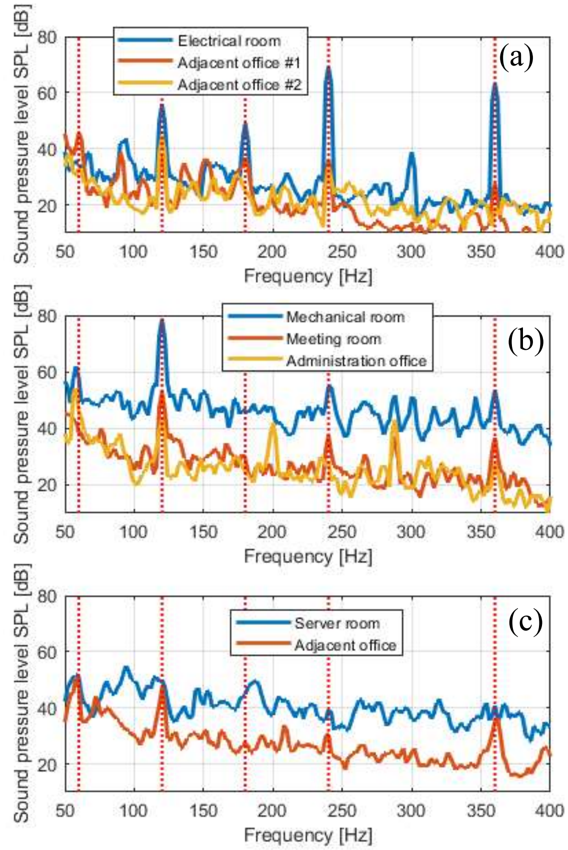


Figure 7: Spectrum of sound pressure levels in dB in (a) an electrical room and two adjacent offices, (b) a mechanical room and two adjacent spaces (a meeting room and an administration office), (c) a server room and an adjacent office. Vertical dashed red lines are placed at frequencies of 60 Hz, 120 Hz, 180 Hz, 240 Hz, and 360 Hz.

4.3 Acoustic metamaterial for noise reduction of electrical transformers

The previous section allowed to identify several frequencies for which the energy of the noise sources in the electrical and mechanical rooms is significant and requires treatment. In particular, the targeted frequencies in this project are: 60 Hz, 120 Hz, 180 Hz, and 360 Hz. The QR has been designed to absorb acoustic energy at a frequency of 120 Hz. For this purpose, its length is set to $L_{\mathrm{QR}} = 702{,}3$ mm (see Eqs. (A3) and (A4) in the appendix). Since the resonator is effective at its fundamental frequency and odd multiples of it, it allows for energy absorption at the frequency of 360 Hz (3×120 Hz). The diameter that provides the best absorption at these two frequencies was determined through a trial-and-error process to be $r_{QR} = 15$ mm.

The absorption of acoustic energy at the other frequencies of 60 Hz and 180 Hz is achieved by the HR. The dimensions of the HR were determined through an optimization process using a genetic algorithm, where the cost function is defined as follows:

$$\varepsilon = \left|f_{pic,1} - 60\right| - \left|f_{pic,i+1} - 180\right| , \qquad (3)$$

where $f_{\mathrm{peak},i}$ is the frequency of the i-th absorption peak. A constraint on the absorption coefficient values of the peaks was set to 0.8. The values of the geometric characteristics of the multi-resonator in Figure 3(b) are indicated in Table 3 below. The only imposed value was that of the wall thickness: $e_{\text{wall}} = 5$ mm.

Table 3: Parameters of the geometry (in mm) of the Helmholtz multi-resonator.

$r_{\text{neck},1}$	$r_{\text{neck},2}$	$h_{\text{neck},1}$	$h_{\text{neck},2}$	$r_{\text{cav},1}$	$r_{\text{cav},2}$	$h_{\text{cav},1}$	$h_{\text{cav},2}$
6	7	7	45	83	72	40	40

A numerical model of the optimized unit cell geometry of the metamaterial is shown in Figure 8. The QR is spirally wrapped to minimize the overall thickness of the material [26]. The arrangement of the two resonators (i.e., QR wrapped around HR) was designed to be 3D printed. This design would help reduce material waste and machining complexity.

Figure 9 represents the absorption coefficient as a function of frequency for the proposed unit cell of the metamaterial. The four absorption peaks occur at the target frequencies, and their amplitudes are above 0.95.

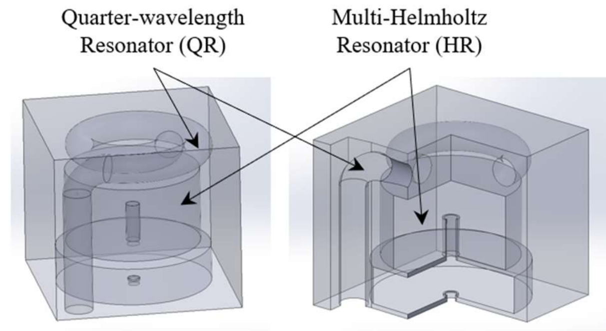


Figure 8: Unit cell of the metamaterial: (a) isometric view, (b) cross-sectional view.

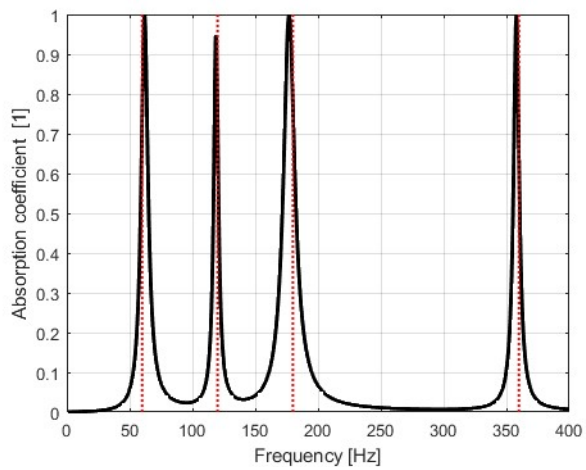


Figure 9: Absorption coefficient of the excited metamaterial under normal incident plane wave. Vertical dashed red lines are placed at frequencies 60, 120, 180, and 360 Hz.

Conclusion

This paper had the main objective of quantifying and assessing the outdoor and indoor noise on the campus of ÉTS in Montreal. All the work was carried out by students enrolled in the "Industrial Acoustics" course (MEC636) at ÉTS as part of their semester project. Their goal was twofold: (i) to apply the theoretical and experimental knowledge they acquired during the course, and (ii) to make their findings useful to the ÉTS community and the residents of the neighborhood.

The first stage of the project involved creating noise maps of the campus and identifying the quietest and noisiest areas. Unsurprisingly, this downtown campus experiences significant noise levels, exceeding the WHO's recommendations (above 55 dB(A)) across most of its surface. Noise levels are higher during the day (before 4:00 PM) due to ongoing construction work in the neighborhood. However, some quieter areas were identified, particularly in the outdoor pedestrian courtyards between buildings and university residences.

The second stage of the project focused on assessing the acoustic quality of various indoor spaces in different campus buildings, such as classrooms, offices, and the library. Generally, the measured classrooms were deemed suitable for learning, with background noise levels below 40 dB(A) and reverberation times below 0.7 s. However, 50 % of the ancillary learning spaces (e.g., gymnasium, cafeteria, atrium, collaborative spaces) exceeded 45 dB(A) and would benefit from acoustic improvements to enhance the acoustic comfort for the student community.

The third and final stage of the project concentrated on the noise from the electrical and mechanical rooms in the campus buildings. These rooms generate noise with characteristic tonal signatures, which can be perceived in adjacent rooms and disturb the personnel working there. One solution is to enclose the noise sources in these rooms. The students in the course used an acoustic camera to locate the main noise sources and designed an acoustic metamaterial specifically for encasing them. Simulation of the proposed metamaterial's acoustic behavior demonstrated its ability to absorb acoustic energy at four frequencies identified as problematic in the measured noise spectra of these rooms and neighboring offices.

This semester project work for the MEC636 course naturally has several limitations that will provide opportunities for future perspectives with other groups of students in the same course. Regarding outdoor measurements, they should be repeated for both studied periods and at various times of the year (only measurements in winter have been conducted so far) to obtain more representative average noise levels. The "NoiseCapture" application [13] is well-suited for this purpose as it is based on a collaborative approach to data production. Furthermore, this tool allows for integrating noise measurement with mobile phones within the course, emphasizing the different mechanisms required to obtain quality measurements with this type of device (e.g., calibration).

The number of measurements is also a limitation in the indoor noise measurement campaign presented in this paper. Longer duration measurements and measurements for more rooms should be conducted in the future. Other indicators could also be calculated and used to analyze the acoustic quality of indoor environments [20, 27]. Subjective measurements through questionnaires could complement the objective measurements taken with microphones, building on advances in research on soundscape characterization and analysis [28-30]. These measurements would help characterize how the campus environments (both outdoor and indoor) are perceived by people and provide better guidance for finding solutions to offer more comfortable acoustic environments. Regarding the metamaterial, it would be necessary to manufacture a cell to experimentally validate the concept and then design an enclosure using multiple cells' tiling to verify its effectiveness *in situ*.

Acknowledgements

The authors would like to express their gratitude to the project coordinators of Noise-Planet for their guidance and for organizing the two "NoiseCapture parties" that facilitated the creation of noise maps for both measurement periods. Furthermore, we would like to thank the administration of ÉTS for their support of this project, as well as the various staff members who opened the doors to their workspaces. We would like to extend a special thanks to Magdalena Stanescu, Building Mechanical and Electrical Engineer at ÉTS, for her time and assistance during the project's laboratory sessions, and to Nathalie Desgagné, Deputy Director of the Office of Prevention and Security, for the fruitful discussions.

Appendix: Modeling of the metamaterial resonators using the transfer matrix method.

Wave propagation through air layers, considering visco-thermal dissipation and radiation at the openings.

The propagation of acoustic waves through an air layer of thickness L is considered using the following transfer matrix:

$$T_{air} = \begin{bmatrix} \cos(\tilde{k}_0 L) & j\tilde{Z}_0 \sin(\tilde{k}_0 L) \\ \frac{j\sin(\tilde{k}_0 L)}{\tilde{Z}_0} & \cos(\tilde{k}_0 L) \end{bmatrix}, \tag{A1}$$

with $\tilde{Z}_0$ the specific impedance of air (Pa.s.m^{-1}), $\tilde{k}_0$ the wavenumber in air. $\tilde{Z}_0$ et $\tilde{k}_0$ are complex and frequency-dependent to account for visco-thermal dissipations in the necks and cavities. For this purpose, the Qunli model [31] is used by calculating the airflow resistivity of the different sections of the duct (e.g., neck, cavity) based on their radius r according to [12]:

$$\sigma_0 = \frac{8\eta}{r^2}, \tag{A2}$$

with η, the dynamic viscosity of air (Pa.s).

The radiation at the openings (i.e., HR necks and QR inlet) was taken into account by applying a correction to the geometric length of the air layer that radiates into a larger air space. The length correction is given by:

$$L' = L + n \cdot 0{,}82 \cdot r\,, \tag{A3}$$

with n=1 if the layer has only one of its two ends radiating (i.e., QR), or n =2 if both ends radiate (i.e., HR necks).

Quarter-wavelength resonator (QR)

The quarter-wavelength resonator was modeled using an air layer, $T_{\mathrm{QR}} = T_{\mathrm{air}}$, whose corrected length is determined by the target frequency f:

$$L'_{\mathrm{QR}} = \frac{c_0}{4f}, \tag{A4}$$

with $f = 120$ Hz and c_0 is the speed of sound in air (m.s^{-1}).

From the transfer matrix T_{QR}, the input impedance of the QR is determined by:

$$Z_{\mathrm{QR}} = \frac{T_{\mathrm{QR},11}}{T_{\mathrm{QR},21}}. \tag{A5}$$

Multi- Helmholtz resonator (HR)

Changes in section (CS) in the HR are taken into account using the following matrix:

$$T_{\mathrm{CS}} = \begin{bmatrix} 1 & 0 \\ 0 & \frac{S_{\mathrm{s}}}{S_{\mathrm{e}}} \end{bmatrix}, \tag{A6}$$

with S_{e}, the surface area of the section upstream of the section change, and S_{s} the surface area of the section downstream of the section change.

Each neck of the HR extends into the cavity. Each cavity has been divided into two parts, delimited by the dashed gray line in Figure 3(b). The parts located before the neck can be considered as quarter-wavelength resonators of length $\left(h_{\mathrm{neck},i} - e_{\mathrm{wall}}\right)$ (with i=1,2 for cavities 1 and 2). These resonators, located in parallel with respect to the propagation direction within the thickness of the HR, have been modeled using the following matrix:

$$T_{\mathrm{res},i} = \begin{bmatrix} 1 & 0 \\ \frac{1}{Z_{\mathrm{res},i}} & 1 \end{bmatrix}, \tag{A7}$$

with $Z_{\mathrm{res},i}$ the acoustic input impedance of each quarter-wave resonator in cavities $i = 1{,}2$, and which is given by the following expression:

$$Z_{\mathrm{res},i} = \frac{Z_0}{j\tan\left(k_0\left(h_{\mathrm{col},i} - e_{\mathrm{paroi}}\right)\right)}. \tag{A8}$$

The total transfer matrix of the HR is given by:

$$T_{\mathrm{multi}} = T_{\mathrm{air},1} \cdot T_{\mathrm{CS},1} \cdot T_{\mathrm{res},1} \cdot T_{\mathrm{CS},2} \cdot T_{\mathrm{air},2} \cdot T_{\mathrm{CS},3} \cdot T_{\mathrm{air},3} \cdot T_{\mathrm{CS},4} \cdot T_{\mathrm{res},2} \cdot T_{\mathrm{CS},5} \cdot T_{\mathrm{air},4}\,. \tag{A9}$$

The calculation of the transfer matrix T_{multi} of the HR allows to find its input impedance from its components T_{11} and T_{21}, according to:

$$Z_{\mathrm{HR}} = \frac{T_{11-\mathrm{multi}}}{T_{21-\mathrm{multi}}}. \tag{A10}$$

Références

[1] World Health Organization - WHO. Guidelines for Community Noise. Geneva, 2019.

