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# canadian acoustics
# acoustique canadienne

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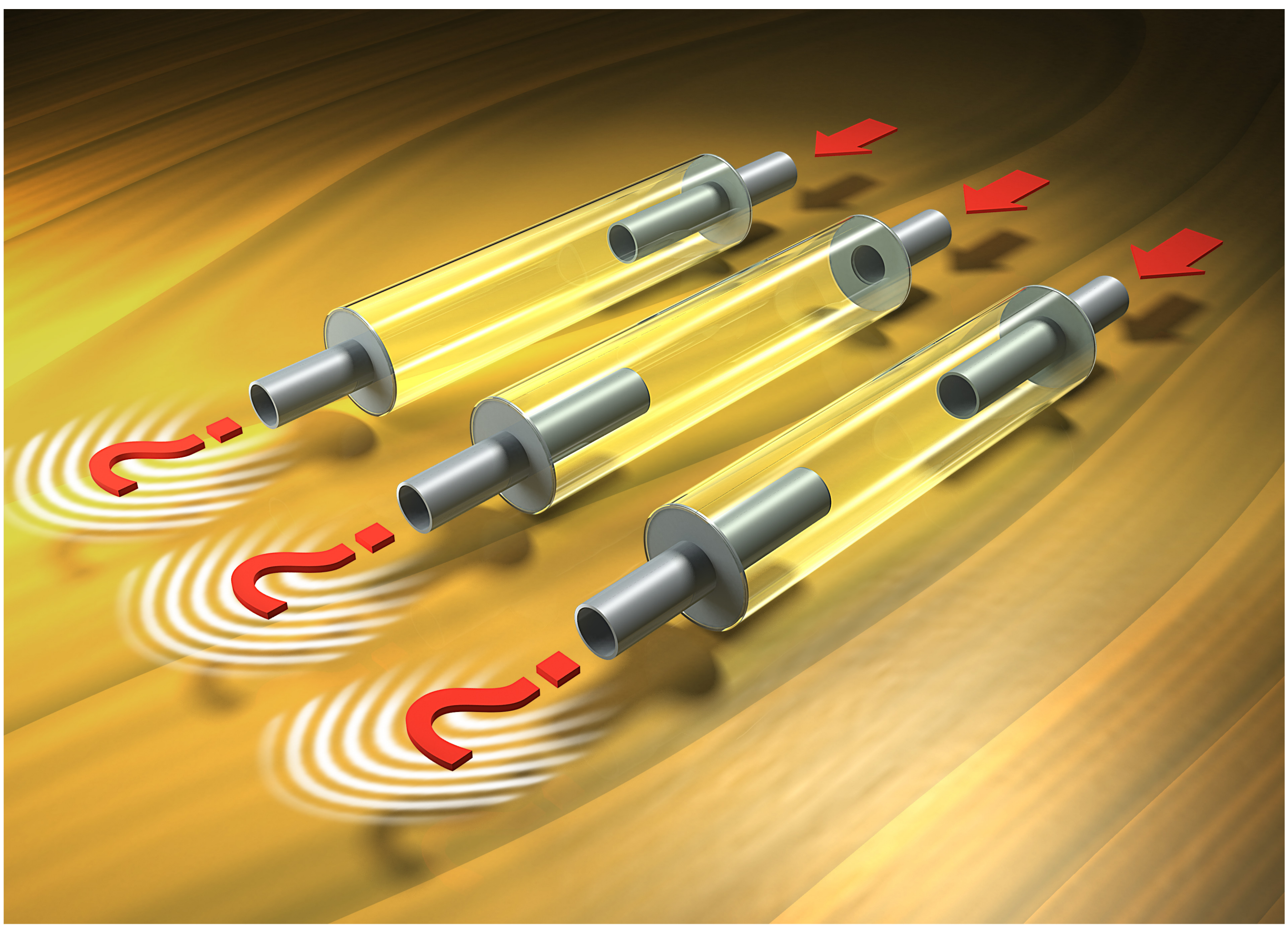

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Éditorial: Les changements et les défis

Editor's note: Changes and challenges

# Printemps 2017

C'est avec un grand plaisir et un honneur tout particulier que j'écris cet éditorial pour me présenter à vous dans mon nouveau rôle de rédacteur en chef.

Tout d'abord, je voudrai remercier les précédents rédacteurs en chef, Jérémie Voix et Ramani Ramakrishnan dont le soutien a permis une transition en douceur. Nous espérons que vous trouverez une croissance cohérente de notre revue dans les mois qui suivront.

L'Acoustique Canadienne a été la référence pour tous les acousticiens canadiens grâce à ses articles à comité de lecture et autres articles de presse traitant des divers aspects de l'acoustique et des vibrations. La revue contient également des informations à propos des récentes recherches, critiques, actualités, nouveaux produits, activités, discussions, etc… en lien avec l'acoustique et les vibrations. Toutes les publications rapportant des nouveaux résultats, des nouvelles applications, des critiques, des tutoriaux ou des notes de recherche, sont par conséquent les bienvenus car elles permettent de diversifier et d'approfondir nos contenus éditoriaux.

Alors que l'Acoustique Canadienne est en bonne forme et que plusieurs activités et projets existent, je souhaiterai vous faire part de ma vision à long terme. Ma mission, avec le soutien de tous les acousticiens canadiens, est

# Spring 2017

It is with great pleasure and honor that I am writing this editorial and presenting myself to you in my new role of Editor-in-Chief.

First of all, I would like to thank the previous Editors-in-Chief, Jeremie Vox and Ramani Ramakrishnan for all the support they have provided. Thanks to their help, this transition was smooth, and hopefully you will find a coherent growth of our journal in the next few months.

Canadian Acoustics has been the reference journal for all Canadian acousticians (and beyond) with refereed scientific articles and new items on all the aspects of acoustics and vibration. The journal also includes information on research, reviews, news, products, activities, discussions, and many more. Papers reporting new applications, as well as review and tutorial papers or shorter research notes are welcomed, and testify the diversity and depth of our contents.

While Canadian Acoustics is in good shape and several activities and plans exist, I would like to share with you my vision. With the support of all the Canadian acousticians, I hope to increase the visibility and diffusion of the journal, which will be still printed every three months.

One of my tasks as Editor-in-Chief will be to expand the Editorial Board, which will be enriched with new members to bring new energy for the several challenges we have. While I hope d'augmenter la visibilité et la diffusion du journal qui est imprimé trimestriellement.

L'une de mes premières tâches en tant que rédacteur en chef sera d'élargir le Comité de rédaction de la revue, qui sera enrichi de nouveaux membres pour apporter une nouvelle dynamique pour mieux aborder les nouveaux défis à venir. Mon principal défi est d'attirer autant de nouvelles publications et de nouveaux lecteurs que possible pour que la revue puisse obtenir un facteur d'impact permettant de consolider la reconnaissance de la revue à l'international. Nous souhaitons également poursuivre les discussions visant à établir le point d'équilibre entre les possibilités permettant de connecter les acousticiens canadiens et celles permettant d'offrir davantage d'espace aux annonces et autres actualités, tout en préservant la forte robustesse scientifique de la revue. En d'autres mots, notre but est d'enrichir le contenu de la revue.

Par conséquent, je travaille actuellement à la planification d'un numéro spécial en 2019 qui abordera des sujets émergents en lien avec l'acoustique et les vibrations, afin d'élargir la revue à de nouveaux lecteurs et auteurs.

L'acoustique est une question vaste qui emploie des centaines de spécialistes à travers le Canada dans divers domaines tels que l'enseignement, la recherche, la consultation, l'industrie, etc... C'est pour refléter cette diversité, et pour permettre à tous de découvrir de nouveaux professionnels, que l'Acoustique Canadienne a commencé les séries de numéros spéciaux dits «régionaux» afin de donner l'occasion aux particuliers, aux groupes et aux entreprises situés dans les plus grandes zones des grandes villes canadiennes de porter à la connaissance de tous les cas rencontrés dans leurs domaines de spécialisation.

Selon le nombre d'articles soumis par les équipes et les professionnels travaillant en recherche et dans l'industrie dans les numéros spéciaux dédiés à Montréal (en 2015), à la région du Grand Toronto (en 2016) et à Halifax (cette année), il est évident que nous sommes une grande communauté. Même si je dois avouer que j'espérai une plus grande participation et un plus

that the journal will obtain an impact factor to receive more international recognition, my main mission is to attract as many readers and papers as possible. We will keep alive the discussion about the equilibrium point between the strong scientific robustness of the journal and its capability to connect all the Canadian acousticians and to offer a space to host announcements and news. In other words, we want to increase the content of our journal.

For this scope, I am working to plan some special issues in 2019 on emerging topics, in order to attract new researchers and practitioners.

Acoustics is a broad subject matter that employs hundreds of specialists across the country in diverse fields such as teaching, research, consulting, industry, and others. To reflect such a diversity and to help each of us to discover new professionals, the Canadian Acoustics started the series of special "regional" issues to offer an opportunity to individuals, groups and companies located within the greater areas of major cities in Canada to show case their specialty.

Based on the number of articles submitted by teams and individuals from the research and industrial sectors for the special issues in Montreal (in 2015), in the GTA (in 2016), and the in Halifax (this year), there is no doubt that we are a large community. I should confess that I hoped a larger participation and number of submissions for the present special issue dedicated to Halifax. In any case, it is already time for us to work for the regional issue for 2018 which will be dedicated to Vancouver (and the province of British Columbia).

I am also happy to share with you a special announcement on Local Chapters of the CAA that our president Frank Russo has sent to me: "*In an effort to support the flourishing of the CAA, the board has recently decided to support the development of local chapters of the CAA. A small seed grant will be made available to any member in good standing who has a plan to develop a new local chapter. If you are interested, please contact the President of the CAA with a brief 1-page proposal.*"

grand nombre de soumissions pour le numéro spécial consacré à Halifax, il est déjà temps pour nous de travailler sur le numéro spécial de 2018 qui sera consacré à Vancouver et à la province de la Colombie Britannique.

Je suis également heureux de partager avec vous la déclaration prononcée par notre président Frank Russo, à propos des sections locales de la ACA : "*Dans un effort pour soutenir l'épanouissement de l'ACA, le conseil d'administration a récemment décidé de soutenir le développement des chapitres locaux rattachés à l'ACA. Une petite subvention d'implantation sera disponible à tout membre en règle qui souhaiterait développer un nouveau chapitre local. Si vous êtes intéressé, merci de faire parvenir une courte proposition de projet (1 page) au président de l'ACA"*.

Avant de terminer cet éditorial, je tiens à remercier toutes les personnes présentes et le nouveau comité de rédaction en activité depuis mai 2017 : M. Romain Dumoulin, notre nouveau rédacteur en chef adjoint, le Dr. Cécile Le Cocq, notre nouvelle directrice de publication et le Dr. Olivier Valentin, notre nouveau relecteur-réviseur.

Je vous souhaite, à toutes et à tous, une bonne lecture et un été agréable !

Umberto Berardi
Rédacteur en chef.

Before closing this editorial, I would like to present and thank all the new editorial board members. Starting since May 2017, Mr. Romain Dumoulin will be our new Deputy Editor, Dr. Cécile Le Cocq will be our new Journal Manager, and Dr. Olivier Valentin will be our new Copyeditor.

I wish you a pleasant reading, and a pleasant summer

Umberto Berardi
Editor-in-chief


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# SOCIOPHONETIC RESEARCH AT THE UNIVERSITY OF NEW BRUNSWICK

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## Résumé

Dans notre laboratoire, nos intérêts de recherche se trouvent au carrefour de la sociolinguistique et de la phonétique. Nous utilisons les techniques de la phonétique acoustique dans l'analyse de la variation régionale, sociale et stylistique dans les variétés de français acadien parlées au Nouveau-Brunswick. Parmi les traits segmentaux analysés figurent la consonne /r/ et l'affrication de /t, d/ ; l'analyse des traits prosodiques comprend entre autres l'application des métriques rythmiques aux unités prosodiques telles que la phrase accentuelle.

**Mots-clés :** changement et variation linguistique, français acadien, phonétique acoustique

## Abstract

Research in our laboratory is situated at the interface of sociolinguistics and phonetics. Acoustic phonetic techniques are used to analyze regional, social and stylistic variation in the varieties of Acadian French spoken in New Brunswick. Studies analyze segmental features such as the consonant /r/ and the affrication of /t, d/; the study of prosodic features includes the application of rhythm metrics to prosodic units such as the accentual phrase.

**Keywords:** Acadian French, acoustic phonetics, language variation and change

## 1 Introduction

The field of sociophonetics lies at the interface of sociolinguistics and phonetics. Broadly speaking, researchers in this area study variation in the sound system of one or more languages with a view to understand how social factors – such as age, gender, social class, social networks – structure this variation. The research spans both speech production and perception. A central theme is the study of the origin and spread of change in sound systems. Acoustics provides several instrumental techniques that are used for this study. Indeed, the fine-grained details revealed by acoustic techniques have contributed significantly to the development of this field.

## 2 Laboratory Setting

Work in our laboratory at the University of New Brunswick in Fredericton, NB, focuses on variation and change in Acadian French. Spoken predominantly in Canada's Atlantic region, this dialect is distinct from Laurentian French, which is found in Québec, Ontario and provinces and territories to the west and north. Our interest is in regional (geographic), social and stylistic variation in this dialect.

We currently work with two databases. The RACAD (*Reconnaissance automatique de l'acadien*) corpus is a recorded set of readings by 140 speakers from the five main francophone regions of New Brunswick [1]. Based on the TIMIT corpus of American English, the RACAD corpus was designed for the development of automatic speech recognition systems. Of special interest to linguists are the two "calibration sentences", which include words that are expected to show significant regional differences in pronunciation. Three extralinguistic sources of variation can be studied: region, age and gender.

The other database that we work with is a set of sociolinguistic interviews conducted according to the protocol of the PFC (*Phonologie du français contemporain*) project (www.projet-pfc.net). Participants in this database are twelve native speakers of Acadian French from Tracadie, a small town in northeastern New Brunswick [2]. The three main extralinguistic factors of interest are age, gender and style (including reading, spontaneous conversation and semi-directed interview styles). It is worth noting that this project has an international context; PFC is a collaborative research program that brings together researchers from many countries who analyze French spoken in different parts of the French-speaking world and who often use these data to inform phonological theory.

Acoustic analyses – which include spectrographic and oscillographic analyses, formant analysis, F0 tracking, duration and intensity measurement – are carried out with *Praat* software (www.fon.hum.uva.nl/praat/).

The students who participate in our research projects are usually undergraduates. They study acoustic phonetics in an introductory level course in phonetics, offered in third year, and in specialized seminars, offered in fourth year.

* cicho@unb.ca

## 3 Recent Research

Our research focuses on identifying sources of systematic variation in speech production at both segmental (that is, vocalic and consonantal) and prosodic (or suprasegmental) levels.

### 3.1 Segmental variation

<u>The /r/ consonant</u> Spectrographic analyses of RACAD data show that this consonant is pronounced both as an alveolar trill, [r], (in the front of the mouth) and as a velar or uvular fricative or trill, [R], (in the back of the mouth). Effects of age on this variation suggest that there is an ongoing change from [r] to [R]; that is, younger speakers use the (back) [R] variant more often than older speakers. Furthermore, northern regions have a greater frequency of the innovative variant [3]. An interesting finding is that this change appears to be limited to syllable-onset positions in a word, as in [frã.sɛ] "F<u>r</u>ançais" and [ka.ra.kɛt] "Ca<u>r</u>aquet".

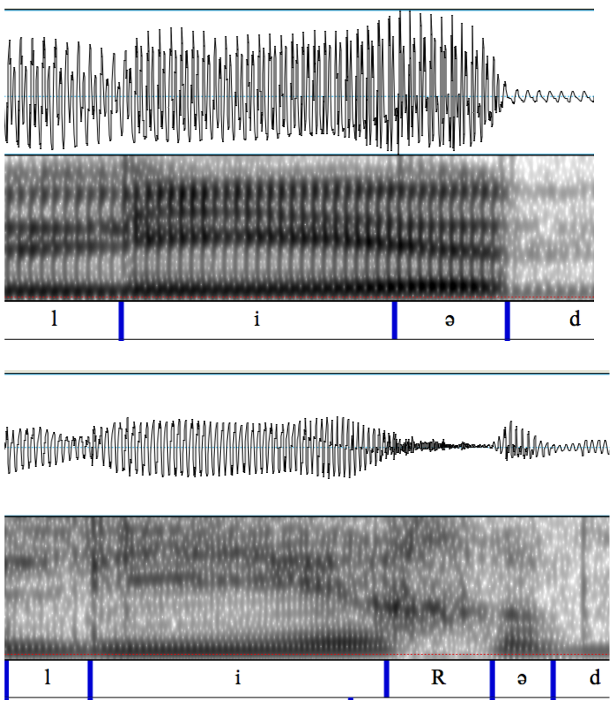


**Figure 1:** Oscillographic traces and spectrograms of "lire d(ans)". Upper figure show an off-glide pronunciation of the /r/ consonant; that is, "lire" is pronounced [liə] (spoken by a 44-year-old male from Moncton/Dieppe in southern New Brunswick). Lower figure shows a fricative pronunciation of /r/ in the back of the mouth; that is, "lire" is pronounced [liRə] (spoken by a 20-year-old female from Edmundston in northern New Brunswick).

In postvocalic syllable-final position, the articulatory realization of the /r/ consonant seldom resembles a consonantal constriction such as [r] or [R]. Instead, speakers produce vowel-like pronunciations, as in [liə, liɐ] "lire", a process called r-vocalization. See Figure 1 for a comparison of these vocalic and consonantal pronunciations. To determine the precise nature of the vocalic pronunciations (called off-glides), we are examining vowel formant trajectories from the start of the vowel to the end of the word. Dr Paul De Decker, a visiting scholar at our laboratory from the Memorial University of Newfoundland, is collaborating on this research question.

<u>Affrication of /t, d/</u> The affrication of the /t/ and /d/ consonants before certain vowels and glides is one of the most distinguishing features of Laurentian French. These affricate consonants are generally alveolar [ts, dz], as in [pə.ʦit] "petite" and [a.ka.ʣi] "Acadie". Until recently, this affrication was absent from most varieties of Acadian French; speakers said [pə.tit] and [a.ka.di], with a [t, d] stop. Figure 2 illustrates both the [ts] affricate and the [t] stop pronunciations of the first /t/ in "petite".

Based on our work with the RACAD data, age effects show that younger speakers have greater frequencies of the affrication feature that do older speakers, leading to the inference that this is an ongoing sound change [4]. Region effects suggest that speakers from northern New Brunswick are the leaders in this change.

However, in certain words the place of articulation of the affricate is sometimes palatal and not alveolar, as in [vã.ʤy] "vendu" instead of [vã.ʣy]. Because it is not always possible to hear the difference between these two pronunciations, we use acoustic analyses – spectral cuts and measures of centre of gravity – of the fricative section of the affricate to determine which variant a speaker is producing. The palatal pronunciation is an innovation that is found mainly in southern regions of New Brunswick.

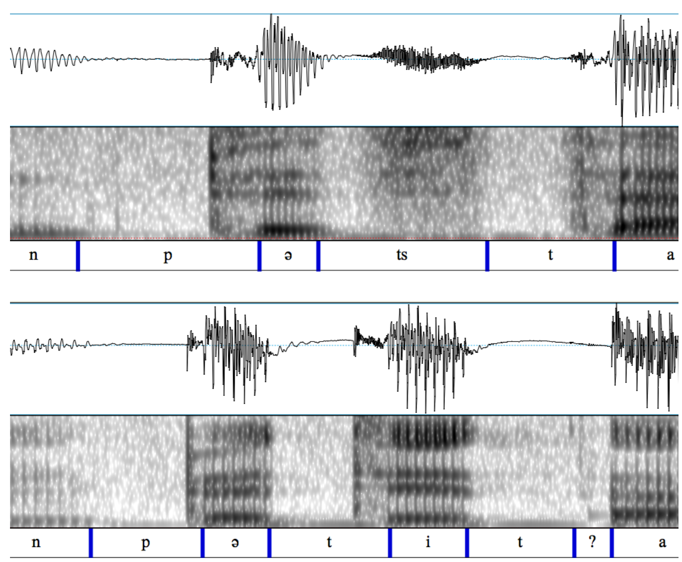


**Figure 2:** Oscillographic traces and spectrograms of "(une) petite a(gence)". Upper figure shows a [ts] affricate pronunciation of the first /t/ in "petite". Note that the speaker has deleted the /i/ vowel that follows this /t/; that is, "petite" is pronounced [pətst] (spoken by a 24-year-old female from Shippagan in northern New Brunswick). Lower figure shows a [t] stop pronunciation of the /t/ consonant; that is, "petite" is pronounced [pətit] (spoken by a 25-year-old male from Paquetville in northern New Brunswick).

### 3.2 Prosodic variation

Timing of segmental intervals  Our recent research on prosodic features has been looking at rhythm and, more specifically, at speech timing. We apply rhythm metrics to measure variability in the durations of vocalic and consonantal intervals. Results for the RACAD data show that there are significant timing differences between northern and southern varieties in New Brunswick: northern varieties have greater variability in both vocalic and consonantal interval durations [5]. Also, the %V metric, which measures the percentage of the speech in an utterance that is vocalic, shows higher values among southern speakers than among those from northern regions. Ongoing research is examining how processes such as r-vocalization and the affrication of /t, d/ contribute to this durational variability.

Timing of accentual phrases  We have also applied rhythm metrics to study the timing of prosodic units that are larger than segments, such as syllables and accentual phrases. The accentual phrase, also called a rhythmic group, is a sequence of syllables that is demarcated by a stress. Analyses of the PFC-Tracadie data show important stylistic differences in the timing of these units: spontaneous speech style has greater variability in accentual phrase duration than reading style [6]. In our current research we are examining the makeup of the accentual phrase [7]. Figure 3 illustrates the segmentation of part of one of the sentences in the RACAD corpus into segments, syllables and accentual phrases.

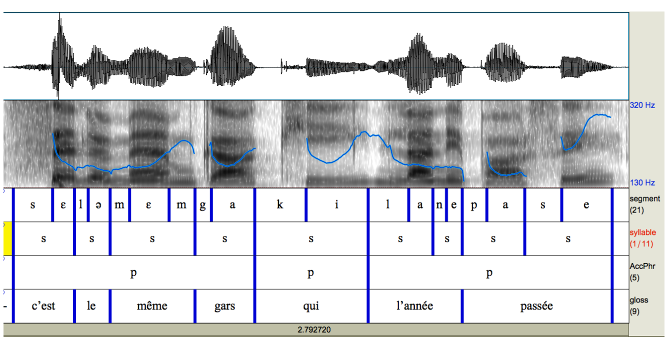


**Figure 3:** Oscillographic trace, spectrogram and F0 curve of "C'est le même gars qui l'année passée ...". The four tiers below the acoustic analyses indicate: segments, syllables, accentual phrases and word glosses. (spoken by a 55-year-old female from Moncton/Dieppe in southern New Brunswick).

## 4 Other Applications

Our initial research on rhythm metrics was developed with Dr Sid-Ahmed Selouani and colleagues at the Shippagan, NB, campus of the Université de Moncton. In the context of this collaboration, we applied rhythm metrics to automatic speech recognition systems that are being developed to distinguish between groups of speakers. This work has focused primarily on the speech of native and non-native speakers of Arabic [8].

Finally, many of our students speak French as a second or third language, and some of them are interested in analyzing their own pronunciation in French. As a result, in their class research projects these students choose to carry out acoustic analyses that compare their pronunciations with those of the native French speakers in our databases. In this sense, work with acoustic phonetic techniques and with large speech databases serves as a language-learning tool.