[2] R. Martin, P. Deshaies, M. Poulin. Avis sur une politique québécoise de lutte au bruit environnemental : pour des environnements sonores sains. Québec : INSPQ, 2016.

[3] T. B. de Souza, K. C. Alberto, S. A. Barbosa. Evaluation of noise pollution related to human perception in a university campus in Brazil. Applied Acoustics157,107023, 2020. https://doi.org/10.1016/j.apacoust.2019.107023.

[4] P. H. T. Zannin, M. S. Engel, P. E. K. Fiedler, F. Bunn. Characterization of environmental noise based on noise measurements, noise mapping and interviews: A case study at a university campus in Brazil. Cities 31,317–327, 2013. https://doi.org/10.1016/j.cities.2012.09.008.

[5] D. Çolakkadıoğlu, M. Yücel, B. Kahveci, Ö . Aydınol. Determination of noise pollution on university campuses: a case study at Çukurova University campus in Turkey. Environ Monit Assess 190,203, 2018. https://doi.org/10.1007/s10661-018-6568-8.

[6] M. S. Ragettli, S. Goudreau, C. Plante, M. Fournier, M. Hatzopoulou, S. Perron, A. Smargiassi. Statistical modeling of the spatial variability of environmental noise levels in Montreal, Canada, using noise measurements and land use characteristics. Journal of Exposure Science and Environmental Epidemiology 26, 597–605, 2016.

[7] Vivre en ville. Gestion intégrée du bruit environnemental : Trousse d'outils pour un climat sonore agréable., Collection Passer à l'Action, 26 p. vivreenville.org, 2020.

[8] Comité Campus ÉTS. Rapport final : Consultations pour le développement urbanistique du Campus ÉTS. https://www.etsmtl.ca/ETS/A-propos/developpement-campus-ets, 2018.

[9] R. Martin. Vision et orientations gouvernementales en matière de lutte contre le bruit environnemental au Québec. Québec, Canada: Ministère de la Santé et des Services sociaux, 2019.

[10] F. Laville, O. Doutres, J. Voix. MEC636 : Acoustique industrielle », Notes de cours, École de technologie supérieure, 2023.

[11] F. A. Laville. Une expérience d'enseignement de l'acoustique industrielle intégrant pédagogie de la coopération, projet de session et laboratoire informatique [A teaching experience in the industrial acoustics integrating pedagogy of cooperation, computer laboratory and session project]. Canadian Acoustics.;40(2),11–14, 2012.

[12] J.-F. Allard, N. Atalla. Chapter 11: Modelling multilayered systems with porous materials using the transfer matrix method. Propagation of sound in porous media: modelling sound absorbing materials (2nd ed., pp. 243-281). Chichester: John Wiley & Sons, 2009.

[13] J. Picaut , N. Fortin, E. Bocher, G. Petit, P. Aumond, G. Guillaume. An open-science crowdsourcing approach for producing community noise maps using smartphones. Building and Environment. 148,20–33, 2019. https://doi.org/10.1016/j.buildenv.2018.10.049.

[14] G.Guillaume, P. Aumond , E. Bocher, Can A., Écotière D., Fortin N., et al. NoiseCapture smartphone application as pedagogical support for education and public awareness. The Journal of the Acoustical Society of America. 151(5), 3255–3265, 2022. https://doi.org/10.1121/10.0010531

[15] Mecanum. (n.d.). LF-ANT | Caméra acoustique 3D. https://mecanum.com/fr/measuring-instruments/acoustic-camera/ (accessed May 1, 2023).

[16] J. Frechette-Viens, N. Quaegebeur, N. Bastien. A low-latency acoustic camera for transient noise source localization. Berlin Beamforming Conference.Berlin, 2020.

[17] American National Standard. Acoustical Performance Criteria, Design Requirements, and Guidelines for Schools, Part 1: Permanent Schools. ANSI/ASA S12.60-2010/Part 1 (R2015). Melville, NY : Acoustical Society of America, 2015.

[18] ASHRAE handbook, chap. 48 « Noise and vibration control», 2011.

[19] J. Romeu, M. Genescà, T. Pàmies, S. Jiménez. Street categorization for the estimation of day levels using short-term measurements, Applied Acoustics, 72(8), 569-577, 2011. https://doi.org/10.1016/j.apacoust.2010.09.012.

[20] G. Minelli, G. E. Puglisi, A. Astolfi. Acoustical parameters for learning in classroom: A review. Building and Environment.208,108582, 2022. https://doi.org/10.1016/j.buildenv.2021.108582.

[21] K. Verdière, R. Panneton, S. Elkoun, T. Dupont, P. Leclaire. Comparison between parallel transfer matrix method and admittance sum method. The Journal of the Acoustical Society of America136(2) : EL90–EL95, 2014. https://doi.org/10.1121/1.4885481.

[22] N. Sharafkhani. An ultra-thin multi-layered metamaterial for power transformer noise absorption. Building Acoustics.29,53-62, 2022. https://doi.org/10.1177/1351010X211041704.

[23] A. Gilchrist, E. N. Allouche, D. Cowan. Prediction and mitigation of construction noise in an urban environment. Canadian Journal of Civil Engineering. 30,659–672, 2003. https://doi.org/.1139/l03-019.

[24] Gouvernement du Québec. Loi sur la santé et la sécurité du travail. Décret 781-2021. Gazette officielle du Québec, 153(24), 2722-2733, 2021.

[25] D. K. Meinke, E. H. Berger, D. P. Driscoll, R. L. Neitzel, K. Bright. Chapter 10 : Noise Control Engineering. In The noise manual. (6th edition). Falls Church, VA: American Industrial Hygiene Association, 2022.

[26] G. Catapane, G. Petrone, O. Robin, K. Verdière. Coiled quarter wavelength resonators for low-frequency sound absorption under plane wave and diffuse acoustic field excitations. Applied Acoustics, 2023. 209,109402. https://doi.org/10.1016/j.apacoust.2023.109402.

[27] Astolfi A., Minelli G., Puglisi G. E. A basic protocol for the acoustic characterization of small and medium-sized classrooms. The Journal of the Acoustical Society of America. 152(3),1646–1659, 2022.

[28] International Organization for Standardization. Acoustics — Soundscape — Part 2: Data collection and reporting requirements. ISO/TS 12913-2:2018, 2013. https://www.iso.org/standard/75267.html

[29] F. Aletta, T. Oberman, Ö .Axelsson, H. Xie, Z. Yuan, S. K. Lau, et al. Soundscape assessment: towards a validated translation of perceptual attributes in different languages. In: INTER-NOISE and NOISE-CON Congress and Conference Proceedings. Seoul; p. 3137–3146, 2020.

[30] S. Mancini, A. Mascolo, G. Graziuso, C. Guarnaccia. Soundwalk, Questionnaires and Noise Measurements in a University Campus: A Soundscape Study. Sustainability. 13(2),841, 2001.

[31] W. Qunli. Empirical relations between acoustical properties and flow resistivity of porous plastic open-cell foam. Applied Acoustics.25,141–148, 1998. https://doi.org/10.1016/0003-682X(88)900 90-4.

Comparison of Various Algorithms: Research on Piano Audio Signal Feature Identification

Shuang Hao *[1]

[1]College of Film and Television, Hebei University of Science and Technology, Shijiazhuang, Hebei, China

Résumé

Cet article présente brièvement les méthodes d'extraction des caractéristiques des signaux audio de piano et notamment les algorithmes basés sur la déformation temporelle dynamique (DTW), le réseau neuronal à rétropropagation (BPNN) et le réseau neuronal à convolution (CNN), qui peuvent reconnaître les caractéristiques audio de piano. Les trois algorithmes de reconnaissance ont été comparés dans les expériences de simulation suivantes. Il a été constaté que pour certains extraits audio de piano à une ou plusieurs notes, les résultats de reconnaissance de l'algorithme CNN étaient cohérents avec les résultats standard, l'algorithme BPNN présentait quelques différences et l'algorithme de reconnaissance basé sur DTW présentait les différences les plus importantes. Avec l'augmentation du nombre de notes dans l'extrait audio de piano, la précision de reconnaissance de tous les algorithmes a diminué, mais c'est l'algorithme CNN qui a le moins diminué, et sa performance de reconnaissance était la plus importante.

Mots-clés : piano, caractéristiques audio, réseau neuronal convolutionnel, déformation temporelle dynamique

Abstract

This article briefly introduced feature extraction methods for piano audio signals and algorithms based on dynamic time warping (DTW), back-propagation neural network (BPNN), and convolutional neural network (CNN), which can recognize piano audio features. The three recognition algorithms were compared in the subsequent simulation experiments. It was found that for some single-note and multi-note piano audios, the recognition results of the CNN algorithm were consistent with the standard results, the BPNN algorithm had some differences, and the DTW-based recognition algorithm had the most differences. As the number of notes in the piano audio increased, the recognition accuracy of all the algorithms decreased, but the CNN algorithm decreased the least, and its recognition performance was highest under the same number of notes, followed by the BPNN algorithm, and the DTW-based recognition algorithm was the lowest.

Keywords: piano, audio feature, convolutional neural network, dynamic time warping

1 Introduction

The emergence time of music is difficult to examine, but throughout the development of human society, music has gradually entered people's lives and become a form of art [1]. Moreover, with the development of computer and internet technology, both music recognition and retrieval [2] require accurate identification of audio signal characteristics [3]. Digital music technology can accurately and quickly identify piano audio signals. Not only can it assist students in correcting piano intonation errors, but it can also quickly convert audio signals into piano music symbols, further assisting in the composition of piano music. Li [4] utilized model recognition technology to design a multi-note model based on HMM and confirmed its practicality. Wu [5] used a convolutional neural network (CNN) to identify piano sheet music. The experimental results verified the accuracy of the algorithm. Wang et al. [6] developed an audio identification method based on a combination of CNN and a generative adversarial network. The experiments on the AViD corpus and DAIC-WOZ dataset demonstrated that compared to other existing methods, this method effectively reduced recognition errors for depression. This article briefly introduced the feature extraction methods for piano audio signals as well as three algorithms - dynamic time warping (DTW), back-propagation neural network (BPNN), and CNN - that can recognize piano audio features. Then, these three recognition algorithms were compared through simulation experiments.

2 Extracting signal features from piano audio

When recognizing the audio signal of a piano, i.e., converting piano audio signals to sheet music with note sequences, it is first necessary to extract the features of the signal (features are indicators that reflect the characteristics of an audio, and using the indicators can distinguish different audio signals). The methods for extracting audio features include the linear prediction cepstral coefficient (LPCC) method and the Mel-frequency cepstral coefficient (MFCC) method. The LPCC method, which is based on linear predictive analysis and assumes that the audio signal is a linear autoregressive signal [7], is an audio feature extraction method that can effectively extract excitation information from the audio signal. However, in reality, audio signals are often not linear regression relationships, so the anti-noise ability of this feature is poor.

* esh968864@yeah.net

Compared with the LPCC method, because the Mel frequency scale is closer to the human ear's perception of audio signals, the audio features obtained by the MFCC method are more compatible with the auditory effect of the human ear [8].

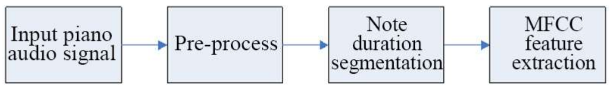


Figure 1: Piano audio feature extraction process based on MFCC

Figure 1 shows the relevant process.

① The piano audio signals to be identified are pre-processed by windowed framing [9]. The formulas for windowing and framing are:

$$\begin{cases} S_w(n) = s(n) \times w(n) \\ w(n) = \begin{cases} 0.54 - 0.46\cos\left(\dfrac{2\pi n}{N-1}\right) & \text{if } 0 \leq n \leq N-1 \\ 0 & \text{else} \end{cases} \end{cases} \tag{1}$$

where $S_w(n)$ denotes the windowed speech digital signal, $s(n)$ denotes the original speech digital signal, $w(n)$ denotes the Hamming window function, and N denotes the length of the digital signal.

② When extracting features from a piano audio signal consisting of single notes, MFCC can be used following the previous step. However, in actual piano audio signals, there are often continuous multiple notes, so the signal needs to be segmented into note values. The energy entropy ratio of each frame of audio signal is used to form a common energy entropy ratio (ratio of energy to spectral entropy in music signals) graph that changes with frame number [10]. Then, each valley point in the energy entropy ratio graph is taken as the starting point of a single note, and the corresponding point whose energy entropy ratio difference from the starting point is less than a preset threshold is taken as the end point of that single note, by moving along the frame time axis in order.

③ MFCC feature extraction is applied to each frame of the audio signal after being segmented by note duration [11].

3 Algorithms for recognizing piano audio signal features

3.1 Piano audio feature recognition method based on DTW

The previous text described the method of extracting piano audio signal features. By using the extracted MFCC features, it is possible to recognize the notes of the piano audio signal. Matching the template library of notes with the audio signal to be recognized is a method of note recognition for piano audio signals. The template library of notes stores the MFCC characteristics of different notes. The basic principle is to extract the MFCC features of the audio in the note template library and the audio to be recognized, and then perform pattern matching based on their MFCC features.

Even the same note on a piano may have different durations when played. When using a note template library to match audio to be recognized, it is highly likely that the durations will be inconsistent [12]. If the audio signal is linearly stretched or compressed as a way to match the duration of the template notes, the duration transformation of the individual segments in the signal under different circumstances will be ignored, which will result in information loss. The DTW algorithm can perform non-linear bending of audio, solving the problem of inconsistent lengths between the audio to be recognized and the template audio. Its steps are:

① Pre-emphasis, windowed framing, note duration segmentation, and MFCC feature extraction are carried out on the audio signal as described in the previous chapter.

② The DTW algorithm is used to calculate the distance between the audio to be recognized and different template audios.

③ The template audio with the smallest distance from the audio to be recognized is used as the recognition result.

3.2 Piano audio feature recognition method based on deep learning

Deep learning algorithms have gradually been applied to audio recognition. The BPNN is a classic deep learning algorithm. When used for piano audio feature recognition, the audio is first pre-processed and MFCC feature parameters are extracted. Then, the MFCC feature parameters are input into the BPNN for forward calculation in the hidden layer, and ultimately the note recognition results are output in the output layer. Compared with the DTW algorithm, the BPNN for piano audio recognition requires a large amount of data for training, but it does not need any template audio to recognize piano audio signals after training, and there is no need to adjust the note duration [13].

CNN can also be used for recognizing audio signals. The steps are as follows:

① The audio signal is processed as described previously, including pre-emphasis, windowed framing, segmentation of note duration, and extraction of MFCC features.

② The extracted MFCC features are into the CNN. The formula of convolution operation is:

$$O_i = f(O_{i-1} \otimes W_i + B_i) \tag{2}$$

where O_i and O_{i-1} are the feature maps outputted from the i-th layer and $(i-1)$-th layer, W_i is the weight in the structure of the i-th layer, B_i is the bias in the structure of the i-th layer, and $f(\cdot)$ is the activation function.

③ After obtaining the convolutional feature map, in order to reduce the subsequent computational complexity, it is compressed in the pooling layer. The specific operation is to use a pooling box to gradually slide on the feature map, and compress the feature data in the pooling box. In simple terms, multiple data in the pooling box are merged into one data, which can be the average of multiple data or the maximum value among the multiple data. The former is mean pooling, and the latter is max pooling [14].

④ The pooled convolution feature maps are then classified using the softmax function in the fully connected layer, and the corresponding music note recognition results are input into the output layer. Finally, the recognition results are output.

When the algorithm is in its training phase, the computed recognition results are compared with the corresponding label results in the training set, and this article uses cross-entropy to compute the error between them. The error is then evaluated to see if it has converged to the target range. If it reaches the preset range, the training is finished; if not, the weighted parameters in the algorithm are adjusted according to the error in the opposite direction.

4 Simulation experiments

4.1 Experimental data

The dataset used for the simulation experiments was the MAESTRO dataset. This dataset was obtained in cooperation with the organizers of international piano competitions. During the process of collecting the original piano data, a high-precision MIDI capture and playback system was used to ensure the accuracy of the data as much as possible. This dataset contained approximately 200 hours of piano audio (16-bit PCM stereo at a sampling rate of 48 kHz) and corresponding MIDI data. The MIDI data included key velocity and pedal position, which can be considered that the data set carried the note labels corresponding to the audio data.

4.2 Experimental setup

Simulation experiments were conducted on three piano audio recognition algorithms based on DTW, BPNN, and CNN, respectively. Firstly, the feature dimension of MFCC was set to 24 after orthogonal experiment. The Hamming window was used, the frame length was set to 20 ms, and the frame shift size was 10 ms.

In the DTW-based piano audio recognition algorithm, the most important part is the note template library used for matching and recognition. For the audio of notes in the template library, the dimension of extracted MFCC features was also set to 24. The Hamming window was used in windowed framing, with a frame length of 20 ms and a frame shift of 10 ms.

In the BPNN-based piano audio recognition algorithm, the relevant parameters are as follows. The number of nodes in the input, hidden, and output layers were 24, 128, and 88, respectively; the activation function was the Relu function. The epoch was set as 200.

In the CNN-based piano audio recognition algorithm, the relevant parameters are shown below. There were three convolutional layers and three mean pooling layers. During the training process, a regularization with a random dropout of 0.4 was used, and the number of epochs was set to 200.

During the testing of three piano audio recognition algorithms, the first step was to test the recognition of single-note piano audio signals by the three algorithms. Afterwards, the recognition of multi-note piano audio signals with different numbers of notes was tested, with the number of notes set at 5, 10, 15, 20, and 25, respectively. The recognition performance of the three algorithms on the piano audio signals in the aforementioned two experimental projects was tested. Precision, recall rate, and F-score were chosen as the performance indicators.

4.3 Experimental results

Due to space limitations, only waveforms of partial single- and multiple-note piano audios are shown in Figure 2. The single-note piano audio waveform in Figure 2 corresponds to the note named "c", pronounced as "do"; the multiple-note piano audio waveform corresponds to the note named "c^1def^1cd". The recognition results for the piano audio in Figure 2 using three different audio recognition algorithms are shown in Table 1. According to Table 1, the recognition results obtained by the CNN algorithm were consistent with the standard results, the BPNN algorithm differed slightly, and the DTW recognition algorithm differed the most from the standard results.

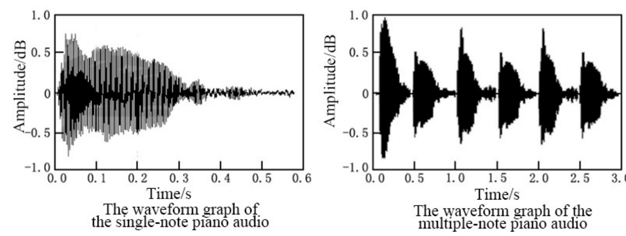


Figure 2: The waveform graphs of some single-note and multiple-note piano audios

Table 1: The recognition results of some single and multi-note piano audios

	Single-note piano audio in Figure 2	Multiple-note piano audio in Figure 2
Standard note result	c	c^1def^1cd
The recognition result of the DTW algorithm	e	$e^1ce^1f^1ce$
The recognition result of the BPNN algorithm	d	c^1ddf^1cd
The recognition result of the CNN algorithm	c	c^1def^1cd

Three recognition algorithms' recognition performance on single-note piano audio is shown in Figure 3. For the DTW recognition algorithm, the precision, recall rate, and F-value of recognizing single-note piano audio were 0.69, 0.68, and 0.68, respectively. For the BPNN algorithm, the values were 0.85, 0.84, and 0.84, respectively. For the CNN recognition algorithm, the values were 0.98, 0.97, and 0.97, respectively. It was seen from Figure 3 that the CNN recognition algorithm had the highest accuracy in recognizing single-note piano audio, followed by the BPNN algorithm, and the DTW recognition algorithm had the lowest accuracy.

The performance of three identification algorithms in identifying piano audio with multiple notes is shown in Figure 4. According to Figure 4, as the number of musical notes in the piano audio increased, the precision, recall rate, and F-values of the three identification algorithms all decreased. However, in the process of increasing the number of notes, the DTW recognition algorithm showed the greatest decrease in recognition performance, while the BPNN algorithm

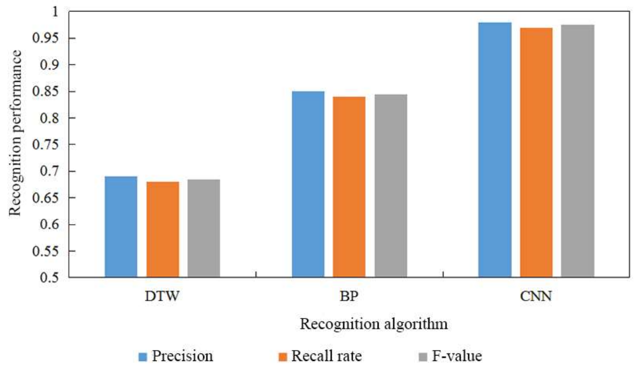


Figure 3: Performance of three recognition algorithms in identifying single-note piano audio

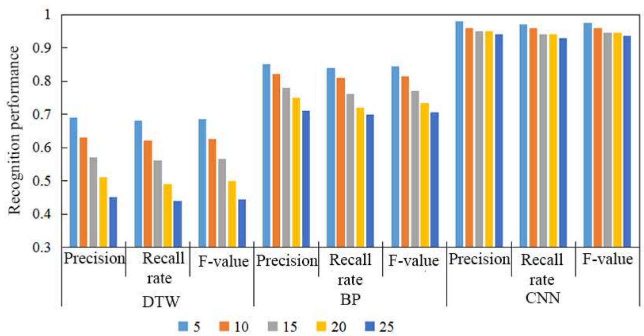


Figure 4: Recognition performance of three recognition algorithms on multiple-note piano audio

showed a relatively small decrease, and the CNN recognition algorithm showed the smallest decrease. When the number of multiple notes was the same, the recognition performance of the CNN algorithm was still the best, followed by the BPNN algorithm, and the DTW algorithm was the worst.

5 Conclusion

This paper briefly introduced the feature extraction methods of piano audio signals and three algorithms, namely DTW, BPNN, and CNN, that can recognize piano audio features. In the subsequent simulation experiments, the three recognition algorithms were compared, and the results are as follows. (1) Among the recognition results of the waveform graphs of some single-note and multi-note piano audios, the recognition results obtained by the CNN recognition algorithm were consistent with the standard results, while the BPNN algorithm differed slightly and the DTW recognition algorithm differed the most. (2) Faced with single-note piano audio signals, the CNN recognition algorithm presented the highest recognition accuracy for the single-note piano audio, followed by the BPNN algorithm, and the DTW recognition algorithm had the lowest accuracy. (3) As the number of notes in the piano audio signal increased, the recognition performance of the three recognition algorithms slightly reduced, among which the DTW recognition algorithm had the most significant reduction, followed by the BPNN algorithm, and the CNN recognition algorithm had the least reduction. Moreover, under the same number of notes, the recognition performance of the CNN algorithm was still the best, followed by the BPNN algorithm, and the DTW algorithm was the worst.

References

[1] S. A. Herff, K. N. Olsen, and R. Dean. Resilient memory for melodies: The number of intervening melodies does not influence novel melody recognition. *Q. J. Exp. Psychol*., 71:1150-1171, 2018.

[2] M. Bomgardner. MATERIALS Schlumberger pilots new lithium extraction. *Chem. Eng. News*, 99:10, 2021.

[3] Y. Wang. Research on Handwritten Note Recognition in Digital Music Classroom Based on Deep Learning. *J. Internet Technol*., 22:1443-1455, 2021.

[4] X. Li. Construction and analysis of hidden Markov model for piano notes recognition algorithm. *J. Intell. Fuzzy Syst*., 37:3293-3302, 2019.

[5] R. Wu. Research on automatic recognition algorithm of piano music based on convolution neural network. *J. Phys. Conf. Ser*., 1941:1-7, 2021.

[6] Z. Wang, L. Chen, L. Wang, and G. Diao. Recognition of audio depression based on convolutional neural network and generative antagonism network model. *IEEE Access*, 8:101181-101191, 2020.

[7] M. Brousmiche, J. Rouat, and S. Dupont. Multimodal Attentive Fusion Network for audio-visual event recognition. *Inform. Fusion*, 85:52-59, 2022.

[8] P. Hoffmann, and B. Kostek. Bass Enhancement Settings in Portable Devices Based on Music Genre Recognition. *J. Audio Eng. Soc*., 63: 980-989, 2015.

[9] X. Wang. Research on the improved method of fundamental frequency extraction for music automatic recognition of piano music. *J. Intell. Fuzzy Syst*., 35:1-7, 2018.

[10] Z. Xiao, X. Chen, and L. Zhou. Real-Time Optical Music Recognition System for Dulcimer Musical Robot. *J. Adv. Comput. Intell. Intell. Inform*., 23 :782-790, 2019.

[11] C. Shuo. The construction of internet plus piano intelligent network teaching system model. *J. Intell. Fuzzy Syst*., 37: 5819-5827, 2019.

[12] R. Wu. Research on automatic recognition algorithm of piano music based on convolution neural network. *J. Phys. Conf. Ser.,* 1941:1-7, 2021.

[12] O. Francesconi, A. Ienco, C. Nativi, and S. Roelens. Effective Recognition of Caffeine by Diaminocarbazolic Receptors. *ChemPlusChem*, 85:1369-1373, 2020.

[13] Y. Luo, and H. Yang. Teaching Applied Piano Singing While Playing Based on Xindi Applied Piano Pedagogy:Taking Fujian Vocational College of Art as an Example. *J. Contemp. Educ. Res*., 6:123-135, 2022.

[14] W. H. Lai, and C. Y. Lee. Query By Singing/Humming System Using Segment-based Melody Matching for Music Retrieval. *WSEAS Trans. Syst*., 15:157-167, 2016.