## References

[1] Cichocki, W., Selouani, S.-A. & Beaulieu, L. (2008). The RACAD speech corpus of New Brunswick Acadian French: Design and applications. *Canadian Acoustics* 36: 3-10.

[2] Cichocki, W. (2014). The PFC-Tracadie Corpus: A variety of Acadian French spoken in northeastern New Brunswick (Canada). In Barysevich, A., D'Arcy, A. & Heap, D. (eds) *Proceedings of Methods XIV. Papers from the 14th International Conference on Methods in Dialectology*, Bern, Peter Lang, 149-157.

[3] Cichocki, W. & Perreault, Y. (to appear). Contribution à l'étude phonétique et géolinguistique du R en français parlé au Nouveau-Brunswick. In Hallion, S. & Rosen, N. (eds) *Actes du colloque «Les français d'ici»*. Québec: Presses de l'Université Laval.

[4] Cichocki, W. & Perreault, Y. (to appear). L'assibilation des occlusives /t/ et /d/ en français parlé au Nouveau-Brunswick : nouveau regard sur la question. In Arrighi, L. & Gauvin, K. (eds) *Les français d'ici : Regards croisés*. Québec: Presses de l'Université Laval.

[5] Cichocki, W., Selouani, S-A., Ayed, A.B., Paulin, C. & Perreault, Y. (2013). Variation of rhythm metrics in regional varieties of Acadian French. *Proceedings of Meetings on Acoustics* 19, 035082. [DOI: 10.1121/1.4800686].

[6] Cichocki, W. (2015). The timing of Accentual Phrases in read and spontaneous speech: Data from Acadian French. *Canadian Acoustics* 43(3): 109-110.

[7] Cichocki, W. (to appear). Profil temporel du groupe rythmique en français acadien : variabilité stylistique. *Repères DoRiF*.

[8] Selouani, S-A., Alotaibi, Y., Cichocki, W., Gharssallaoui, S. & Kadi, K. (2015). Native and non-native class discrimination using speech rhythm- and auditory-based cues. *Computer Speech and Language* 31: 28-48.

# Assessing And Mitigating The Risk of Multibeam Echosounder Use Near Endangered Beaked Whales In The Gully Marine Protected Area

**Paul Macnab [*1], Shannan Murphy [†1] and Alexandre Normandeau [‡2]**
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## Résumé

Un navire de recherche équipé d'un échosondeur multifaisceaux a permis au gouvernement du Canada de collecter des données bathymétriques et géologiques superficielles dans la zone de protection marine (ZPM) du Gully où se trouve un canyon incisant la marge continentale. Cette opportunité a nécessité des mesures spécifiques pour protéger l'écosystème local, notamment le benthos sensible et les baleines à bec commune menacées d'extinction. Plusieurs mesures réglementaires ont été déclenchées par une proposition d'avril 2017 visant à cartographier à nouveau le canyon. Dans cette étude, nous examinons le projet et les décisions communes prises par le promoteur et les autorités fédérales, notre objectif principal étant d'informer sur l'évolution du dialogue interne sur les impacts du bruit anthropique.

**Mots clefs:** échosondeurs multifaisceaux, ZPM du Gully, LEP, Baleines à bec commune, évaluation environnementale

## Abstract

A charter with multibeam sounders presented the Government of Canada an opportunity to collect bathymetric and surficial geology data in the Gully Marine Protected Area (MPA), locus of a shelf-incising canyon granted broad ecosystem protection with specific measures enacted for sensitive benthos and endangered northern bottlenose whales. Several regulatory processes were triggered by an April 2017 proposal to remap the canyon. Here we review the project and joint decisions reached by the proponent and federal authorities, our primary aim to inform the evolving domestic dialogue on anthropogenic noise impacts.

**Keywords:** multibeam echosounders, Gully MPA, SARA, northern bottlenose whales, environmental assessment

## 1 Introduction

The Gully Marine Protected Area (MPA) lays offshore Nova Scotia at 44°N - 59°W, east of Sable Island National Park Reserve. Declaration in 2004 made the Gully Canada's first *Oceans Act* MPA in the Atlantic. Science conducted before and after designation highlights a wide range of habitats and species. Diverse cold water corals and abundant cetaceans remain focal points for research and management [1]. Numerous fish, turtles and mammals listed under the *Species at Risk Act* (SARA) are found in the Gully; most notably, northern bottlenose whales (NBW) belonging to the endangered Scotian Shelf population (n=143). The central canyon is afforded a pair of strict legal protections: the MPA Regulations foreclose commercial extraction at surface, midwater and seabed depths >600m; and SARA prohibits the destruction of NBW Critical Habitat (CH) comprising Zone 1 of the MPA canyon core [2].

The Gully has attracted waves of research since the 1960s, a decade that saw the last of Canada's commercial whale hunt and the dawn of federal offshore geoscience. Contemporary investigations have been broad and multi-disciplinary, covering many scientific fields since the 1990s.

Seabed features, benthic communities and near-bottom processes have been focal [3] as have cetaceans [4]. The MPA co-evolved with intensifying study during a period that saw growing domestic interest in underwater noise as a ubiquitous stressor and jurisprudence in relation to marine geological research [e.g., 5, 6]. Meanwhile, increasing biological consensus that beaked whales are particularly susceptible to noise disturbance made it inevitable that Gully science using sound for mapping, oceanography and fisheries research would come under scrutiny and trigger regulatory reviews.

## 2 Multibeam-equipped vessel of opportunity

Refit of the CCGS *Hudson* prompted the Bedford Institute of Oceanography to issue a request for vessel services in April 2017. RV *Coriolis II* based in Quebec was awarded the charter. An interdepartmental proposal was assembled to advantage the ship's hull-mounted multibeam echosounders (MBES) coincident with semi-annual Gully oceanography stations of the Atlantic Zone Monitoring Program. The aim was to map depths and seabed properties using standard methods [see 3] along a 59 nautical mile transect starting shallow on Sable Island Bank and progressing downslope in 2 segments: the first running 28 nm along a feeder channel and down the main axis to an upper-canyon sampling station; the second tracing the thalweg for ~30 nm before

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terminating at the canyon mouth. MBES would ensonify the MPA for about 8 hours during total RV occupancy <1 day.

## 3 Environmental risk assessment

Three interrelated permitting processes were triggered by the proposal: activity plan approvals required by the MPA Regulations; adverse impacts and allowable harm permit considerations pertaining to SARA; and a determination under the serious harm provisions of the *Fisheries Act*. All had in common a single assessment of risks posed by the MBES. Reference materials were drawn from DFO advisories; reviews of impacts and global thresholds; manufacturer specifications and documentation; and prior Gully clearances issued in 2006 for a 12 kHz MBES survey.

RV Coriolis II carries 2 Kongsberg MBES: an EM 2040 acquires at user-selected frequencies between 200-400 kHz to 600m; an EM 302 utilizes 30 kHz signals capable of sounding to several kms. The EM 2040 emits at 218 dB re 1 μPa @ 1m producing a 140° fan of 0.5°x1° beams. The nominal EM 302 source level (SL) is 237 dB re 1 μPa @ 1m with a transmission pattern composed of 1°x1° beamforms. Simultaneous operation was proposed to depth maxima for the EM 2040. Serious effects on fish and turtles were not anticipated. While baleen whales use the Gully, including endangered North Atlantic right and blue whales, MBES were not expected to mask vocals and assumed hearing spectra. Toothed whales vocalizing and hearing at higher frequencies were expected to discern and possibly react to MBES. Beaked whale risk factors were assessed to be of greatest concern given direct EM 302 frequency overlaps.

Prudent best-practice mitigation measures were adopted to minimize risk in the MPA: power ramp-up; daytime operations; wildlife observation; and a shallow-to-deep sail path offering theoretical egress to the open ocean via the canyon mouth. Two measures were central to decisions that allowed the survey to proceed: SL reductions and CH avoidance. A reportedly new Kongsberg mammal protection function was enabled to attenuate SLs by the -20 dB maximum thereby achieving a hypothetical 180 dB safety threshold for cetacean injury or harm at 20m below the hull. The survey line was truncated at the upper-canyon sampling station as per recommendations made and agreed to at a meeting of the Gully MPA Advisory Committee, a multistakeholder group that also provides input to DFO on NBW recovery. Segment 2 was less essential to specific project goals (e.g., off-shelf transport), so restricting MBES from deep parts of the MPA avoided potential CH stressors.

## 4 Discussion

Statutory obligations and implications for marine scientific research are in a state of evolution domestically and internationally. The present case study illustrates how scientific investigators using sound energy in MPAs known to support acoustically sensitive species are becoming increasingly subject to the same environmental assessment and permit requirements imposed on commercial ventures. In this instance, the research goal—high quality bathymetry and interpretable surficial geology in relation to bedforms and sediment transport dynamics—was partially met in the upper reaches of the Gully. Not surveying the canyon core leaves knowledge gaps that may need to be addressed with further surveys, especially if MBES data analyses reveal active transport of sediment from the sandy bank tops to the canyon deeps. The crucial role of seabed maps in benthic science and conservation planning is largely self-evident; less obvious is how MBES might contribute to endangered cetaceans. One illustration: contaminants detected in tissue samples [7] have been speculatively tied to pollution arriving via the off-shelf vectors under study here.

With its linked science and conservation histories, the Gully MPA offers a testbed for policy development and regulatory application without, it is hoped, incurring legal challenges. The Gully and similarly protected habitats present opportunities for collaborative fact-finding and interdisciplinary research. Acousticians and corresponding engineering capacities in Halifax and elsewhere in Canada appear ready to tackle many remaining challenges; e.g., sound loss models examined for this project [8] could be revisited for MPAs and domestic research platforms, with predictions made for -10/-20 dB SL attenuations where available. Purposeful MBES signal processing to actively detect vocalizing whales or those swimming close enough to be imaged is another frontier. Considerable potential also exists for designing and implementing coupled aural-visual studies of cetacean behavioural response in the presence of overt noise commanded and controlled by qualified personnel, not least in the Gully where cetaceans have been studied intensively since 1988 and autonomous acoustic recorders deployed nearly continuously since 2003.

## Acknowledgments

Andrew Cogswell lent project support as did Centre Interdisciplinaire de Développement en Cartographie des Océans. Gully Advisory Committee members provided timely input. Perspectives and commentary reflect views of the authors and not necessarily those of their employers.

## References

[1] DFO. The Gully MPA Management Plan. 2008.

[2] DFO. Recovery Strategy for the Northern Bottlenose Whale (*Hyperoodon ampullatus*), Scotian Shelf population. 2016.

[3] R.A. Pickrill and B.J. Todd. The multiple roles of acoustic mapping in integrated ocean management, Canadian Atlantic continental margin. *Ocean Coast. Manag.*, 46: 601-614, 2003.

[4] H.B. Moors-Murphy. Submarine canyons as important habitat for cetaceans, with special reference to the Gully: A review. *Deep-Sea Research II*, 104: 6-19, 2015.

[5] Living Oceans Society v. Canada (Minister of Fisheries and Oceans), 2009 FC 848.

[6] Qikiqtani Inuit Association v. Canada (Minister of Natural Resources), 2010 NUCJ 12.

[7] DFO. Contaminant monitoring in the Gully Marine Protected Area. *Can. Sci. Advis. Sec. Sci. Advis. Rep*. 2009/002.

[8] X. Lurton. Modelling the sound field radiated by multibeam echosounders for acoustical impact assessment. *Applied Acoustics* 101: 201-221, 2016.