Announcing the Publication of the Sixth Edition of RR-331: The Guide and a History of the Transition to the ASTC Rating

Jeffrey Mahn *[1]
[1]National Research Council Canada

Résumé

La sixième édition du rapport de recherche RR-331 : Le Guide est annoncé et un historique des recherches menées au Conseil national de recherches du Canada pour soutenir l'introduction de l'évaluation ASTC dans le Code national du bâtiment : Canada est présenté.

Mots clefs : ASTC, Code national du bâtiment : Canada, Conseil national de recherches Canada, Le Guide

Abstract

The sixth edition of the research report, RR-331: The Guide is announced and a history of the research conducted at the National Research Council Canada to support the introduction of the ASTC rating in the National Building Code of Canada is presented.

Keywords: ASTC, National Building Code Canada, flanking, National Research Council Canada, The Guide

The Acoustics Group at the Construction Research Center is pleased to announce the publication of the sixth edition of the research report, RR-331 The Guide. The latest editions of RR-331 and the companion RR-33x reports can be downloaded from the NRC Publications Archive: https://publications-cnrc.canada.ca/eng/home/. The new edition of RR-331 includes updates to both the guidance for calculating the ASTC rating for different building constructions and to the accompanying worked examples. References to the withdrawn standard, ISO 15712 have been replaced with references to the equations found in the newer ISO 12354 standard. The new edition also includes a history of the minimum requirements for airborne noise in the National Building Code of Canada and the decades of research conducted at the NRC to support the transition to the ASTC rating in 2015. The history is being shared with the readers of Canadian Acoustics as part of this announcement of the new edition of RR-331.

Prior to 1941, the lack of consistency or rules for the construction of houses in Canada resulted in the occurrence of significant quality failures amongst the housing stock. To address this problem and the crisis of homelessness during the Great Depression [1], the first National Building Code of Canada (NBC) was published in 1941 under the joint sponsorship of the Department of Finance and the National Research Council of Canada [2]. The new building code included protection from neighbor noise by requiring a "sound transmission loss, or sound reduction, of not less than 45 decibels". ASTM International would not introduce the new sound transmission class rating until 1961 [3] and so the 1941 NBC defined the sound transmission loss as the arithmetic average of the sound transmission loss values measured in at least nine one-half octaves with twice as many measurements made at frequencies below 1024 Hz as above it. Appendix I of the NBC included figures and the accompanying sound transmission loss ratings of typical wall constructions, many of which included wood studs, wood lath and coats of lime plaster without thermal insulation in the wall cavities.

The next edition of the NBC published in 1960 added the wording that "walls and floors…separating dwelling units, sleeping rooms occupied separately or suites in residential buildings shall be designed to prevent the transmission of objectionable noise between occupancies" [4]. The requirements for the sound insulation were changed from a sound transmission loss value to a maximum airborne level of extraneous noise based on the use of a space. For example, rooms for sleeping were allowed to have a maximum airborne level of extraneous noise of 30 dB for single rooms or 40 dB for dormitories. There was also a maximum permissible airborne noise level produced by occupancy of 80 dB for the sleeping rooms.

These requirements remained in place until 1970 when the NBC added the requirement that the "sound transmission class ratings for construction shall be determined in accordance with ASTM E-90-66T" [5]. A minimum sound transmission class of 45 for walls and floors between dwelling units was required. The tables of Part 9 of the NBC included sound transmission class ratings where rating I was for a sound transmission class of 50 or higher, considered to give "good resistance to transmission of airborne sound", rating II was for sound transmission class ratings between 45 and 50 considered to provide "fair resistance to airborne sound" (note that this was the minimum requirement of the NBC) and rating III was for a sound resistance rating less than 45 which was not acceptable.

The 1975, 1977, 1980 and 1985 editions of the NBC would keep the sound insulation requirements unchanged. It wasn't until the 1990 edition that the requirements were changed so that the "sound transmission class ratings shall be

* jeffrey.mahn@nrc-cnrc.gc.ca

determined in accordance with ASTM E413 using results from measurements in accordance with ASTM 90 or ASTM E366." The requirements stated that an STC rating of at least 50 was required for walls and floors between dwelling units and a STC rating of at least 55 was required for walls between dwellings and elevator shafts or refuse chutes [6]. It was around this time that a series of consortiums led by various combinations of Mohamed Sultan, Dave Quirt and Alf Warnock extended and updated the table of STC ratings in Part 9 of the NBC.

The NBC would continue to state the minimum requirements for sound insulation in terms of the STC rating until the 2015 edition [7] when the Apparent Sound Transmission Class (ASTC) rating was introduced. This change in the requirements from the STC rating to the ASTC rating was only achieved after decades of research funded through collaboration between the acoustics group at the National Research Council Canada and industry groups.

Alf Warnock led the initial push to establish flanking studies at the NRC and in 1993, the very first flanking transmission laboratory in the world, a four-room facility was established at the NRC. The four-room flanking transmission laboratory was developed by Robin Halliwell with the occasional support by Dave Quirt. The four-room facility would be used extensively by industry consortia in the 1990s for the evaluation of the effect of fire stops on the acoustical separation between dwellings. See for example, the publications by Trevor Nightingale and Mohamed Sultan [8] and by Trevor Nightingale and Robin Halliwell [9] which describe how a fire stop at the wall / floor junction can degrade the apparent sound insulation of party walls. This finding became the motivation for future research and for changing the acoustical requirements in the NBC from a STC rating to an ASTC rating that included the contribution of the flanking paths.

Subsequent research began to look at the junction details. From around 1995 to 2009, Trevor Nightingale led consortiums of partners brought together to fund the research as acknowledged in the research reports that were produced (see for example [9–12]). Additional partnerships were developed to support the research as different details of the construction were investigated including the junctions, the gypsum board and interlayers.

In 1995, Trevor Nightingale published a paper in Applied Acoustics [13] which demonstrated that the draft EN 12354-1 standard [14] tended to underestimate the transmission loss of a flanking path due to the simplifications made in the model to eliminate the use of the radiation efficiencies which are often not known and are difficult to measure. In a paper coauthored by Eddy Gerretsen and Trevor Nightingale [15] it was noted that the underlying assumptions and simplifications applicable to heavy, monolithic constructions such as concrete may not be directly applicable to lightweight constructions.

The development of the standards ISO 10848 parts 1 and 3 [16, 17] allowed for a path forward around these issues for lightweight elements by creating a framework for measurements of the flanking level differences between flanking surfaces. The direct measurement of the sound pressure level in the source and receiver rooms eliminated the need to calculate or measure the resonant component of the sound reduction index and the radiation efficiencies of the elements included in the flanking path. However, the measurements became more complex than those for heavy, monolithic constructions since the measurement of the flanking paths required the ability to apply shielding to all of the paths in the source and the receiver rooms not included in the flanking path under consideration. The shielding was also required if sound intensity was used to measure the flanking transmission loss [18].

The need arose for a facility that was both more flexible than the four-room facility and that could allow for more complex measurements including the ability to add a structural load to the walls. Fortunately, there was enough interest amongst the consortium of industrial partners to build a new flanking transmission laboratory. The eight-room flanking transmission laboratory was constructed in 2004 and 2005 under a program led by Trevor Nightingale with strong support by Robin Halliwell. Also involved in the work were Timothy Estabrooks, Brian Fitzpatrick, Frances King, Donald MacMillan and Joshua Wu [19]. The facility was commissioned in December 2005 and the first measurements were made in 2006.

One of the uses for the new flanking transmission laboratory was a research program to look at the design details for sound insulation of wooden multi-use buildings for Japan and Korea. The laboratory study demonstrated that it was possible to meet the stringent heavy impact sound insulation requirements of Japan and Korea with wood framed construction. There were other aspects to the work as well including a subjective study of different impact noise sources [20] and an investigation [21, 22] of the applicability of the bang machine described in the standards JIS 1418-2 [23] and KS F 2810-2 [24].

In 2006, as part of the fourth phase of the timber framed research, Dave Quirt and Trevor Nightingale collaborated to develop the Guide for Sound Insulation in Wood Frame Construction [25]. Phase five of the research developed the first version of the online Guide. These were the precursors to the RR-33x series of research reports which would provide guidance for the most common types of residential constructions. The Guide for Sound Insulation in Wood Frame Construction and the online guide were important not only due to the knowledge they contained, but also because they developed the understanding for how to present data which would be used for the RR-33x series. It was around this same time that the creation of an interactive web-based design software was discussed [26].

It was also around this time that Trevor Nightingale submitted a code change request to Codes Canada to replace the acoustic requirements in the National Building Code of Canada which were written in terms of the STC rating with the ASTC rating which included the contribution of structure-borne noise via the flanking paths. In 2007, the acoustic group was asked to draft background discussion papers on the costs and benefits of the proposed updates for both airborne and impact sound for the Standing Committee on Environmental Separation. The task was discussed by various Codes Committees and the Task for Airborne Sound Transmission was eventually approved.

The first meeting of the Task Group on Airborne Sound Transmission took place on the 7th of September, 2010 and production of a detailed technical guide to be referenced in the NBC was proposed in that meeting. The detailed technical guide, now known as Research Report RR-331, Guide to Calculating Sound Transmission in Buildings was developed as part of a special interest group, the SIG-ASTC group which included the partners listed in the acknowledgements at the front of Research Report RR-331. The first SIG-ASTC group (the first of four) was convened by Berndt Zeitler in 2012 with the support of Trevor Nightingale. Dave Quirt conducted much of the analysis work and he developed a solution for dealing with the issues of sound leakage through some concrete block masonry and cross-laminated timber elements. The first edition of RR-331, published in 2013 was authored by Dave Quirt, Berndt Zeitler, Stefan Schoenwald, Ivan Sabourin and Trevor Nightingale.

The 2015 edition of the NBC introduced the ASTC rating. The Code directly references RR-331 in section A-5.8.1.4 where it is described as being a source of "technical concepts, terminology, and calculation procedures relating to the detailed and simplified ASTC calculation methods." This Guide includes numerous worked examples and references to readily-available sources of pertinent data.

With the change in the acoustical requirements for dwellings in the NBC to the ASTC rating, the requirements were more representative of the situation in actual dwellings, but the trade off was that the design of dwellings for protection from neighbor noise became more complicated. No longer could an architect, designer or builder simply choose a wall with a laboratory measured STC rating of 50 from a catalogue to meet the requirements of the NBC. Now, the transmission of structure-borne noise needed to be estimated or found in data from special flanking facilities, depending on the type of structure. Fortunately, The Guide and the RR-33x series of research reports explain how to use the ISO prediction method in the context of the ASTM descriptors used in North America as well as data for the calculations.

Section A-5.8.1.4 of the 2015 NBC also notes that "For many common constructions, the calculations required by Article 5.8.1.4. can be performed using software tools, such as soundPATHS, which is available on NRC's Web site." The web application, soundPATHS [27] was conceived as part of the projects on flanking in timber-framed constructions and the first version was developed by Dave Quirt and the NRC group at the University of Western Ontario in London. The work on soundPATHS was introduced to the world in 2009 by Dave Quirt [28] during the Internoise conference in Ottawa where Trevor Nightingale was co-president and Bradford Gover was technical program co-chair. soundPATHS was completely revised in 2019 to make it more accessible and to include more junction and element data.

The sixth edition of RR-331 represents a significant update to the Guide. References to the withdrawn standard, ISO 15712 have been replaced by references to the latest ISO 12354 standard. In addition, numerous changes have been made based on feedback from both the industry consortium, the SIG ASTC group which funded the work as well as end users who have made comments and recommendations for improvements.

The RR-33x research reports are all living documents with more data added as it becomes available from ongoing research projects. To continue to support the acoustic requirements in the NBC, the NRC has recently completed the construction of a new four-room flanking transmission laboratory to replace the original four-room and eight-room facilities. An additional two-room flanking facility is also under construction as are new facilities to measure the sound transmission loss of wall elements and the sound transmission loss and impact noise of floor elements. These new facilities and the research they will make possible represent the continued commitment by the NRC to support the wellbeing of Canadians and the growth of Canadian industry.

While RR-331 only focuses on the calculation of the ASTC rating for airborne noise, there is research underway to develop similar guidance for impact noise. Protection from impact noise such as people walking or dropping objects on floors is not yet considered in the National Building Code of Canada, but that is expected to change in the future. An industry consortium SIG Impact has been created to develop guidance and a companion document to RR-331. The new guide for impact noise will be published after impact noise requirements have been added to the National Building Code of Canada, most likely in the 2030 edition at the earliest.

Acknowledgments

Thank you to David Quirt and to Trevor Nightingale for sharing their memories of the research projects at the NRC and the development of The Guide.

References

[1] Archer, J.W., A Brief history of the National Buildings Code of Canada, Journal of the Ontario Building Officials Association, (March), 24–25, (2003).

[2] "National Building Code: 1941", National Research Council of Canada, Ottawa, Canada (1941), doi.org/10.4224/40001282.

[3] Northwood, T.D., Sound ‑ Insulation Ratings and the New ASTM Sound ‑ Transmission Class, The Journal of the Acoustical Society of America, 34(4), 493‑501, (1962).

[4] National Research Council of Canada. Associate Committee on the National Building Code, "National Building Code of Canada: 1960", National Research Council of Canada. Associate Committee on the National Building Code, Ottawa, Canada (1963), doi.org/10.4224/40001400.