# DEMONSTRATING THE FEASIBILITY OF NEAR-REAL-TIME VESSEL NOISE MAPPING TO MANAGE MARINE MAMMAL NOISE IMPACTS

**Loren Horwich[1], Jeremy Prowse[1], Alex MacGillivray[2], Bruce Martin[1,3], Sue Molloy[4], Wayne Renaud[5], David Hannay[2]**
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## Résumé

Le bruit produit par l'homme dans les océans peut provoquer des séquelles physiques et des perturbations comportementales chez les créatures marines. Chez les mammifères marins, il gêne leurs utilisations des sons pour la recherche de nourriture, la communication, la navigation, la socialisation et la reproduction. Les progrès dans les enregistreurs acoustiques, les observatoires océaniques, le suivi des navires, et la modélisation du bruit nous permettent d'étudier et contrôler les effets du bruit généré par le trafic maritime sur la vie marine. Cet article traite d'une étude pour l'agence spatiale canadienne visant à examiner la faisabilité d'une interface web contrôlée par l'utilisateur qui fournit une prédiction en temps quasi-réel du bruit du trafic maritime dans les habitats de vie marine. '*ShipNoiseView*' intègre la position en direct du navire depuis le système d'identification automatique par satellite (AIS) avec la télédétection en temps réel des données océanographiques et les modèles validés de propagation du bruit de navire. Grâce à cet outil, il est possible d'estimer les niveaux sonores cumulatifs du navire et de contrôler l'effet du bruit sur la vie marine grâce au suivi et l'atténuation en temps réel.

**Mots clés:** bruit océanique anthropique, mammifères marins, modélisation, navigation

## Abstract

Man-made ocean noise can cause physical injury and behavioral disturbance to marine life. It hampers marine mammals' use of sound for foraging, communicating, navigating, socializing, and mating. Advancements in acoustic recorders, ocean observatories, vessel tracking, and noise modelling allow us to study and manage the effects of vessel noise on marine life. This paper discusses a study for the Canadian Space Agency to investigate the feasibility of a user-controlled web interface that provides near-real-time prediction of vessel noise in marine life habitats. '*ShipNoiseView*' integrates live vessel position data from the Satellite-Automatic Identification System (S-AIS) with real-time remote sensing of oceanographic data and verified vessel noise propagation models to assess cumulative vessel sound levels and to manage the effect of noise on marine life through real-time monitoring and mitigation.

**Keywords:** anthropogenic ocean noise, marine mammals, modelling, shipping

## 1 Introduction

Vessel traffic is a leading cause of ocean noise and increases stress hormones in marine mammals.[1] The ocean ecosystem is critical to the health of our planet and the future of our fisheries, and marine mammals hold important balancing roles in maintaining its stability. Predicting and measuring noise fields from vessels offers a chance to manage noise emissions using the International Maritime Organization's system of speed restrictions, areas-to-be-avoided, Notices to Mariners, and vessel traffic lanes. [2]

JASCO Applied Sciences completed a study for the Canadian Space Agency investigating the feasibility of developing a vessel traffic control system based on real-time prediction of vessel noise to evaluate the cumulative effects of shipping in a region and to propose mitigation actions to reduce the noise and impacts to marine life. A prototype tool was developed and implemented for a case study in the Saguenay-St. Lawrence Marine Park, QC, which is a region of interest due to the proximity of shipping lanes to protected critical beluga habitat.

## 2 Method

### Prototype Tool: ShipNoiseView

*ShipNoiseView* is a web-based map interface for users to view the sound contributions from one or more transiting vessels and explore alternate transit scenarios to mitigate noise impacts. It uses modern web services and map displays to integrate S-AIS tracks, remote sensing data, and an acoustic propagation modeling expert system. The model, which requires no manual data handling to run, is hosted on a remote server and deployed in real time via web-based user input.

Using the modelled acoustic field results, a user can assess the impact to marine life from a vessel transit (Figure 1). If sound levels exceed recommended thresholds, the watch officer may explore noise-reducing options for the vessel track, such as reducing speed, changing the route, grouping vessels together or changing propulsion systems for hybrid vessels.

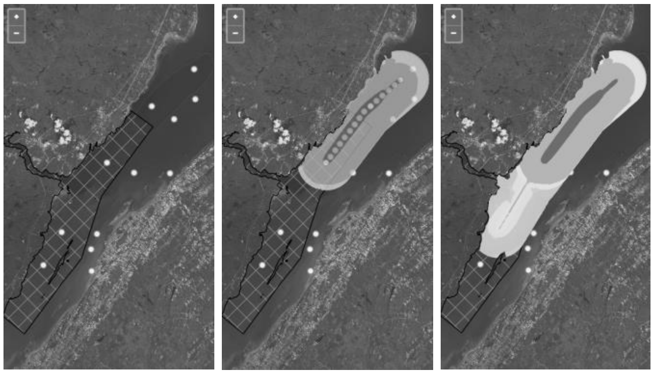

**Figure 1:** *ShipNoiseView* mitigation scenario: Current vessel locations (left), sound field for selected vessel track (middle), and reduced sound field with mitigation by speed reduction (right).

**Acoustic Model**

The vessel noise footprints are predicted using JASCO's Cumulative Vessel Noise Model, which was developed to calculate regional sound fields from multiple vessel tracks using precomputed propagation loss (PL) curves unique to a regional environment. The PL curves are modelled using JASCO's Marine Operations Noise Model (MONM), an in-house parabolic equation based propagation model based on the USN Range-dependent Acoustic Model (RAM). The curves define how sound attenuates as it travels through water and sediments, independent of the sound source. The study area is divided into 14 acoustic 'zones,' characterized by their depth and bottom type. Each zone's PL curves are calculated for a variety of propagation scenarios that depend on range, deci-decade frequency band, source and receiver depths, and the sound speed profile. Modelling the PL curves is the most computationally heavy step in calculating the sound field, so precomputing and storing them as easily accessible tables allows the Cumulative Vessel Noise Model to generate the user-requested sound fields in near-real-time by selecting and combining the appropriate PL curves for each vessel track scenario.

The Cumulative Vessel Noise Model divides the vessel track data into a series of time steps and calculates the regional sound field for each using a gridded representation of the region. A 100x100 m grid is overlaid on the acoustic propagation zones, and each vessel source is centred in the appropriate grid cell. The vessel's source signature is applied and propagated into adjacent grid cells using a weighted average of PL curves for the acoustic zones crossed along its path. The radial sound field for each source is summed across all grid cells for each time step, resulting in an instantaneous regional sound field for the time snapshot with contributions from all vessel tracks submitted to the model (Figure 2). The regional fields are then summed temporally to get the cumulative sound field for the full duration of the vessel tracks.

# 3 Discussion

## 3.1 Continuing Development

In addition to the run-time benefits, precomputing PL curves by fully characterizing the environment in advance makes the Cumulative Vessel Noise Model extremely versatile in its ability to accept input data from a variety of sources. The model currently uses static and historical average environmental data. It could easily include in-situ measurements or real-time remote sensing to more accurately select environmental propagation conditions. Its resolution can be easily improved by dividing the study area into finer depth and geo-acoustic zones.

Similarly, the prototype *ShipNoiseView* assigns frequency-dependent vessel source signatures from a list of published modelled and measured surrogate spectra based on the vessel type, size, and speed. Acoustic recorders could be employed to provide vessel-specific measured source levels, as well as data for marine mammal detections, ambient noise, and model verification.

Vessel noise is persistent and regional, and its affect on mammals in different hearing groups is based on the frequency of the noise being produced. It therefore must be assessed on a cumulative scale considering the auditory characteristics of specific mammal types in the region. The acoustic model can provide marine mammal audio-weighted cumulative acoustic fields and ranges to injury and disturbance thresholds based on published impact criteria. In addition to mitigating noise from targeted high-risk vessel transits, assessing cumulative noise emissions over long timescales highlights variations caused by vessel traffic patterns or seasonal environmental changes.

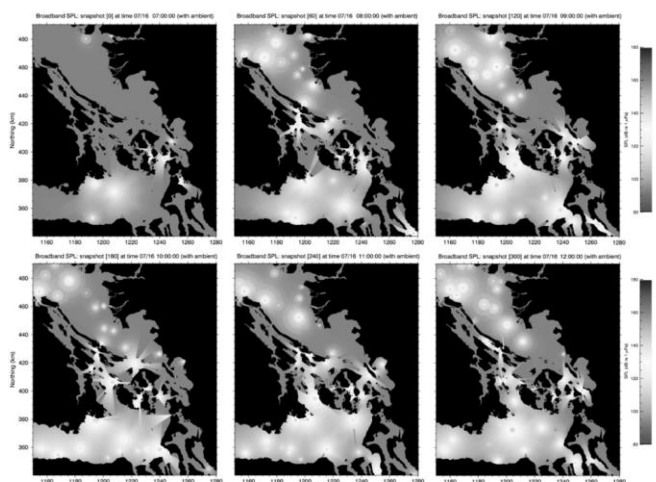

**Figure 2:** Cumulative Vessel Noise Model time 'snapshot' acoustic fields in the Salish Sea, BC. Time step is 1 hour between successive snapshots. [3]

## 3.2 Implementation and Integration

The Canadian Coast Guard's Marine Communication and Traffic Service (MCTS) monitor thousands of vessels transiting Canadian waterways daily. *ShipNoiseView* has been proven to be feasible as a stand-alone system, but could also be used in an existing marine traffic communication network capable of sending real-time navigation aids for mitigating ocean noise impacts. Using 96-hour Pre-Arrival Information Reports provided to Transport Canada along with the S-AIS position reports, the model could run automatically in the background as soon as vessels are identified as proceeding to Canadian waters and ports, with alerts sent to MCTS operators when the model calculates that specific sound level thresholds will be exceeded.

Comprehensive ocean noise management will require ongoing research and centralized data from many scientific efforts, including real-time marine mammal monitoring, detailed vessel traffic reporting, reliable vessel source sound level measurements, live remote sensing of environmental data, and acoustic monitoring for model verification and regional ambient noise conditions. Together with advancements in data acquisition and availability, as well as state of the art technologies for communication and management, an innovative real-time noise prediction tool such as S*hipNoiseView* could play a key role in transforming data from scientific and government sectors into operational decision aids for mitigating the impacts of ocean noise on marine life.

## Acknowledgments

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## References

[1] Rolland, R.M., S.E. Parks, K.E. Hunt, M. Castellote, P.J. Corkeron, D.P. Nowacek, S.K. Wasser, and S.D. Kraus. 2012. Evidence that ship noise increases stress in right whales. *Proceedings of the Royal Society B: Biological Sciences*.

[2] United Nations. 2017. *International Maritime Organization Profile: Overview* (webpage). Copyright UN-Business Action Hub and the UN.

[3] MacGillivray, A., Z. Li, G. Warner, and C. O'Neill. 2014. *APPENDIX 9.8-B Regional Commercial Vessel Traffic Underwater Noise Exposure Study*. Technical report prepared by JASCO Applied Sciences for Port Metro Vancouver. 80 pp.

# Air Dome Acoustics

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## Résumé

Le dôme pneumatique est une structure gonflable composée d'une ou plusieurs couches de membrane continue et flexible ancrée dans le sol qui est gonflée et soutenue grâce à de l'air comprimé. Dans cet article, nous abordons les résultats de mesurages acoustiques effectués à l'intérieur d'un dôme pneumatique. Dans ce type de structure, les valeurs de temps de réverbération aux fréquences moyennes, sont de plus de 8 secondes, ce qui a pour conséquence de réduire l'intelligibilité des instructions données par les entraineurs lors des phases d'entraiment. Avec l'aide d'un logiciel d'acoustique architecturale, l'ajout de matériau absorbant dans la coupole du dôme a été évalué comme possible correction acoustique pour réduire le temps de réverbération.

**Mots clefs :** acoustique des salles, simulation, modèle virtuel, dôme pneumatique

## Abstract

Air dome is an air supported structure, it is a building composed by one or more layers of continuous flexible membrane anchored to the ground, inflated and supported by pressurized air. In the present work are shown the results of acoustic measurements inside an air dome. In this type of structure, reverberation time values, at the medium frequencies, are over 8 s and therefore with an excessive reverberation the ability to speech understanding is significantly reduced, so that during the training phases it does not correctly allow to understand the instructions of coaches. With the help of a software for architectural acoustics, a possible acoustic correction to reduce reverberation time was evaluated by introducing some sound absorbing material into the turned area of the air dome.