[5] National Research Council of Canada. Associate Committee on the National Building Code, "National Building Code of Canada: 1970", National Research Council of Canada. Associate Committee on the National Building Code, Ottawa, Canada (1971), doi.org/10.4224/40001705.

[6] National Research Council of Canada. Associate Committee on the National Building Code, "National Building Code of Canada: 1990", National Research Council of Canada, Ottawa, Canada (1992), doi.org/10.4224/40001453.

[7] Canadian Commission on Building and Fire Codes, "National Building Code of Canada: 2015", National Research Council of Canada, Ottawa, Ontario (2015), DOI: 10.4224/40002005.

[8] Nightingale, T.R.T. and Sultan, M.A., Sound isolation and fire resistance of assemblies with fire stops, Construction Technology Update; No. 16, (1998).

[9] Nightingale, T.R.T. and Halliwell, R.E., "Flanking Transmission at Joints in Multi-Family Dwellings. Phase 1: Effects of Fire Stops at Floor/Wall Intersections", National Research Council Canada, Ottawa, Canada (1997), http://dx.doi.org/10.4224/20331633.

[10] Nightingale, T.R.T., Halliwell, R.E., and Quirt, J.D., "Flanking Transmission in Multi-Family Dwellings Phase II: Effects of Continuous Structural Elements at Wall/Floor Junctions", National Research Council Canada, Ottawa, Canada (2002), http://dx.doi.org/10.4224/20386172.

[11] Nightingale, T.R.T., Halliwell, R.E., Quirt, J.D., and King, F., "Flanking Transmission at the Wall/Floor Junction in Multifamily Dwellings - Quantification and Methods of Suppression", National Research Council Canada, Ottawa, Canada (2005), http://doi.org/10.4224/20377682.

[12] Nightingale, T.R.T., Quirt, J.D., King, F., and Halliwell, R.E., "Flanking Transmission in Multi-Family Dwellings: Phase IV", National Research Council Canada, Ottawa, Canada (2006), http://dx.doi.org/10.4224/20377185.

[13] Nightingale, T.R.T., Application of the CEN draft building acoustics prediction model to a lightweight double leaf construction, Building Acoustics, 46(3), 265–284, (1995).

[14] "BS EN 12354-1:2000 Building acoustics. Estimation of acoustic performance in buildings from the performance of elements. Airborne sound insulation between rooms", BSI, London, UK (2000).

[15] Gerretsen, E. and Nightingale, T.R.T., Prediction Models in Building Acoustics: Introduction to the Special Session at Forum Acusticum 1999 in Berlin, Building Acoustics, 6(3), 151–158, (1999).

[16] "ISO 10848-1:2017 - Acoustics - Laboratory measurement of the flanking transmission of airborne and impact sound between adjoining rooms - Part 1: Frame document", International Standards Organization, Geneva, Switzerland (2017).

[17] "ISO 10848-3:2017 - Acoustics - Laboratory measurement of the flanking transmission of airborne and impact sound between adjoining rooms - Part 3: Application to Type B elements when the junction has a substantial influence", International Standards Organization, Geneva, Switzerland (2017).

[18] Nightingale, T.R.T. and Halliwell, R.E., Measuring the in-situ airborne sound insulation using the acoustic intensity technique, in: Proceedings of Acoustics Week in Canada 1999, Victoria, B.C. (1999), pp. 56–57.

[19] Estabrooks, T., King, F., Nightingale, T.R.T., and Sabourin, I., NRC-IRC Flanking Sound Transmission Facility, in: Proceedings of Acoustics Week in Canada 2009, Niagara-on-the-Lake, Ontario (2009).

[20] Bradley, J.S., Zeitler, B., and Schoenwald, S., Objective and subjective assessment of lightweight wood-framed floor assemblies in response to footstep and low-frequency impact sounds, in: Proceedings of InterNoise 2011, Osaka, Japan (2011).

[21] Schoenwald, S., Zeitler, B., and Nightingale, T.R.T., "Influence of receive room properties on impact sound pressure level measured with heavy impact sources.", National Research Council of Canada, Ottawa, Canada (2010).

[22] Schoenwald, S., Zeitler, B., and Nightingale, T.R.T., Dependency between standardized heavy impact sound pressure level and receiving room properties, in: Proceedings of InterNoise 2011, Osaka, Japan (2011).

[23] "JIS A 1418-2 Acoustics — Measurement of floor impact sound insulation of buildings — Part 2: Method using standard heavy impact sources", Japanese Standards Association (JSA), Tokyo, Japan (2019).

[24] "KS F 2810-2 Method for field measurement of floor impact sound insulation —part 2: method using standardized heavy impact sources", Korean Industrial Standards, (2001).

[25] Quirt, J.D., Nightingale, T.R.T., and King, F., "RR-219: Guide for Sound Insulation in Wood Frame Construction", National Research Council Canada, Ottawa, Canada (2006), http://doi.org/10.4224/20377048.

[26] Nightingale, T.R.T., New design guide for controlling flanking sound, Solplan Review, (January 132), 18–19, (2007).

[27] soundPATHS Calculator, https://soundpaths.nrc-cnrc.gc.ca/#/

[28] Quirt, J.D., Controlling Air-borne and Structure-borne Sound in Buildings, in: Proceedings of InterNoise 2009, Ottawa, Canada (2009).

CANADIAN ACOUSTICAL ASSOCIATION

Minutes of the Board of Directors Meeting
Wednesday, 17 May 2023 13:00 PM – 17:00 PM (EDT)
by Zoom videoconference

1. Call to Order

Meeting called to order at ~13:00 PM (EDT).

Present online: Jérémie Voix (chair), Umberto Berardi, Victoria Duda, Bill Gastmeier, Bryan Gick, Dalila Giusti, Michael Kiefte, Hugues Nélisse, Roberto Racca, Frank Russo, Mehrzad Salkhordeh.

2. President's Report (Jérémie Voix)

Jérémie reported on the recent migration of the Association's websites; the main CAA site theme was refactored (outsourced in India) and a new website running the latest version of PHP and WordPress was ready to be launched (http://test.caa-aca.ca/), thanks to the work of webmaster Philip Tsui. As part of an ongoing initiative to enhance the CAA's documentation, a Wiki has been built starting with Michael Kiefte's efforts to revamp the Association's old Operations Manual. Access permissions still need to be finalized but the site was demonstrated live to the directors. Jérémie also reported that the migration of the OJS site from OpenJournalSystem.com to PKP, completed in July 2022, had been working without issues since that time. Migration of scripts (for example, to generate listings of BoD, sustaining subscribers, advertisers, members, etc.) was just re-finalized by Cécile Le Cocq in early April 2023. The budget of $700 (10h) was entirely allocated to Cécile's work to program the automation system and shall be expensed.

Moving on to new business, Jérémie announced some changes in roles within the Association that had taken place or would be proposed. He informed the Board that Bernard Feder had resigned from the position of Advertising Coordinator on April 28th after holding that role continuously since 2014. The Board expressed their gratitude for Bernard's dedicated support. Jérémie indicated that he would take up the role himself, as the new automated Ad Management system simplified the task considerably. He then mentioned that he had been actively seeking a successor as President and that Umberto would be willing to accept the role if someone would take up his duties as Editor-in-Chief. Jérémie proposed that a switch in their respective positions be put to the approval of the CAA members at the next AGM.

Next, Jérémie outlined a proposal to introduce fixed 3-year mandates for the executive officers' positions, the first year being an "elect" role and the last year a "past" role both assisting the current active officer, with executive mandates overlapping each other staggered by one year. In an extensive discussion which dealt with concepts such as having a "nomination committee," incorporating equality, diversity, and inclusiveness (EDI) considerations in the selections, etc., broad consensus emerged for a suggestion made by Umberto to double the proposed mandate duration (with 2 years in "elect," "current" and "past" position) for stronger continuity. It was agreed that procedural details of the proposed executive mandate structure would be elaborated in the period leading to the Fall BoD meeting and reviewed at that event. Frank suggested that when putting forward the scheme at the subsequent AGM, a Gantt chart will be presented showing the mandates of current directors (normally, a maximum of six consecutive one-year terms) to encourage members to consider filling positions as they become vacant. Roberto advised including, as part of the registration process for AWC2023, an option for delegates to express interest in joining the CAA Board.

Jérémie concluded his report by mentioning that the two active CAA Local Chapters for the Greater Toronto Area (GTA) and Montreal are now using Google Groups as their communication platform, so people interested in either group can simply register at https://groups.google.com/u/1/g/caa-gta-local-chapter for GTA (moderated by Mikk

Tomme) or https://groups.google.com/u/1/g/caa-montreal-chapter for Montreal (moderated by Romain Dumoulin and Raphael Duée).

3. **Treasurer's Report** (Dalila Giusti)

Given that no formal budget presentation or approval takes place at the Spring BoD meeting, Dalila only gave an informational overview of the Association's finances. She characterized them as being in good shape and likely to benefit from the current rise in interest rates. She noted that the bulk of the liquid assets were held in investment funds with fixed maturity terms (3 to 5 years) and guaranteed principal, so that the capital was never at risk; based on past performance the rate of return over the term to maturity could be as high as 30%. Of five GIC investments currently held, the three accounting for most of the invested capital would be maturing between 2024 and 2027. Dalila indicated that the tax return for the 2022 fiscal year had to be filed and was due by June 30, 2023.

4. **Secretary's Report** (Roberto Racca)

Roberto opened his presentation with a tally of memberships, which showed a precipitous drop in paid-up regular members to a mere 73 – a loss of more than 50 from the number reported at the Fall 2022 Board meeting and of about 70 year-on-year from Spring 2022. The number of student members, at 24, had picked up from a dip the previous Fall to slightly exceed the level of a year earlier, but there had been a disconcerting loss in sustaining subscribers: only 9 had renewed their support out of the 20 that were current as of the Fall 2022 Board meeting. Roberto speculated that while the drop could be attributable to people having become disillusioned with membership because of the erratic publication of the journal in 2022 (due to overwhelming technical challenges), a more likely scenario was that renewals had not taken place because the automated reminder e-mails either had not been received or had been overlooked. He pointed out the revealing fact that among those who had failed to renew were several directors, people strongly vested in the CAA whom only lack of reminders would have caused to lapse (this announcement was met with shocked realization and an immediate flurry of online renewals).

On a similar note, Roberto remarked that about half of the seven advertisers who had placed insertions in the previous year of Canadian Acoustics (the first to feature the online purchase of ad space through the journal site as a form of yearly "subscription") had not done so for the 2023 publication year. Although this might be partly in reaction to the erratic publishing in 2022, he noted, it is more likely reflected lack of reminding (again, missed auto-notification e-mails?) and follow-up regarding the online renewal process. Roberto advised that directly reaching out to the advertisers would be important whether to allay concerns they might have about publication regularity or to simply prompt them on the renewal schedule.

Discussion followed mostly on the matter of potential pitfalls in the membership expiration reminder and the purchase / renewal process via the OJS portal that could have led to the mass drop in paid-up memberships. Roberto noted that various hurdles could be contributing singularly or in combination with the failure of notifications to be seen and acted upon: missed or delayed sending of automated e-mails, interception of the same by increasingly draconian spam filters, and ultimately e-mail fatigue. To help counter the latter, he suggested redesigning the e-mailed notifications with an attractive and recognizable-looking HTML based layout, to make them stand out in inboxes and preview panes. He further noted that the current way of simply linking from the e-mail to the OJS renewals page (which is thwarted by the authentication stage, forcing users to figure out their way ahead) is a prominent cause of frustration, as the regular help requests that he receives attest to, and potential abandonment. He recommended implementing in the notifications a "Renew now" button taking directly to the secure payment portal with applicable user details transferred programmatically. Jérémie indicated that he would explore these options with OJS software developers, as well as investigating possible causes of non-delivery of renewal notifications.

5. **Awards Report** (Victoria Duda)

Victoria informed the Board that a reminder about the prizes application deadline of April 30 had been posted two weeks before that date on the Association's Twitter and LinkedIn channels and e-mailed to the awards coordinators; however, she noted that an initial typo in the reminder text (later corrected) that indicated May 30 as the deadline might have elicited some confusion. She then provided an update on the status of the various awards requiring submissions, the majority of which had received at least 2 applications (up to a high of 5) save for the Bregman Student Prize and Northwood Student Prize for which no entries had been received. The Hétu Prize in Acoustics formerly coordinated by the late Alberto Behar still needed a new coordinator to be appointed. The Canada-Wide Science Fair Award in Acoustics remained to be adjudicated, as did the Student Presentation Awards connected with the upcoming annual conference in the autumn. Following an earlier recommendation of the Board the monetary value of most awards had been raised to $650 from the prior $500, and the total pool of all prizes amounted to $11,400. Victoria announced the rekindling of the Directors' Award for best student paper and best regular paper published in Canadian Acoustics over the prior year; she presented the list of qualifying articles and invited as many members of the Board as were able to participate in the scoring process over the coming weeks.

6. **Editor's Report** (Umberto Berardi)

Umberto informed the Board that all initial production difficulties with the new company now printing Canadian Acoustics had been solved. He said that the main challenge currently facing the editorial team was the small number of submissions received as authors prioritized other journals over Canadian Acoustics; in particular, the journal does not yet seem to be a target for researchers in some disciplines such as audiology or bioacoustics. Umberto provided an update on the status of the current publication year:

- Journal issue vol. 51 - n.1/2023 was ready to be published (4 articles accepted and at the stage of copyediting)
- Journal issue vol. 51 - n.2/2023 would be a regular issue (5 papers already in the second round of review, so no problems were anticipated in having content)
- Journal issue vol. 51 - n.3/2023 would be the AWC proceedings issue – so it was expected that this would be a strong issue as customary for AWC.

Umberto noted that it had been challenging to train members of the editorial board to work with the new production workflow system. He also raised the recurrent theme that there were still gaps in the editorial board, as it had been difficult to recruit managing editors, enlist reviewers, and attract paper submissions in the fields of bioacoustics, physical acoustics / ultrasound, and underwater acoustics.

Umberto also recommended that communications with awards recipients mention the invitation to publish an article on their research in Canadian Acoustics.

7. **Social Media Editor Report** (Romain Dumoulin)

No formal report was presented; Jérémie briefly updated the Board on behalf of Romain on the following of the Association's channels on LinkedIn and Twitter and steadily ongoing posting of content.

8. **Past and Upcoming Meetings**

a. **ISO TC43 Plenary Montréal 2023** (Jérémie Voix, ÉTS)

Jérémie gave a short summary of the event, which had just taken place a few days before (2-6 May) at the École de technologie supérieure (ÉTS) in Montréal. The event had brought together the members of the TC43 standards subcommittees SC1 (Noise), SC2 (Building Acoustics) and SC3 (Underwater Acoustics) and was well attended by delegates representing TC43 participating countries worldwide.

b. **AWC 2023: Montréal** (Olivier Doutres, ÉTS – Joined Zoom meeting temporarily)

Olivier reported that all preparations were on track for the conference, to be held 3-6 October 2023 at the Hotel Bonaventure in Montréal. He indicated that 156 abstracts had been received to date, a promising sign for strong attendance and technical content given that the abstract submission window would still be open until 1 July. Discussion with members of the Board focused on how the 10,000$ student travel subsidies should most effectively be allocated; suggestions were aimed at encouraging participation from students outside of the province or at any rate farther away than the main urban areas of Montréal, Québec, and Sherbrooke.

c. **AWC/ASA 2024: Ottawa** (Joana Rocha and Sebastian Ghinet)

This proposed event would be jointly hosted with ASA in Spring 2024 in Ottawa.

d. **AWC 2025: Vancouver/Banff** (Umberto Berardi / Fitsum Tariku)

Tentative.

9. Varia

No new matters were raised.

10. Next Meeting

Agreed on a physical meeting on 3 October afternoon in Montréal for Board members attending AWC, with other members joining by video conference.

11. Motion to Adjourn

By Jérémie, at ~17:00 PM (EDT).

Canadian Acoustical Association
Association canadienne d'acoustique

CANADIAN ACOUSTICAL ASSOCIATION ASSOCIATION CANADIENNE D'ACOUSTIQUE

2023 PRIZE WINNERS / RÉCIPIENDAIRES DES PRIX 2023

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

Pierre Grandjean (Université de Sherbrooke)

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

Xinyi Zhang (École de technologie supérieure)

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

Karlee Zammit (University of Victoria)

ECKEL GRADUATE STUDENT PRIZE IN NOISE CONTROL / PRIX ETUDIANT ECKEL EN CONTROLE DU BRUIT

Lucie Gallerand (École de technologie supérieure)

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

Niki Diogouof (University of Victoria)

DIRECTORS' AWARDS / PRIX DES DIRECTEURS

Don Nguyen (McGill University)
Peter Waudby-Smith (Aiolos Engineering)

CONGRATULATIONS / FÉLICITATIONS!


LARSON DAVIS

ACOUSTICS WEEK IN CANADA

MONTRÉAL, QUÉBEC, OCTOBER 3-6, 2023

The **Acoustics Week in Canada will be held from October 3-6, 2023** in downtown Montreal, Quebec. You are invited to be part of this three days conference featuring the latest developments in Canadian acoustics and vibration. The keynote talks and technical sessions will be framed by a welcome reception, conference banquet, technical tour and an exhibition of products and services relating to the field of acoustics and vibration.

Place d'Armes, Old Montréal

Jacques Cartier's Bridge

Take a few days before or after the conference to enjoy the area! Quebec is famous for its fall colors, when trees all over the place turn bright shades of red, orange, and yellow before losing their leaves. It's an annual spectacle that draws tourists from around the world and remains impressive even to those of us who see it every year! Montreal still has important events to offer at this time of year such as the OFF Jazz Festival and the Festival du nouveau cinema.

Venue and Accommodation

The conference will be held at the Plaza in downtown Montreal (https://plazapmg.com/plaza-centre-ville/). A block of rooms is available at the Bonaventure hotel which is 5-minute walk from the conference center. A special conference rate is offered for reservations made under "AWC2023 conference" codename. Extend your stay and enjoy the local area at the same special rate. Please refer to the conference website for further registration details: https://awc.caa-aca.ca

Technical sessions

Plenary, technical, and workshop sessions are planned throughout the conference. More than 160 abstracts have been submitted, and we would like to express our gratitude to all who have participated. Each day will begin with a keynote talk of broader interest and relevance to the acoustics community. Technical sessions are planned to cover all areas of acoustics including:

ACOUSTICAL MATERIALS AND METAMATERIALS / AEROACOUSTICS / ARCHITECTURAL AND BUILDING ACOUSTICS/ ARTIFICIAL INTELLIGENCE IN ACOUSTICS / BIO-ACOUSTICS AND BIOMEDICAL ACOUSTICS / MUSICAL ACOUSTICS / NOISE AND NOISE CONTROL / PHYSICAL ACOUSTICS / PSYCHO- AND PHYSIO-ACOUSTICS / SHOCK AND VIBRATION / SIGNAL PROCESSING / SPEECH SCIENCES AND HEARING SCIENCES / STANDARDS AND GUIDELINES IN ACOUSTICS / ULTRASONICS / UNDERWATER ACOUSTICS

Plenary speakers

Three renowned scientists will grace us with the honor of presenting their work during the keynote sessions:

- Prof. Noureddine Atalla (University of Sherbrooke),
- Prof. Christian Giguère (University of Ottawa)
- Prof. Fabrice Marandola (McGill University).

Key dates and registration details

Register by August 15, 2023 for the lower early registration rates. Please refer to the conference web site: https://awc.caa-aca.ca

Scientific visits and social events

Exclusive scientific visits and entertaining social activities are planned:

- Visit 1: Exclusive tour of the Centre for Interdisciplinary Research in Music Media and Technology (CIRMMT)
- Visit 2: Exclusive tour of the ICAR laboratory (Infrastructure commune en acoustique pour la recherche ÉTS-IRSST) housed at ÉTS.
- Visit 3: Exclusive visit of the Maison symphonique.
- Before and after the Gala Dinner: Powerpoint Karaoke and Jam Session
- During the 3-days of the conference, contribute to the construction of a noise map of Montreal using the "NoiseCapture" application.
- Special Pizza&Beer event for students and volunteers

Exhibition and sponsorship

We express our gratitude to the community-minded Diamond and Silver sponsors for their valuable support. We are excited to announce that an exhibition area will be available on Wednesday, October 4, and Thursday, October 5, showcasing various acoustical equipment, products, and services. Many exhibitors have already registered and will be present at the conference exhibition, providing a great opportunity for in-person and hands-on interaction!

Diamond sponsor	Silver sponsor	Exhibitors
PLITEQ	LogiSon Acoustic Network SVS Sound & Vibration Solutions Canada Inc. irsst	Dalimar Instruments Scantek A7hz HBK Mecanum Soft dB Environmental Noise Control FPInnovations ramo

The conference still offers opportunities for suppliers of products and services to engage the acoustic community through exhibition and sponsorship (Gold level). Companies and organizations that are interested in participating in the exhibition should contact the Exhibition and Sponsorship coordinator for an information package. Suppliers who have not yet reserved their spot should do so promptly, as spaces are filling up quickly.

Students

Students are strongly encouraged to participate. Students presenting papers will be eligible for one of three $500 Best Presentation prizes awarded. Conference bursaries will also be available to those students whose papers are accepted for presentation (see conditions on the conference web site).

Contacts

Conference Chair	Olivier Doutres (ÉTS)	conference@caa-aca.ca
Technical Chair	Thomas Padois (IRSST)	technical-chair@caa-aca.ca
Exhibits and Sponsorship co-ordinator	Julien Biboud (MÉCANUM)	awc2023exhibitors@caa-aca.ca
Student Prizes and Subsidies	Victoria Duda (Univ. of Montréal)	awards@caa-aca.ca

SEMAINE CANADIENNE DE L'ACOUSTIQUE

MONTRÉAL, QUÉBEC, 3-6 OCTOBRE 2023

La Semaine Canadienne de l'acoustique aura lieu du 3 au 6 octobre 2023 au centre-ville de Montréal, au Québec. Vous êtes invités à assister à cette conférence de trois jours durant laquelle les derniers développements en matière d'acoustique et de vibration au Canada seront présentés. Chaque journée débutera par une conférence plénière qui sera suivie de sessions thématiques. Vous pourrez échanger lors de la réception de bienvenue, du dîner de gala mais aussi lors des différentes pauses avec les exposants qui vous présenteront les nouveaux produits et services liés à l'acoustique et à la vibration.

Place d'Armes, Vieux-Montréal

Pont Jacques Cartier

Prenez quelques jours avant ou après la conférence pour profiter de la région! Le Québec est célèbre pour ses couleurs d'automne, lorsque les arbres prennent des teintes vives de rouge, d'orange et de jaune avant de perdre leurs feuilles. C'est un spectacle annuel qui attire des touristes du monde entier et qui reste impressionnant même pour ceux d'entre nous qui le voient chaque année ! Montréal garde aussi quelques évènements de marque à cette période de l'année comme l'OFF Festival de Jazz et le Festival du nouveau cinéma.

Lieu et hébergement

La conférence se déroulera au Plaza dans le centre-ville de Montréal (https://plazapmg.com/plaza-centre-ville/). Des chambres seront disponibles à l'hôtel Bonaventure, à 5 minutes à pied du centre de conférence, avec un tarif spécial pour les réservations faites sous le nom "AWC2023 conference". Prolonger votre séjour à l'hôtel au même tarif afin de profiter du centre-ville et de la région. Veuillez consulter le site web de la conférence pour plus d'informations sur l'inscription : https://awc.caa-aca.ca

Sessions techniques

Des sessions plénières, techniques et des ateliers sont prévues tout au long de la conférence. Plus de 160 résumés ont été soumis, et nous tenons à exprimer notre gratitude à tout.e.s les participant.e.s. Les sessions techniques couvriront tous les domaines de l'acoustique :

AÉROACOUSTIQUE / ACOUSTIQUE DU BÂTIMENT ET ARCHITECTURALE / ACOUSTIQUE BIOMÉDICALE / ACOUSTIQUE MUSICALE/ ACOUSTIQUE PHYSIQUE / ACOUSTIQUE SOUS-MARINE / AUDIOLOGIE / BIOACOUSTIQUE / BRUIT ET CONTRÔLE DU BRUIT / CHOCS ET VIBRATIONS / INTELLIGENCE ARTIFICIELLE EN ACOUSTIQUE / LINGUISTIQUE / MATÉRIAUX ET MÉTAMATÉRIAUX ACOUSTIQUES / NORMES EN ACOUSTIQUE / PSYCHOACOUSTIQUE / TRAITEMENT DU SIGNAL / ULTRASONS

Conférenciers invités

Trois scientifiques de renom nous feront l'honneur de présenter leurs travaux lors des séances plénières:

- Prof. Noureddine Atalla (Université de Sherbrooke),
- Prof. Christian Giguère (Université d'Ottawa)
- Prof. Fabrice Marandola (Université McGill).

Dates clés et détails pour l'inscription

Inscrivez-vous avant le 15 août 2023 pour bénéficier du tarif préférentiel. Veuillez consulter le site web de la conférence : https://awc.caa-aca.ca

Visites scientifiques et événements sociaux

Des visites scientifiques exclusives et des activités sociales divertissantes sont prévues :

- Visite 1 : Visite exclusive du Centre interdisciplinaire de recherche en musique, médias et technologie (CIRMMT)
- Visite 2 : Visite exclusive du laboratoire ICAR (Infrastructure commune en acoustique pour la recherche ÉTS-IRSST) hébergé à l'ÉTS.
- Visite 3 : Visite exclusive de la Maison symphonique (Place des arts).
- Avant et après le dîner de gala : Powerpoint Karaoke et sessions d'improvisation musicale (Jam Session)
- Pendant les 3 jours de la conférence, contribuez à la production d'une carte de bruit de Montréal en utilisant l'application "NoiseCapture" sur votre téléphone cellulaire.
- Évènement spécial Pizza & Bière pour les étudiants et les bénévoles.

Expositions et commandites

Nous remercions chaleureusement nos sponsors Diamant et Argent pour leur soutien précieux. Nous sommes heureux d'annoncer qu'une zone d'exposition sera disponible le mercredi 4 octobre et le jeudi 5 octobre où des équipements, produits et services acoustiques vous y seront présentés. De nombreux exposants se sont déjà inscrits et seront présents dans la zone d'exposition. Une excellente occasion de rencontrer du monde de la communauté et d'apprendre sur les pratiques actuelles !