**Keywords:** room acoustics, simulation, virtual model, air dome

## 1 Introduction

The air domes are pressure static coverings born from the need of golf seasonal game management; they are generally used as coverings for tennis or basketball courts. The concept of air dome was proposed by the architect David H. Geiger at the Expo '70 in Osaka (Japan). The architect Davis Brody was encharged to realize the pavilion; he projected a 30 floor high building, but, due to the frequent earthquakes and typhoons in Japan, he asked for a help to the engineer Geiger. When the congress approved only half of the allocated budget, Geiger decreased the suggested height using an inflatable roof.

Air blowers pump air into the structure, creating a necessary air pressure that allows the fabric envelope to stay inflated. Air structures offer great flexibility and can be used as seasonal, permanent or temporary structures. Despite this flexibility, however, air structures are built to code and are structurally solid– even in inclement weather. An Air dome can be simply assembled and removed by acting on PVC cloth, inflation pressure that keeps the structure raised. PVC cloth is traction and tear resistant.

In order to prevent deterioration from moisture and ultraviolet radiation, these materials are coated with polymers such as PVC and Teflon. The pressure switch cover is essentially composed by a double PVC cover membrane, with a central tunnel with a semi-cylindrical shape stabilized by the introduction of compressed air, closed at both ends by two sails. In this last case, part of the air that is fed into the structure from the blower, is conveyed with a PVC hose between the two membranes, keeping in turn them into pressure and then detached from each other creating an air gap that allows to contain thermal losses. The air domes can take different shapes: ovals, half cylinders and hemispheres.

The air domes have the advantage to be cheaper than the traditional structures, and their installation is quicker, being flexible and very light structures.

Air domes reverberation time values are usually about 7.0 s at the middle frequencies. A reverberation like that is not a bad thing when it is related to sport activities because it can help to make sporting event more exciting and pleasing to the fans, while an excessive reverberation, present in that places, influences negatively acoustic performances, when air domes halls are used for sport training especially during the winter seasons. Therefore in these structures reverberation is excessive and the speech understanding is greatly reduced and during training phases it does not allow to understand coaches instructions correctly.

In some cases large sport halls have been adapted into places for music or for conversation events and the literature

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relates measurements and acoustic corrections of this places. [1, 2]. In California [3] the building of "San Diego Sports Arena" with an initial reverberation time of 5.0 s, was reduced to 2.6 s to perform music shows. For the "Arena of the Compaq Center", before it became the "Sanctuary of the Lakewood Church" [4], a mid-frequency acoustic correction was made. In Europe [5] some authors have proposed to cover the ceiling with absorbent materials as a solution for acoustic correction of sport halls or council rooms [6]. In the present work, using a software for architectural acoustics, it was evaluated a possible acoustic correction for reducing reverberation time, by introducing some sound absorbing materials into the turned area of the air dome.

## 2 Case study

The air dome considered is composed by a vaulted structure made of double-layer PVC sheets. The base has a simple rectangular shape of 35 m x 26 m, a maximum height of 11 m and the volume of 7,000 $m^3$, which is suitable for various sports such as basketball or volleyball. The pressure switch cover is essentially composed by a double PVC cover membrane, with a semi-cylindrical central tunnel stabilized by the introduction of compressed air, closed at both ends by two sails. In this last case, part of the air that is fed into the structure from the blower, is conveyed with a PVC hose between the two membranes, keeping in turn them into pressure and then detached one from each other creating an air gap that allows to contain thermal losses. Figure 1 shows the air dome internal view; Figure 2 shows the axonometric view of air dome; while Figure 3 shows the sections.

**Figure 1:** Air dome internal view

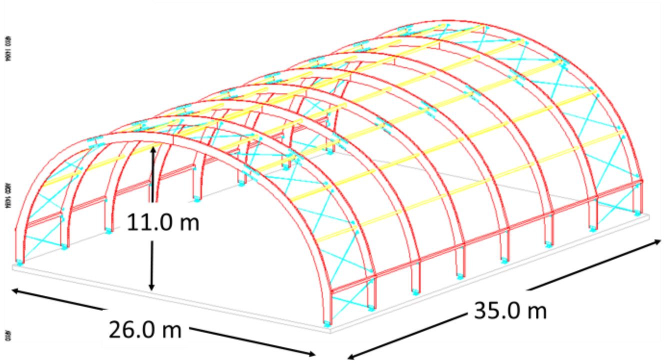


**Figure 2:** Axonometric view of the air dome

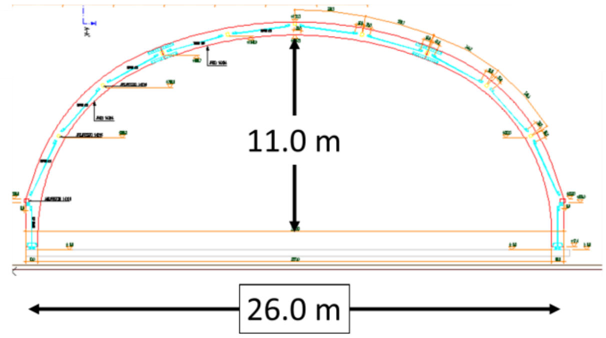


**Figure 3:** Section of the air dome

## 3 Acoustic measurements

In order to analyse the acoustic characteristics of the air dome, acoustic measurements were carried out using an impulsive sound source located in the central part of the playground. Acoustic measurements were done using small firecrackers as impulsive sound source. The height of sound source was 1.3 m from floor using a tripod. A microphone BRAHMA was used for recording impulse responses in nine different receivers located in fixed positions.

The choice of the impulsive sound source generated by the explosion of small firecrackers is due to the lack of electrical power and, as the room is huge, you have an adequate relation between signal and noise. The acoustic measurements were done in empty conditions, without spectators. The recorded impulse responses were elaborated with software Dirac 4.0, analysing the acoustic parameters defined in the ISO 3382-1 [7], such as reverberation time ($T_{30}$), EDT, clarity ($C_{80}$ and $C_{50}$), definition ($D_{50}$), center time (Ts) and sound transmission index for speech intelligibility (STI).

In room acoustic evaluations, clarity represents the degree to which different reflections arrive and are perceived by the listener and it is assessed as an early-to-late arriving sound energy ratio. This ratio can be calculated for either a 50 ms or an 80 ms early time limit, depending on whether it respectively relates to conditions for speech or music [8].

In fact Table 1 reports the optimal values of the acoustic parameters in different musical listening conditions or speech intelligibility.

**Table 1:** Optimal acoustic parameter values for the different listening conditions.

| Parameters | EDT, s | $T_{30}$, s | $C_{80}$, dB | $D_{50}$ |
|---|---|---|---|---|
| Values for musical performances | 1.8 < EDT < 2.6 | $1.6 < T_{30} < 2.2$ | $-2 < C_{80} < 2$ | < 0.5 |
| Values for speech performances | 1.0 | $0.8 < T_{30} < 1.2$ | | > 0.5 |

The acoustic procedure and post processing methodology were similar to those used in other spaces, such as large theatre and Odeon of Pompeii [9], theatre of

Benevento [10]. as well as in many other theatres. Figure 4 shows positions of the sound source of nine receivers (microphones) in the playground. Figures 5 to 10 show the average measured values of different acoustic parameters, in the octave bands from 125 Hz to 4.0 kHz. The values of EDT, $T_{30}$, $C_{50}$, $C_{80}$ , $D_{50}$ and Ts are reported at the interval of standard deviations, while STI = 0.18.

To understand the flutter echo phenomena, the parameter Centre Time (Ts) was considered. The center time (Ts) values of 90-160 ms are usually calculated for typical concert halls, principally at low-mid frequency.

In the air dome the measured values of EDT and $T_{30}$ exceed on average 7.0 s at middle frequencies, while the average values of $D_{50}$ = 0.15 and the average values of $C_{80}$ = -8.0 dB.

The average value of the STI = 0.31. Low values of clarity ($C_{80}$) and definition ($D_{50}$) indexes and a large deviation of these parameters, sensible to the early part of impulse response, denoted that this acoustic parameters change from point to point.