Commanditaire Diamant	Commanditaire Argent	Exposants
PLITEQ	LogiSon Acoustic Network SVS Sound & Vibration Solutions Canada Inc. irsst	Dalimar Instruments Scantek A7hz HBK Hottinger Brüel & Kjær Mecanum Soft dB Environmental Noise Control FPInnovations ramo

La conférence offre toujours aux fournisseurs de produits et de services la possibilité de s'adresser à la communauté acoustique par le biais d'une table d'exposition ou d'une commandite (commandite niveau Or). Les entreprises et les organisations intéressées doivent contacter le coordinateur des expositions et du parrainage pour obtenir un dossier d'information. Ceux qui n'ont pas encore réservé leur place doivent le faire rapidement, car il en reste peu.

Étudiant·e·s

Les étudiant·e·s sont vivement encouragé·e·s à participer à la conférence. Les étudiant·e·s présentant un article seront éligibles pour obtenir un prix de $500 décerné aux 3 meilleures présentations. Des bourses de participation seront également offertes aux étudiant·e·s dont les communications sont acceptées pour présentation (voir conditions sur le site de la conférence).

Contacts

Président de la conférence	Olivier Doutres (ÉTS)	conference@caa-aca.ca
Président technique	Thomas Padois (IRSST)	technical-chair@caa-aca.ca
Coordonnateur des expositions et du parrainage	Julien Biboud (MÉCANUM)	awc2023exhibitors@caa-aca.ca
Prix étudiants et subventions	Victoria Duda (Univ. de Montréal)	awards@caa-aca.ca

CANADIAN ACOUSTICS ANNOUNCEMENTS - ANNONCES TÉLÉGRAPHIQUES DE L'ACOUSTIQUE CANADIENNE

Looking for a job in Acoustics?

There are many job offers listed on the website of the Canadian Acoustical Association!

You can see them online, under http://www.caa-aca.ca/jobs/

August 5th 2015

INTER-NOISE 2023 to be held August 20-23, 2023, in Makuhari Messe (Japan)

We are very pleased to inform you that the website of INTER-NOISE 2023 has been launched. Its link is https://internoise2023.org/.

The INTER-NOISE 2023 is held at Makuhari Messe (https://www.m-messe.co.jp/en/) from August 20-23, 2023, which is sponsored by International Institute of Noise Control Engineering (I-INCE) and is co-organized by Institute of Noise Control Engineering of Japan (INCE/J), Acoustical Society of Japan (ASJ).

August 12th 2022

AWC2023 to be held in Montreal (QC) October 3-6, 2023

The Acoustics Week in Canada will be held from October 3-6, 2023 in downtown Montreal, Quebec. For more information on registration, please visit the conference website: https://awc.caa-aca.ca

Dear Members and Friends of the Canadian Acoustical Association, The Acoustics Week in Canada will be held from October 3-6, 2023 in downtown Montreal, Quebec. You are invited to be part of this three-day conference featuring the latest developments in Canadian acoustics and vibration. The conference will be held at the Plaza in downtown Montreal . A block of rooms is available at the Bonaventure Hotel which is 5-minute walk from the conference centre. Here are some important dates to remember: Abstract submission deadline: June 15, 2023 Paper submission deadline: July 15, 2023 Registration starts: June 16, 2023 Registration deadline for proceeding papers: August 1st 2023 Late registration fees start: August 15, 2023 Plenary, technical, and workshop sessions are planned throughout the conference. Each day will begin with a keynote talk of broader interest and relevance to the acoustics community. Technical sessions are planned to cover all areas of acoustics including: ACOUSTIC METAMATERIAL /AEROACOUSTICS / ARCHITECTURAL AND BUILDING ACOUSTICS / BIO-ACOUSTICS AND BIOMEDICAL ACOUSTICS / EDUCATION IN ACOUSTICS / HEARING PROTECTION DEVICES / ARTIFICIAL INTELLIGENCE IN ACOUSTICS / MUSICAL ACOUSTICS / NOISE AND NOISE CONTROL / PHYSICAL ACOUSTICS / PSYCHO- AND PHYSIO-ACOUSTICS / SHOCK AND VIBRATION / SIGNAL PROCESSING / SPEECH SCIENCES AND HEARING SCIENCES / STANDARDS AND GUIDELINES IN ACOUSTICS / ULTRASONICS / UNDERWATER ACOUSTICS / For more information on registration, please visit the conference website: https://awc.caa-aca.ca Looking forward to seeing you there, Olivier Doutres (ÉTS, conference chair, conference@caa-aca.ca), Thomas Padois (IRSST, technical chair, technical-chair@caa-aca.ca) and Julien Biboud (Mécanum, exhibits and sponsorship coordinator, awc2023exhibitors@caa-aca.ca)

April 5th 2023

AWC2023 New abstract deadline: July 1st, 2023

La date limite a été prolongée jusqu'au 1er juillet pour les résumés de 300 mots : https://awc.caa-aca.ca/index.php/AWC/index/pages/view/AWC2023_AbstractInfo The deadline has been extended until July 1st for 250 words abstracts: https://awc.caa-aca.ca/index.php/AWC/index/pages/view/AWC2023_AbstractInfo

[VERSION FRANÇAISE CI-DESSOUS] The organization of the "Acoustics Week in Canada" (AWC23) is going well. We are pleased to announce that approximately 130 abstracts have already been submitted, and we would like to express our gratitude to all who have participated thus far. Here are some important reminders and updates: For those who have not yet submitted their abstracts, we are pleased to inform you that the deadline has been extended until July 1st for 300 words abstracts. You still have time to contribute and be a part of this exciting event, via https://awc.caa-aca.ca/index.php/AWC/index/pages/view/AWC2023_AbstractInfo Once your abstract has been accepted, please ensure that your two-page article is uploaded by July 15th for inclusion in the September proceedings issue of Canadian Acoustics. It is important to note that at least one author must be registered for the conference by August 1st for the article to be published, otherwise only the abstract will be included in the proceedings issue with Canadian Acoustics journal. Please be aware that late registration fees will come into effect starting August 15th. We hope that you have a wonderful summer vacation and look forward to your active participation in AWC23. Prof. Olivier Doutres [VERSION FRANÇAISE] L'organisation de la "Semaine canadienne de l'acoustique" (AWC23) avance bien. Nous sommes heureux de vous annoncer que près de 130 résumés ont déjà été soumis, et nous tenons à exprimer notre gratitude à tous ceux qui ont participé jusqu'à présent. Voici quelques rappels et mises à jour importants : Pour ceux qui n'ont pas encore soumis leur résumé, nous avons le plaisir de vous informer que la date limite a été prolongée jusqu'au 1er juillet pour les résumés de 300 mots. Vous avez encore le temps de contribuer et de participer à cet événement passionnant, via https://awc.caa-aca.ca/index.php/AWC/index/pages/view/AWC2023_AbstractInfo Une fois que votre résumé aura été accepté, veuillez vous assurer de téléverserr votre article de deux pages d'ici le 15 juillet pour qu'il soit inclus dans le numéro de septembre des actes du journal Acoustique canadienne. Il est important de noter qu'un auteur doit être inscrit à la conférence d'ici le 1er août pour que l'article soit publié; faute de quoi, seul le résumé sera inclus dans les actes de la conférence. Veuillez noter que des frais d'inscription tardive seront appliqués à partir du 15 août. Nous espérons que vous passerez de merveilleuses vacances d'été et nous nous réjouissons de votre participation active à l'AWC23. Prof. Olivier Doutres

June 19th 2023

[AWC23] Important reminders and updates

The organization of the "Acoustics Week in Canada" (AWC23) is still going well. We are pleased to announce that over 160 abstracts have been received, and we would like to express our gratitude to all who have participated.

Here are some important reminders and updates:. We can still accept abstracts, exceptionally, if the authors contact us ASAP.. The 2-page article must be submitted by August 1st. At least one author must be registered for the conference by August 1st for the abstract/article to be published in the proceedings issue with Canadian Acoustics journal.. Please be aware that late registration fees will come into effect starting August 15th.. Note that conference fees are reduced for CAA-ACA members. Membership is available by registering via the association's website, at the link https://jcaa.caa-aca.ca/index.php/jcaa/user/register . If you are already registered or a CAA-ACA member, please login at https://jcaa.caa-aca.ca/index.php/jcaa/login and check if your membership is still in good standing. It's quite possible that you missed the reminders and unintentionally let your membership period expire.. Three renowned scientists will grace us with the honor of presenting their work during the keynote sessions : Noureddine Atalla (University of Sherbrooke), Christian Giguère (University of Ottawa) and Fabrice Marandola (McGill University). . Fantastic scientific visits and social activities are organized : https://awc.caa-aca.ca/index.php/AWC/index/pages/view/AWC2023_SocialEvents We hope that you have (or had) a wonderful summer vacation and look forward to seeing you in October. Prof. Olivier Doutres

July 27th 2023

Acoustics 2023 (Sydney, Australia)

Early Bird Registration Deadline 5 September, 2023! Make sure you don't miss the lower rate early bird deadline by completing your registration now! There will be no extension to this deadline so do not delay.

On behalf of the Australian Acoustical Society and The Acoustical Society of America, the Organising Committee looks forward to welcoming you to the Acoustics 2023 Conference to be held Monday 4 December to Friday 8 December 2023 at the International Convention Centre Sydney (ICC Sydney), Australia. Early Bird Registration Deadline 5 September, 2023! Make sure you don't miss the lower rate early bird deadline by completing your registration now! There will be no extension to this deadline so do not delay. More information can be found online at https://acoustics23sydney.org/

August 23rd 2023

À la recherche d'un emploi en acoustique ?

De nombreuses offre d'emploi sont affichées sur le site de l'Association canadienne d'acoustique !

Vous pouvez les consulter en ligne à l'adresse http://www.caa-aca.ca/jobs/

August 5th 2015

La semaine AWC2023 aura lieu à Montréal (QC) du 3 au 6 octobre 2023

La Semaine canadienne de l'acoustique aura lieu du 3 au 6 octobre 2023 au centre-ville de Montréal, au Québec. Pour plus d'informations sur l'inscription, veuillez consulter le site Web de la conférence. https://awc.caa-aca.ca

Chèr.e.s membres et ami.e.s de l'Association canadienne d'acoustique, La Semaine canadienne de l'acoustique aura lieu du 3 au 6 octobre 2023 au centre-ville de Montréal, au Québec. Vous êtes invités à assister à cette conférence de trois jours durant laquelle les derniers développements en matière d'acoustique et de vibration au Canada seront présentés. La conférence se déroulera au Plaza Centre-Ville de Montréal. Des chambres seront disponibles à l'hôtel Bonaventure, à 5 minutes à pied du centre de conférence. Voici quelques dates importantes de la conférence : Soumission résumés :15 juin 2023 Soumission article de 2 pages : 15 juillet 2023 Ouverture inscription : 16 juin 2023 Limite inscription pour publication article : 1 août 2023 Ouverture inscription tardive : 15 août 2023 Des sessions plénières, techniques et des ateliers sont prévus tout au long de la conférence. Chaque journée débutera par une conférence plénière d'intérêt pour la communauté de l'acoustique. Des sessions techniques sont également prévues pour couvrir tous les domaines de l'acoustique, à savoir . ACOUSTIQUE DU BÂTIMENT ET ARCHITECTURALE / ACOUSTIQUE BIOMÉDICALE / ACOUSTIQUE MUSICALE / ACOUSTIQUE PHYSIQUE / ACOUSTIQUE SOUS-MARINE / AÉROACOUSTIQUE / AUDIOLOGIE / BIOACOUSTIQUE / BRUIT ET CONTRÔLE DU BRUIT / CHOCS ET VIBRATIONS / ENSEIGNEMENT DE L'ACOUSTIQUE / INTELLIGENCE ARTIFICIELLE EN ACOUSTIQUE / LINGUISTIQUE / MÉTAMATÉRIAUX ACOUSTIQUES/ NORMES EN ACOUSTIQUE / PSYCHOACOUSTIQUE / PROTECTEURS AUDITIFS / TRAITEMENT DU SIGNAL / ULTRASONS / Pour plus d'informations sur l'inscription, veuillez consulter le site Web de la conférence. https://awc.caa-aca.ca Au plaisir de vous voir à la conférence, Olivier Doutres (ÉTS, président de la conférence, conference@caa-aca.ca), Thomas Padois (IRSST, président technique, technical-chair@caa-aca.ca) and Julien Biboud (Mécanum, coordonnateur des expositions et du parrainage, awc2023exhibitors@caa-aca.ca)

April 5th 2023

AWC2023 : Extension au 1er juillet pour les résumés

La date limite a été prolongée jusqu'au 1er juillet pour les résumés de 300 mots : https://awc.caa-aca.ca/index.php/AWC/index/pages/view/AWC2023_AbstractInfo The deadline has been extended until July 1st for 250 words abstracts: https://awc.caa-aca.ca/index.php/AWC/index/pages/view/AWC2023_AbstractInfo

[ENGLISH MESSAGE BELOW] L'organisation de la "Semaine canadienne de l'acoustique" (AWC23) avance bien. Nous sommes heureux de vous annoncer que près de 130 résumés ont déjà été soumis, et nous tenons à exprimer notre gratitude à tous ceux qui ont participé jusqu'à présent. Voici quelques rappels et mises à jour importants : Pour ceux qui n'ont pas encore soumis leur résumé, nous avons le plaisir de vous informer que la date limite a été prolongée jusqu'au 1er juillet pour les résumés de 300 mots. Vous avez encore le temps de contribuer et de participer à cet événement passionnant, via https://awc.caa-aca.ca/index.php/AWC/index/pages/view/AWC2023_AbstractInfo Une fois que votre résumé aura été accepté, veuillez vous assurer de téléverserr votre article de deux pages d'ici le 15 juillet pour qu'il soit inclus dans le numéro de septembre des actes du journal Acoustique canadienne. Il est important de noter qu'un auteur doit être inscrit à la conférence d'ici le 1er août pour que l'article soit publié; faute de quoi, seul le résumé sera inclus dans les actes de la conférence. Veuillez noter que des frais d'inscription tardive seront appliqués à partir du 15 août. Nous espérons que vous passerez de merveilleuses vacances d'été et nous nous réjouissons de votre participation active à l'AWC23. Prof. Olivier Doutres [ENGLISH VERSION] The organization of the "Acoustics Week in Canada" (AWC23) is going well. We are pleased to announce that approximately 130 abstracts have already been submitted, and we would like to express our gratitude to all who have participated thus far. Here are some important reminders and updates: For those who have not yet submitted their abstracts, we are pleased to inform you that the deadline has been extended until July 1st for 300 words abstracts. You still have time to contribute and be a part of this exciting event, via https://awc.caa-aca.ca/index.php/AWC/index/pages/view/AWC2023_AbstractInfo Once your abstract has been accepted, please ensure that your two-page article is uploaded by July 15th for inclusion in the September proceedings issue of Canadian

Acoustics. It is important to note that at least one author must be registered for the conference by August 1st for the article to be published, otherwise only the abstract will be included in the proceedings issue with Canadian Acoustics journal. Please be aware that late registration fees will come into effect starting August 15th. We hope that you have a wonderful summer vacation and look forward to your active participation in AWC23. Prof. Olivier Doutres

June 19th 2023

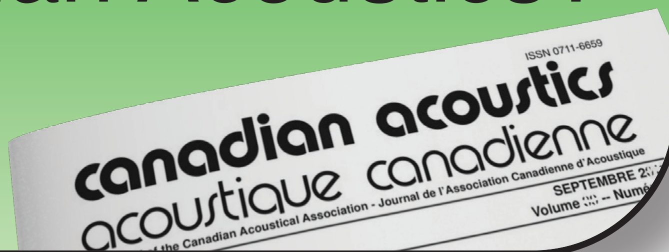
Rappels et mises à jour importants

L'organisation de la "Semaine canadienne de l'acoustique" (AWC23) avance toujours très bien. Nous sommes heureux de vous annoncer que plus de 160 résumés ont été reçus, et nous tenons à exprimer notre gratitude à tou.te.s les participant.e.s.

Voici quelques rappels et mises à jour importants :. Nous pouvons encore accepter des résumés, de manière exceptionnelle, si les auteurs nous contactent rapidement.. L'article de 2 pages doit être soumis d'ici le 1er août. Au moins un auteur doit être inscrit à la conférence d'ici le 1er août pour que l'article/résumé soit publié dans les actes de la conférence.. Veuillez noter que des frais d'inscription tardive seront appliqués à partir du 15 août.. Notez que les frais d'inscription à la conférence sont réduits pour les membres de la CAA-ACA. L'adhésion peut se faire en s'inscrivant via le site de l'association, au lien https://jcaa.caa-aca.ca/index.php/jcaa/user/register . Si vous êtes déjà inscrit ou membre de la CAA-ACA, veuillez vous connecter au lien https://jcaa.caa-aca.ca/index.php/jcaa/login et vérifier si votre adhésion est toujours en règle. Il est tout à fait possible que vous n'ayez pas vu les rappels et que vous ayez involontairement laissé expirer votre période d'adhésion.. Trois scientifiques de renom nous feront l'honneur de présenter leurs travaux lors des sessions plénières : Noureddine Atalla (Université de Sherbrooke), Christian Giguère (Université d'Ottawa) et Fabrice Marandola (Université McGill).. Des visites scientifiques passionnantes et des activités sociales sont prévues : https://awc.caa-aca.ca/index.php/AWC/index/pages/view/AWC2023_SocialEvents Nous espérons que vous passerez (ou avez passé) de merveilleuses vacances d'été et nous nous réjouissons de vous voir en Octobre. Prof. Olivier Doutres

July 27th 2023

Why publish in Canadian Acoustics?



Because, it is...

- A respected scientific journal with a 40-year history uniquely dedicated to acoustics in Canada
- A quarterly publication in both electronic and hard-copy format, reaching a large community of experts worldwide
- An Open Access journal, with content freely available to all, 12 months from time of publication
- A better solution for fast and professional review providing authors with an efficient, fair, and constructive peer review process.

Pourquoi publier dans Acoustique canadienne ?

Parce que, c'est...

- Une revue respectée, forte de 40 années de publications uniquement dédiée à l'acoustique au Canada
- Une publication trimestrielle en format papier et électronique, rejoignant une large communauté d'experts à travers le monde
- Une publication "accès libre" dont le contenu est disponible à tous, 12 mois après publication
- Une alternative intéressante pour une évaluation par les pairs, fournissant aux auteurs des commentaires pertinents, objectifs et constructifs

Application for Membership

CAA membership is open to all individuals who have an interest in acoustics. Annual dues total $120.00 for individual members and $50.00 for student members. This includes a subscription to ***Canadian Acoustics***, the journal of the Association, which is published 4 times/year, and voting privileges at the Annual General Meeting.

Subscriptions to *Canadian Acoustics* *or* Sustaining Subscriptions

Subscriptions to ***Canadian Acoustics*** are available to companies and institutions at a cost of $120.00 per year. Many organizations choose to become benefactors of the CAA by contributing as Sustaining Subscribers, paying $475.00 per year (no voting privileges at AGM). The list of Sustaining Subscribers is published in each issue of ***Canadian Acoustics*** and on the CAA website.

Please note that online payments will be accepted at http://jcaa.caa-aca.ca

Address for subscription / membership correspondence:

Name / Organization ____________________

Address ____________________

City/Province ____________ Postal Code ________ Country ____________

Phone ____________ Fax ____________ E-mail ____________

Address for mailing Canadian Acoustics, if different from above:

Name / Organization ____________________

Address ____________________

City/Province ____________ Postal Code ________ Country ____________

Areas of Interest: (Please mark 3 maximum)

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

For student membership, please also provide:

________ ________ ________ ________

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

I have enclosed the indicated payment for:
[ ] CAA Membership $ 120.00
[ ] CAA Student Membership $ 50.00

Corporate Subscriptions (4 issues/yr)
[ ] $120 including mailing in Canada
[ ] $128 including mailing to USA,
[ ] $135 including International mailing

[ ] Sustaining Subscription $475.00
(4 issues/yr)

Please note that the preferred method of payment is by credit card, online at http://jcaa.caa-aca.ca

For individuals or organizations wishing to pay by check, please register online at *http://jcaa.caa-aca.ca* and then mail your check to:

Executive Secretary, Canadian Acoustical:
Dr. Roberto Racca
c/o JASCO Applied Sciences
2305–4464 Markham Street
Victoria, BC V8Z 7X8 Canada

Formulaire d'adhésion

L'adhésion à l'ACA est ouverte à tous ceux qui s'intéressent à l'acoustique. La cotisation annuelle est de 120.00$ pour les membres individuels, et de 50.00$ pour les étudiants. Tous les membres reçoivent ***l'Acoustique Canadienne***, la revue de l'association.

Abonnement pour la revue *Acoustique Canadienne* et abonnement de soutien

Les abonnements pour la revue ***Acoustique Canadienne*** sont disponibles pour les compagnies et autres établissements au coût annuel de 120.00$. Des compagnies et établissements préfèrent souvent la cotisation de membre bienfaiteur, de 475.00$ par année, pour assister financièrement l'ACA. La liste des membres bienfaiteurs est publiée dans chaque issue de la revue ***Acoustique Canadienne***..

Notez que tous les paiements électroniques sont acceptés en ligne http://jcaa.caa-aca.ca

Pour correspondance administrative et financière:

Nom / Organisation ____

Adresse ____

Ville/Province ____________________ Code postal __________ Pays ____________________

Téléphone ____________ Téléc. ____________ Courriel ____________________

Adresse postale pour la revue Acoustique Canadienne

Nom / Organisation ____

Adresse ____

Ville/Province ____________________ Code postal __________ Pays ____________________

Cocher vos champs d'intérêt: (maximum 3)

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

Prière de remplir pour les étudiants et étudiantes:

__________ ______________ ____________________ __________

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

Cocher la case appropriée:
[ ] Membre individuel 120.00 $
[ ] Membre étudiant(e) 50.00 $

Abonnement institutionnel
[ ] 120 $ à l'intérieur du Canada
[ ] 128 $ vers les États-Unis
[ ] 135 $ tout autre envoi international

[ ] Abonnement de soutien 475.00 $ (comprend l'abonnement à *L'acoustique Canadienne*)

Merci de noter que le moyen de paiement privilégie est le paiement par carte crédit en ligne à *http://jcaa.caa-aca.ca*

Pour les individus ou les organisations qui préféreraient payer par chèque, l'inscription se fait en ligne à *http://jcaa.caa-aca.ca* puis le chèque peut être envoyé à :

Secrétaire exécutif, Association canadienne d'acoustique :
Dr. Roberto Racca
c/o JASCO Applied Sciences
2305–4464 Markham Street
Victoria, BC V8Z 7X8 Canada

Canadian Acoustical Association
Association canadienne d'acoustique

Board of Directors - Conseil d'administration

OFFICERS - OFFICIERS

President
Président

Jérémie Voix
ÉTS, Université du Québec
president@caa-aca.ca

Executive Secretary
Secrétaire

Roberto Racca
JASCO Applied Sciences
secretary@caa-aca.ca

Treasurer
Trésorier

Dalila Giusti
Jade Acoustics Inc.
treasurer@caa-aca.ca

Editor-in-Chief
Rédacteur en chef

Umberto Berardi
Ryerson University
editor@caa-aca.ca

DIRECTORS - ADMINISTRATEURS

Victoria Duda
Université de Montréal
victoria.duda@umontreal.ca

Bill Gastmeier
HGC Engineering
bill@gastmeier.ca

Bryan Gick
University of British Columbia
gick@mail.ubc.ca

Michael Kiefte
Dalhousie University
mkiefte@dal.ca

Andy Metelka
SVS Canada Inc.
ametelka@cogeco.ca

Hugues Nelisse
Institut de Recherche Robert-Sauvé en Santé et Sécurité du Travail (IRSST)
hugnel@irsst.qc.ca

Joana Rocha
Carleton University
Joana.Rocha@carleton.ca

Mehrzad Salkhordeh
dB Noise Reduction Inc.
mehrzad@dbnoisereduction.com

Upcoming Conference Chair
Directeur de conférence (future)

Olivier Doutres
École de technologie supérieure (ÉTS)
conference@caa-aca.ca

Past Conference Chair
Directeur de conférence (passée)

Len Zedel
Memorial University of Newfoundland
zedel@mun.ca

Benjamin Zendel
bzendel@mun.ca

Past President
Président sortant

Frank A. Russo
Ryerson University
past-president@caa-aca.ca

Awards Coordinator
Coordinateur des prix

Victoria Duda
Université de Montréal
awards-coordinator@caa-aca.ca

System Administrator
Administrateur système

Cécile Le Cocq
ÉTS, Université du Québec
sysadmin@caa-aca.ca

Webmaster
Webmestre

Philip Tsui
RWDI
web@caa-aca.ca

Social Media Editor
Rédacteur média sociaux

Romain Dumoulin
Soft dB
media@caa-aca.ca

SUSTAINING SUBSCRIBERS - ABONNÉS DE SOUTIEN

The Canadian Acoustical Association gratefully acknowledges the financial assistance of the Sustaining Subscribers listed below. Their annual donations (of $475 or more) enable the journal to be distributed to all at a reasonable cost.
L'Association Canadienne d'Acoustique tient à témoigner sa reconnaissance à l'égard de ses Abonnés de Soutien en publiant ci-dessous leur nom et leur adresse. En amortissants les coûts de publication et de distribution, les dons annuels (de 475$ et plus) rendent le journal accessible à tous les membres.

Acoustec Inc.
Jean-Philippe Migneron - 418-496-6600
info@acoustec.qc.ca
acoustec.qc.ca

Acoustex Specialty Products
Mr. Brian Obratoski - 2893895564
Brian@acoustex.ca
www.acoustex.net

AcoustiGuard-Wilrep Ltd.
Mr. William T. Wilkinson - 888-625-8944
wtw@wilrep.com
acoustiguard.com

BKL Consultants Ltd.
Mark Bliss - 604-988-2508
bliss@bkl.ca
www.bkl.ca

dB Noise Reduction
Mehrzad Salkhordeh - 519-651-3330 x 220
mehrzad@dbnoisereduction.com
dbnoisereduction.com

HGC Engineering Ltd.
Mr. Bill Gastmeier -
bill@gastmeier.ca
hgcengineering.com

Hottinger Bruel & Kjaer inc.
Andrew Khoury - 514-695-8225
andrew.khoury@hbkworld.com
bksv.com

Jade Acoustics Inc.
Ms. Dalila Giusti - 905-660-2444
dalila@jadeacoustics.com
jadeacoustics.com

JASCO Applied Sciences (Canada) Ltd
Roberto Racca - +1.250.483.3300 ext.2001
roberto.racca@jasco.com
www.jasco.com

Pliteq Inc.
Wil Byrick - 416-449-0049
wbyrick@pliteq.com
www.pliteq.com