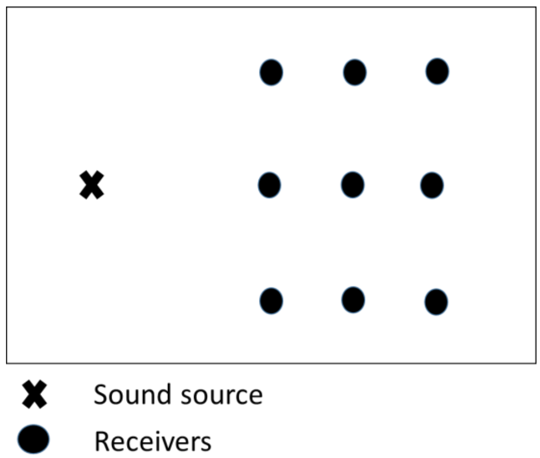


**Figure 4:** Plant with the indication of the sound source (X) and receivers (microphones) (o).

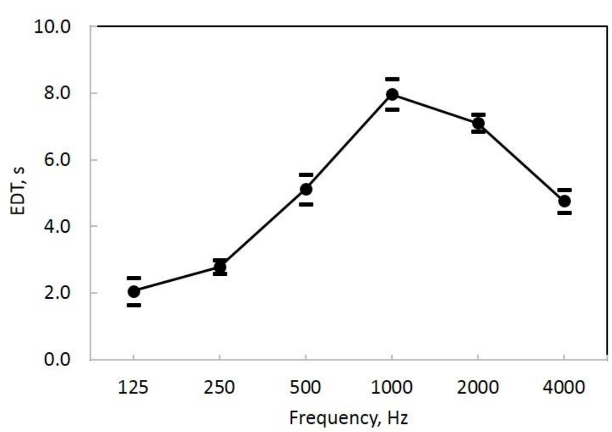


**Figure 5:** Measured average values of EDT ± 1 standard deviation.

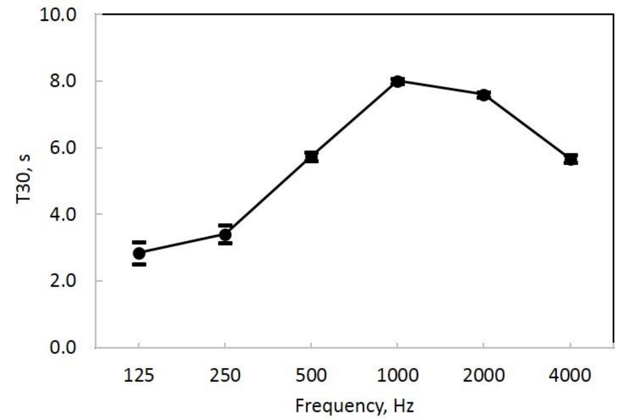


**Figure 6:** Measured average values of T30 ± 1 standard deviation.

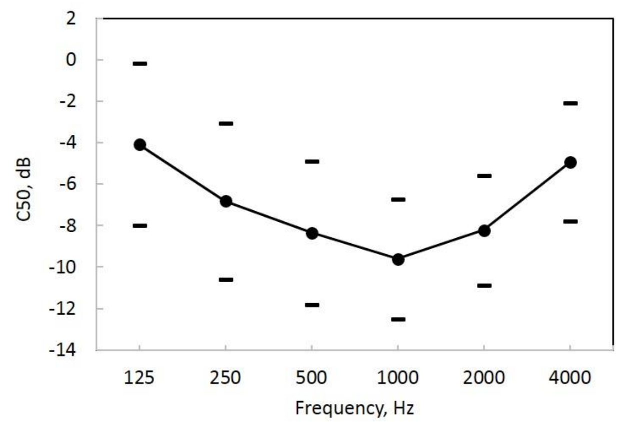


**Figure 7:** Measured average values of $C_{50}$ ± 1 standard deviation.

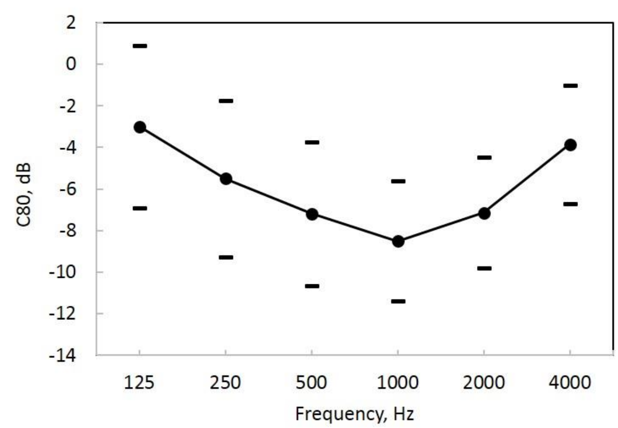


**Figure 8:** Measured average values of $C_{80}$ ± 1 standard deviation.

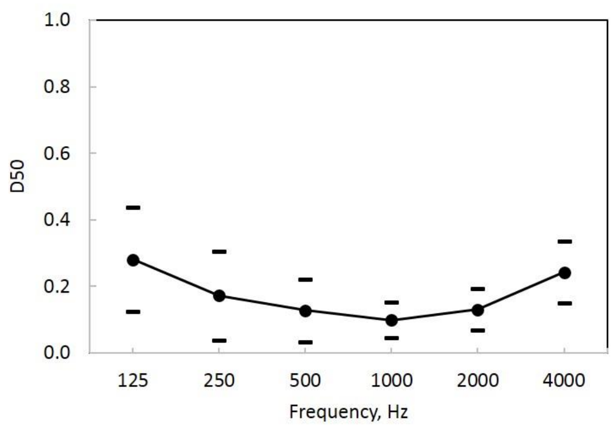


**Figure 9:** Measured average values of D50 ± 1 standard deviation.

For EDT and $T_{30}$ only at a frequency of 125 Hz there is a sensible variation of the standard deviation. The measurement results suggest that a room in the observed state might not be well suited neither for musical concerts nor for speech intelligibility.

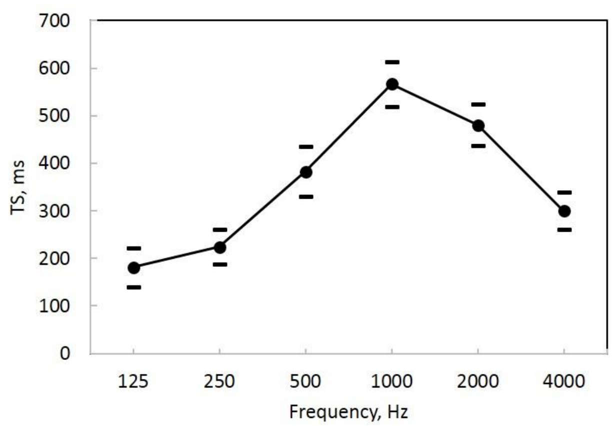


**Figure 10:** Measured average values of Ts ± 1 standard deviation.

The air dome does not perform criteria of good listening for speech. Furthermore to better understand acoustic conditions of the room they made measurements of the sound pressure levels during a basketball match; the measurements were performed with a Class 1 sound level meter, model SOLO 01dB, the LeqA = 72.5.0 dBA.

## 4 Air dome virtual model

To evaluate a possible solution to reduce reverberation time and allow the achievement of an adequate acoustic comfort to listen to music shows, the software for architectural acoustics "Odeon" was used with a virtual model realized by a 3D cad [11, 12].

Odeon is a software that uses the theory of rays with the method of images for the acoustic simulation. Using the software Odeon the calibration of the numeric model is required. Chosen the T30 as a reference parameter, the calibration consists in the change of the absorbent coefficient values of the walls so that the reverberation time measured doesn't coincide with the theoretic one. In the specific case the calibration was stopped when the difference between the time measured and the time calculated is inferior to 5% of all the octave bands calculated included between 125 and 4000 Hz.

Figure 11 shows 3D virtual model with the omnidirectional sound source and the receivers positioned in the measurement points put in the playground.

Table 2 shows the absorbent coefficient values of the air dome absorbent coefficient values used for the virtual model calibration.

**Table 2:** Octave band air dome sound absorption values

| Frequency, | 125 | 250 | 500 | 1 k | 2 k | 4 k |
|---|---|---|---|---|---|---|
| Air dome | 0.28 | 0.22 | 0.14 | 0.08 | 006 | 0.02 |
| Ground | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 |

The absorbent coefficient values decrease with the increase of the frequency, because at low frequencies the reverberation time is lower. Furthermore Figure 12 shows the 3D air dome virtual model with the indication of the sound reflection under the vault.

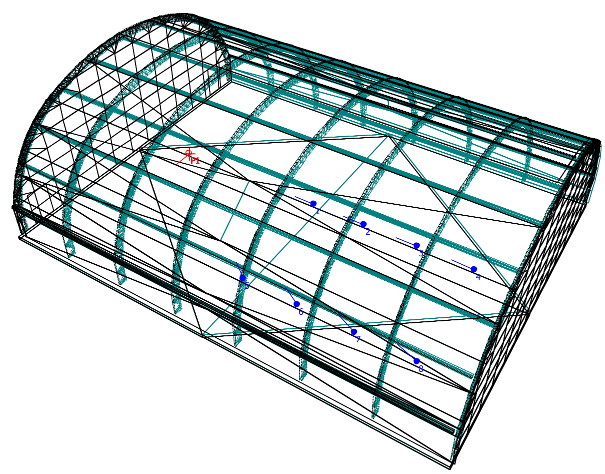

**Figure 11:** 3D air dome virtual model with omnidirectional sound source and receivers

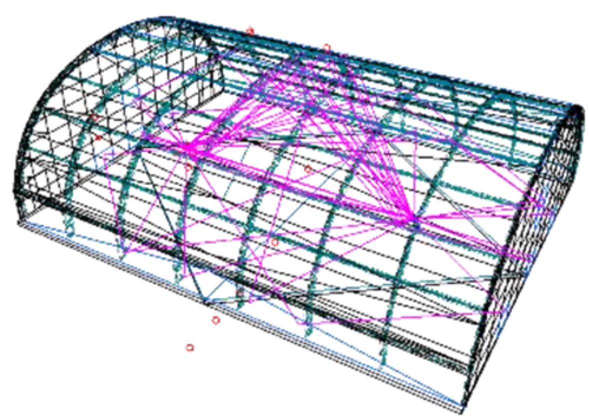

**Figure 12:** 3D air dome virtual model with the indication of the sound reflection under the vault

## 5 Materials for acoustic correction

To obtain the sound absorption coefficients to be used for a computer simulation of the dome acoustic correction, an impedance tube (tube of Kundt) was used according to ISO 10534-2 [13]. With this method is possible to obtain the absorbent coefficient measurements at normal incidence using samples of diameter 10 cm inside the tube. The impedance tube had an internal diameter of 10 cm (which correspond to an upper frequency limit of 2000 Hz). Various hypothesis have been evaluated for the acoustic correction of the air dome. Polyester absorbent panels, with thickness of 3.0 cm were chosen. The configuration considered corresponds to the suspended panels, so the panels are inserted as baffles.

Table 3 reports the octave band values of measured sound absorption coefficient for sample backed with anechoic termination. This average value of the absorbent coefficient is obtained from measurements with four different specimens (thickness 3.0 cm), the value of absorbent coefficient at 4.0 kHz is obtained by extrapolation of measured data. Figure 13 shows the particular of the material used for the acoustic correction; the material has closed cells, so it does not absorb moisture so that it can resist for long without an expensive maintenance.

**Table 3:** Octave band sound absorption coefficients measured according to ISO 10534-2 method

| Frequency, | 125 | 250 | 500 | 1 k | 2 k | 4 k |
|---|---|---|---|---|---|---|
| Alfa ($\alpha$) | 0.4 | 0.4 | 0.4 | 0.8 | 0.8 | 0.8 |

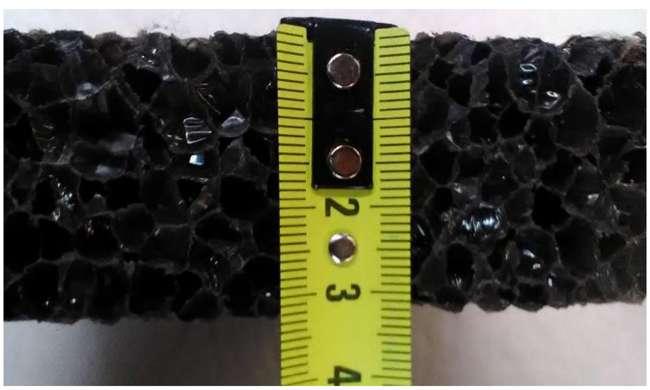

**Figure 13:** Particular of the material used for the acoustic correction

## 6 Acoustic correction

For the considered hypothesis it has been estimated an installation of a suspended panels made of soundproof material placed under webbed beams. The suspended panels are composed of 530 $m^2$ of absorbent material.

Figure 14 shows a 3D air dome virtual model with the absorbent panels under the vault. In this configuration the reverberation time estimated at 1.0 kHz is about 3.5 s. The presence of audience on the seats is negligible because the area covered by audience is little extended.

Figure 15 shows the comparison between reverberation time measured versus calculated time with the acoustic correction.

Figure 16 shows the comparison between $D_{50}$ versus calculated with acoustic correction Figure 17 shows the comparison between $C_{80}$ versus calculated with acoustic correction.

Figure 18-A shows the impulse response of the central point, obtained with the Odeon software, closer to the source in the configuration without acoustic correction. Whereas, Figure 18-B shows the impulse response obtained with the Odeon software for the same point, when sound absorbing material for acoustic correction is applied under the dome.

Figure 19-A shows the impulse response of the central point, obtained with the Odeon software, farther to the source in the configuration without acoustic correction. Whereas, Figure 19-B shows the impulse response obtained with the Odeon software for the same point, when sound absorbing material for acoustic correction is applied under the dome. In the absence of acoustic correction, the effect of reflection on the dome can be noted, while in the presence of acoustic correction this effect is far more limited.

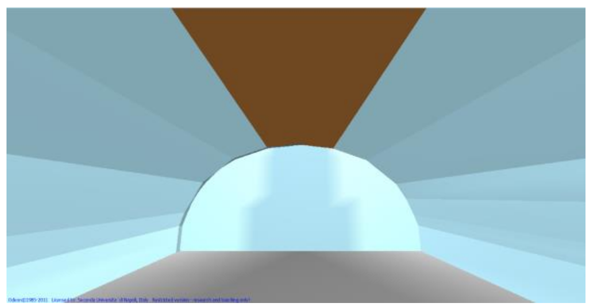

**Figure 14:** 3D air dome virtual model with the absorbent panels under the vault

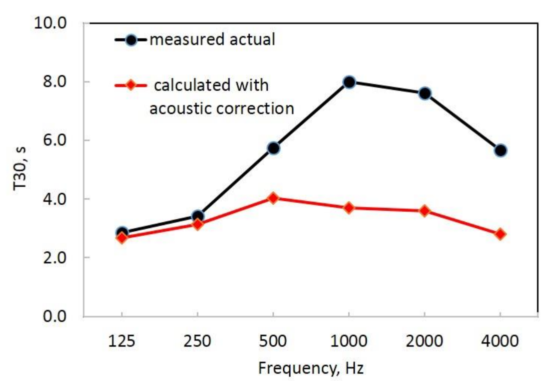


**Figure 15:** Comparison between reverberation time measured versus calculated time with acoustic correction.

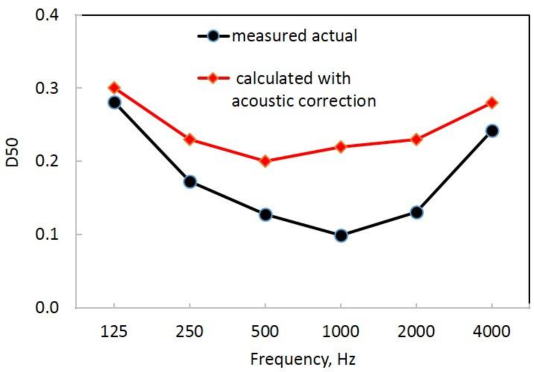


**Figure 16:** Comparison between $D_{50}$ measured versus calculated with acoustic correction

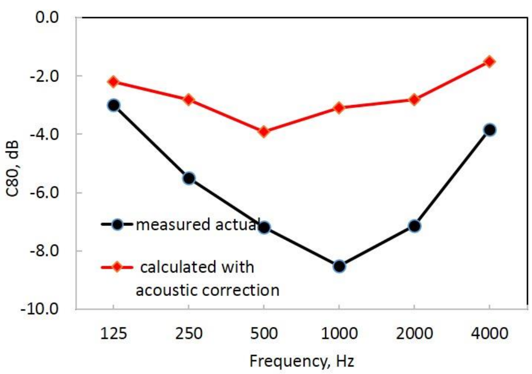


**Figure 17:** Comparison between $C_{80}$ measured versus calculated with acoustic correction

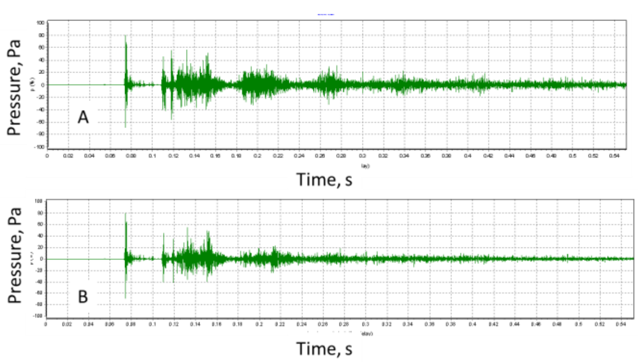


**Figure 18:** (A) Impulse response of the central point obtained with the Odeon software, without acoustic correction. (B) Impulse response when sound absorbing material for acoustic correction is applied under the dome.

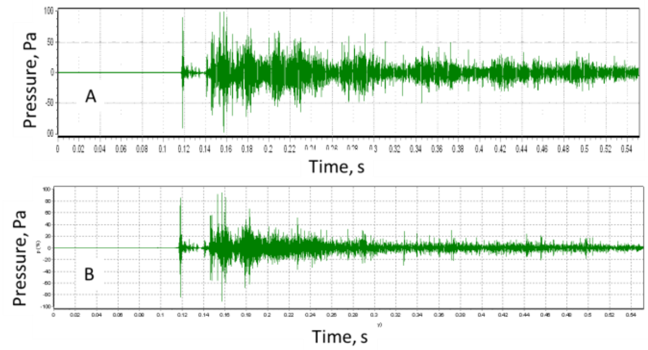


**Figure 19:** (A) Impulse response of the central point obtained with the Odeon software, without acoustic correction. (B) Impulse response when sound absorbing material for acoustic correction is applied under the dome.

## 7 Discussion

Several authors have provided the optimal values of the acoustic parameters in order to listen to music or speech since the reverberation time is not enough to assess the acoustic goodness of a room. For a good understanding of speech, it is worth evaluating the definition $D_{50}$ (in a room with good speech understanding conditions, $D_{50} > 0.50$). For good music listening, it is worth evaluating the parameter $C_{80}$ (in a hall, the values of $C_{80}$ should be in the range between -2 dB and 2 dB for good music listening).

For the air dome the $D_{50}$ values decrease with the frequency, because the reverberation time values increase. The value of $T_{30}$ at the frequency of 1.0 kHz is about 8.0 s, so in this configuration the speech understanding is very low. After the acoustic correction the reverberation time at the frequency of 1.0 kHz is 2.5 s. The value of STI = 0.42.

## 8 Conclusions

This paper shows the possibility to obtain a good acoustic correction inside an air dome, using absorbent material under the vault. The sound absorbing material chosen for the acoustic correction shows good absorbing values and is a moisture proof material, as moisture is very frequent in these spaces. The study of the acoustics of the air dome carried out with acoustic measurements and the architectural acoustics software "Odeon" gives useful information to improve the acoustic of the room. The proposed hypothesis of acoustic correction has been designed in order to reduce reverberation time, consequently improving the listening condition during the training activities.

## References

[1] K.A. Penman , J.S. Englund, and N. Stanier. Reverberation and noise levels in sports areas, *Journal of the Acoustical Society of America*, 62, 1046-1048, 1977.

[3] M. Klasco, and R. Paolini. San Diego Sports Arena.*dB*, November / December, 29 -34, 1987.

[4] B. Russ. From Sports Arena to Sanctuary: Taming a Texas-sized reverberation time. *Acoustics Today*, 7, 24-28, 2011.

[5] R. Bošnjakovic and D. Tomic. Acoustical Treatment of Multipurpose Sport Halls. *Proceedings of 3rd Congress of the Alps Adria Acoustics Association*, 27–28 September, Graz – Austria. 2007.

[6] G. Iannace. Sport Hall Acoustics. *Noise & Vibration Worldwide*, 46, 22-29, 2015. doi:10.1260/0957-4565.46.10.22

[7] ISO-3382, Acoustics – Measurement of the reverberation time of rooms with reference to other acoustical parameters

[8] M. Barron, *Auditorium Acoustics and Architectural Design*, E&FN SPON, London, 1993

[9] U. Berardi, G. Iannace and L. Maffei. Virtual reconstruction of the historical acoustics of the Odeon of Pompeii, *Journal of Cultural Heritage*, 19, 555-566, 2015. doi: 10.1016/j.culher.2015.12.004

[10] G. Iannace and A. Trematerra. The rediscovery of Benevento Roman Theatre Acoustics. *Journal of Cultural Heritage* 15(6). 698-703, 2014. doi: 10.1016/j.culher.2013.11.012

[11] www.odeon.dk, user manual - version 11

[12] X. Zeng, C. L. Christensen and J. H. Rindel. Practical methods to define scattering coefficients in a room acoustics computer model. *Applied Acoustics*, 67, 771 – 786, 2006.

[13] EN ISO 10534-2. Acoustics - Determination of sound absorption coefficient and impedance in impedances tubes Transfer-function method

# SHAPE OPTIMIZATION OF REACTIVE MUFFLERS USING THRESHOLD ACCEPTANCE AND FEM METHODS

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## Résumé

L'optimisation de forme des silencieux réactifs sous contrainte d'espace a une grande importance dans la conception d'environnements moins bruants. Dans ce travail, les performances acoustiques de trois types de silencieux soumis à un espace limité sont étudiées. Une analyse d'optimisation de forme est effectuée en utilisant un algorithme d'optimisation appelé Threshold Acceptance (TA). La conception optimale obtenue est analysée par la méthode des éléments finis (FEM : Finit Element Method). Cette approche numérique est basée sur la maximisation de la perte de transmission acoustique (STL : Sound Transmission Loss) à l'aide de la méthode de transfert de matrices (TMM : Transfer Matrix Method) qui est une méthode de modélisation basée sur le modèle de propagation d'onde plane. La solution en élément finis utilisée pour analyser la STL est basée sur la méthode de puissance acoustique, un code de calcul standard utilisé pour analyser en 3D l'atténuation acoustique des silencieux par la méthode FEM. La capacité acoustique des silencieux obtenus est évaluée en comparant la solution FEM à la méthode analytique. Les résultats montrent que la valeur maximale de la STL est précisément située à la tonalité ciblée. En outre, la performance acoustique du silencieux avec tube à l'entrée et à la sortie prolongée se trouve être supérieure aux autres types de silencieux. Par conséquent, cette approche fournit un schéma rapide pour l'optimisation de la forme des silencieux réactifs.

**Mots clefs :** silencieux réactifs, algorithme d'optimisation, méthode de transfert de matrices, puissance acoustique

## Abstract

The shape optimization of reactive muffler under space constraint becomes of great importance in the design of quieter environments. In this paper, the acoustical performance of three different expansion-chamber mufflers with extended tube under space constraint is presented. A shape optimization analysis is performed using a novel scheme called Threshold Acceptance (TA). The best design obtained by the shape optimization method is analyzed by Finite Element Method (FEM). This numerical approach is based on the maximization of the Sound Transmission Loss (STL) using the Transfer Matrix Method (TMM). The TMM method is a modelling method based on the plane wave propagation model whereas the FEM solution is based on the acoustical power method. A standard computational code is used to analyze the sound attenuation of the mufflers by the FEM method in 3D. The acoustical ability of the mufflers is than assessed by comparing the FEM solution with the analytical method. Results show that the maximal STL is precisely located at the desired targeted tone. In addition, the acoustical performance of muffler with inlet and outlet extended tube is found to be superior to the other ones. Consequently, this approach provides a quick scheme for the shape optimization of reactive mufflers.

**Keywords:** reactive muffler, threshold acceptance, transfer matrix method, sound acoustic power

## 1 Introduction

The use of mufflers for exhaust noise attenuation with limited space in vehicles and machinery pushes the researchers to develop different numerical modelling methods [1-2]. The most common type of linear acoustic model applies classical electrical filter theory. This theory is widely known as the transfer matrix method (TMM) [3]. Though, it is also referred to as the 4-pole parameter method [4-6]. A technique that combines the use of transfer matrix approach and finite element method in the study of duct acoustics is reported after by Craggs in 1989 [7].

Since the muffler space dimension is often limited to meet the demands of operation and maintenance, there are increasing interests in designing mufflers in order to optimize the STL using shape optimization methods [8]. A simple expansion chamber muffler is studied by Bernhard [9] by using a shape optimization method with a non-constrained space condition. To obtain a good acoustical performance for the shape optimization of mufflers, novel schemes have appeared such as Genetic Algorithm (GA) and Simulating Annealing (SA) [10]. Yeh and al. [11] studied the shape optimal design of a double expansion-chamber muffler under space constraints by using SA and GA optimizers. Their study reveals that either SA or the GA is applicable in the optimization analysis. Both algorithms are much easier to use compared to gradient-based optimizers which require a good starting.

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This paper is built on the acoustic attenuation study of three types of expansion-chamber mufflers with extended tubes under space constraints by using a deterministic acceptance criterion optimizer named Threshold Acceptance combined with a finite element analysis.

## 2 Mathematical models

The reactive mufflers adopted for the noise reduction in this work are composed of three types of inlet/outlet extended tube mufflers as shown in figure 1 (a, b and c). The three kinds of mufflers are left inlet extended tube, right outlet extended tube and inlet/outlet extended tube. The different acoustical elements of the mufflers (acoustic pressure *p* and acoustic particle velocity *u*) are illustrated in figure 2 (a, b and c). These elements within the left and right extended tube mufflers are represented by seven nodes and for the inlet and outlet extended tube muffler are represented by eight nodes.

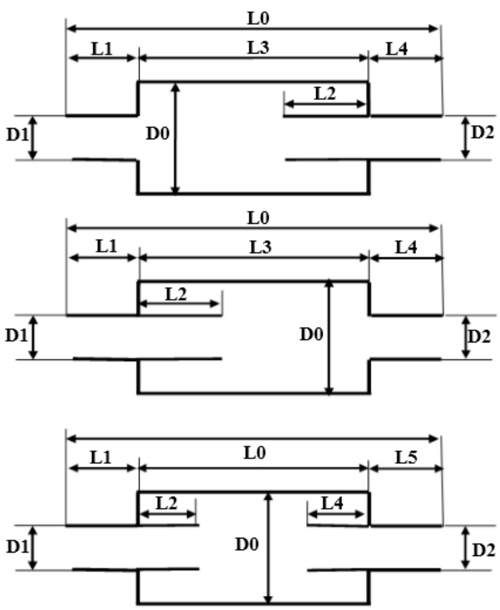



**Figure 1:** Sketches of expansion-chamber mufflers with extended tube: (a) inlet, (b) outlet, (c) inlet & outlet side.

Two different approaches were used to analyze the acoustical performance of the three mufflers under chosen limited space (L= 1.5 m, D0 = 0.3). These approaches are the sound acoustic power and the transfer matrix methods. The most widely used acoustical performance to characterize the sound attenuation of the mufflers is the sound transmission loss (STL). This value depends only on the muffler and not on the sound source. It is considered as the best parameter to use when comparing different methods and designs [12]. The optimization method is developed using Matlab tool. For the FEM and simulation analysis a standard computational code named COMSOL Multiphysics Tools is used.

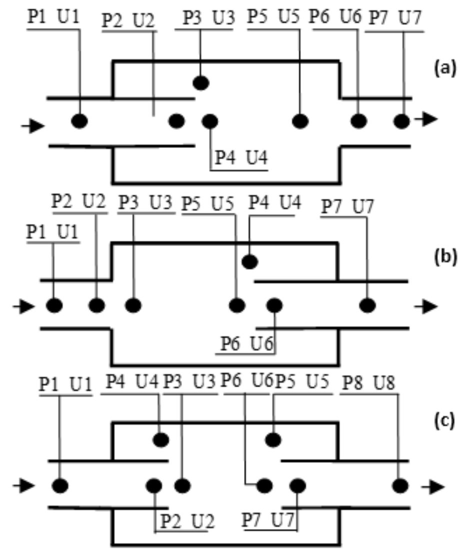



**Figure 2:** Sketches of the one-dimensional plane wave propagation of expansion-chamber mufflers with extended inlet/outlet tube: (a) inlet, (b) outlet, (c) inlet & outlet side.

## 3 Numerical assessment

### 3.1 Shape optimization method

A system of four-pole matrix evaluating the acoustical performance (sound transmission loss) is used and derived by using a decoupled numerical method called Transfer Matrix Method (TMM). This method uses 2 x 2 matrices to relate two variables at planes on either side of an acoustic component. The matrices for individual components can be readily combined to form a single and overall matrix that describes the behavior for a multi-component muffler's system [5, 13].

A Threshold Acceptance method, a deterministic acceptance criteria optimizer similar to simulated annealing, is applied to the optimizations of the mufflers.

### 3.2 Theoretical formulation

The acoustical system of four-pole matrix uses 2 x 2 matrix to relate two variables at planes (acoustic pressure ($p$) and volume velocity ($u$)) on either side of an acoustic component [14, 15]. To describe the overall acoustic property of the muffler we need to relate all the individual matrices in one total transfer matrix of the system as:

$$T = T_1\, T_2\, T_3 ... \tag{1}$$

The following general transfer matrix may be written to relate the state variables of straight duct and expansion/contracted ducts respectively for the three kinds of expansion-chamber mufflers with extended tube.

$$T_i = e^{-j\frac{M_i k L_i}{\left(1-M_i^2\right)}} \begin{bmatrix} \cos\left(\frac{kL_i}{1-M_i^2}\right) & j\sin\left(\frac{kL_i}{1-M_i^2}\right) \\ j\sin\left(\frac{kL_i}{1-M_i^2}\right) & \cos\left(\frac{kL_i}{1-M_i^2}\right) \end{bmatrix} \quad (2)$$

$$T_i^{'} = \begin{bmatrix} 1 & 1 \\ 1 & \frac{S_i}{S_{i-1}} \end{bmatrix} \quad (3)$$

Where $T_i$ , $T_i^{'}$ , $M$, $k$, $L$, $S$, $j$ and $i$ are respectively the transfer matrix of straight ducts, the transfer matrix of expansion/contracted ducts, the Mach number, the wave number, the length of element, the area of the element, the imaginary unit section and the $i$ represent the $i^{th}$ node.

For the cross-sectional discontinuity case, the transition elements used are shown in the first column of table 1. By using decreasing element-subscript values with distance from the noise source, the cross-sectional areas upstream and downstream of the transition ($S_3$, $S_2$ and $S_1$) are related through:

$$C_1S_1 + C_2S_2 + S_3 = 0 \quad (4)$$

The constants $C_1$ and $C_2$ are selected to satisfy the compatibility of the cross-sectional areas across the transition.

**Table 1:** Parameters values of transition elements

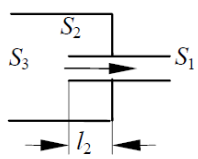
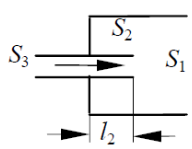
| Element Type | $C_1$ | $C_2$ | K |
|---|---|---|---|
|  | -1 | -1 | $\frac{1-\frac{S_1}{S_3}}{2}$ |
|  | -1 | 1 | $\left(\frac{S_1}{S_3}-1\right)^2$ |

Table 1 also shows the pressure loss coefficient $K$ for each configuration that accounts for conversion of some mean-flow energy and acoustical field energy into heat at the discontinuities. As indicated, $K \leq 0.5$ for area contraction, while $K \rightarrow (S_1 / S_3)^2$ for area expansions at large values of $S_1 / S_3$.

The four-pole matrices of the ducts with cross-sectional discontinuities [15] (for Mach number M = 0) is given by

$$T_{discontinuitate} = \begin{bmatrix} \dfrac{\frac{1}{C_2}}{C_1\left(-j\frac{c_0}{S_2}\cot kl\right)} & \begin{matrix} 0 \\ 1 \end{matrix} \end{bmatrix} \quad (5)$$

The computation of the transfer matrix for the whole silencer is achieved based on the individual matrices which relate the pressure $P$ and mass velocity $V$ at the inlet and outlet. The individual matrices are calculated separately for every sector as:

$$\begin{pmatrix} p_i \\ \rho_0 c_0 u_i \end{pmatrix} = \begin{bmatrix} T_{11} & T_{12} \\ T_{21} & T_{22} \end{bmatrix} \begin{pmatrix} p_o \\ \rho_0 c_0 u_o \end{pmatrix} \quad (6)$$

Where $T_{11}$, $T_{12}$, $T_{21}$ and $T_{22}$ are referred to as the four poles of the acoustical system. The STL of a muffler is calculated as [5]:

$$STL\left(f,Q,R_1,R_2,R_3,R_4\right) = 20\log\left[\left[\frac{Y_n}{Y_1}\right]^{1/2}\left|\frac{T_{11}+T_{12}Y_n+T_{21}Y_1+T_{22}(Y_1/Y_n)}{2}\right|\right] \quad (7)$$

$Y_1$ is calculated for the input pipe and $Y_n$ for the output pipe. Where for the inlet and outlet mufflers:

$R_1=D_1/D_0$, $R_2=D_2/D_0$, $R_3=L_3/L_0$, $R_4=L_2/L_0$, $L_1=1/2(L_0-L_3)$, $L_4=1/2(L_0-L_3)$, $L_0=L1+L3+L5$.

And for Inlet and outlet muffler:

$R_1=D_1/D_0$, $R_2=D_2/D_0$, $R_3=L_3/L_0$, $R_4=L_6/L_0$, $L_1=1/2(L_0-L_3)$, $L_5=1/2(L_0-L_3)$, $L_2=1/2(L_3-L_6)$, $L_0=L1+L3+L5$

Because of the remarkably pure tone noise effect at 300 Hz [11], noise elimination at this frequency by shape optimization is applied.

### 3.3 Threshold Acceptance

Threshold Acceptance method applied in this work is a metaheuristic algorithm. It's a modification of the well-known Simulated Annealing metaheuristic method (SA) [16]. The SA method draws its analogy from the annealing process of solids. The solid is heated to a high temperature and gradually cooled in order to crystallize. It must be cooled slowly such that the atoms have enough time to align themselves to reach a minimum energy state. This analogy can be used in combinatorial optimizations with the states of the solid corresponding to the feasible solution. The energy at each state correspond to the value of objective function and the minimum energy represent the optimal solution [17].

SA always accepts moves to neighboring solutions that improve the objective function value. More precisely, the solution ($S$) in the neighborhood $N(S)$ is accepted as the new current solution if $\Delta \leq 0$, where $\Delta = C(S')-C(S)$ in which C denotes the objective function. To allow the search to escape a local optimum, a stochastic approach is used to direct the search. A move that worsens the objective function value is accepted with a probability $e^{-\Delta/T}$ if $\Delta > 0$. $T$ is a parameter called the Temperature.

The value of $T$ varies from a relatively large value to a small value close to zero. An initial temperature and an optimization temperature are chosen in this interval at each step of optimization of the algorithm. This method of temperature selection is identical to that of the metal cooling process.

The TA algorithm uses a predetermined deterministic sequence to decide whether a new point is selected or not (if worse than the current point), whereas SA method probabilistically determines a new point selection at every iteration.

Dueck and Scheurer [18] simplified the SA procedure by leaving out the probabilistic element in accepting worse solutions. Instead, they introduced a deterministic threshold ($\tau$) and a worse solution is accepted if its difference to the incumbent solution is smaller or equal to the threshold. The new procedure is named Threshold Acceptance.

The key components of TA are the function *g(t)* that determines the lowering of the threshold during the course of the procedure, the stopping criteria as well as the methods used to create initial and neighboring solutions. The main advantages of TA are its conceptual simplicity and its excellent performance on different combinatorial optimization problems [19].

## 3.4 FEM Analysis method

In the second part of this paper, we analyze the acoustical performance of the obtained shape optimized mufflers by using FEM. The available numerical tool used for analyzing muffler performance includes 3D linear acoustic codes with and without mean flow is using FEM methods where the most important effect of flow is included by altering the boundary conditions without considering the mean flow [20].

The following equation defines the attenuation $d_w$ (dB) of the acoustic energy is:

$$d_w = 10\log\left(\frac{w_0}{w_i}\right) \tag{8}$$

Here $w_0$ and $w_i$ denote the outgoing power at the outlet and the incoming power at the inlet respectively. Each of these quantities can be calculated as an integral over the corresponding surface:

$$w_0 = \int_{\partial\Omega} \frac{|p|^2}{2\rho c_s} dA \tag{9}$$

$$w_i = \int_{\partial\Omega} \frac{p_0^2}{2\rho c_s} dA \tag{10}$$

The FEM model solves the problem in the frequency domain using the time-harmonic pressure of the acoustic application mode. The STL is calculated directly with the computational code tool using the acoustic power method at the inlet and at the outlet of the acoustic system. Each model of muffler is simulated using a three dimensional model and is meshed using the Lagrange-quadratic elements. A harmonic pressure of 1*Pa* is specified at the inlet of the muffler and a radiation boundary condition is applied at the inlet and outlet of the muffler. A material with default values of air is created with density of 1.2 kg/m3 and with sound speed of 340 m/s. By using the default values of air, the acoustic damping is not taken into account.

## 3.5 Case studies

To check the transmission loss model on the single inlet chamber muffler a comparison between theoretical and experimental data [3] is realized.  As shown in figure 3, there is a coherence between the theoretical and experimental data. Hence, the transmission loss model is acceptable and can be applied to the studied models.

The available space selected for the mufflers is 0.5 m in width 0.5 m in height and 1.5 m in length. To obtain the best acoustical performance within a fixed space a pure tone noise with 300 Hz is applied for the mufflers as a numerical case. Also to reach an initial transition probability of 0.5 of the TA method, the initial temperature is selected as 0.2 and the flow rate (Q = 0.01 (m3/s)) is preset in advance to simplify the optimization for the mufflers [10]. The selected space constraints ranges for the three types of mufflers are:

$R_1$: [0.1, 0.5], $R_2$: [0.1, 0.5], $R_3$: [0.2, 0.8] and $R_4$: [0.2, 0.8]

After shape optimization of the mufflers, a numerical analysis by FEM is presented in the second part of this work. To assess the acoustical performance of each idealized muffler a 3D simulation analysis is applied for the FEM. The used parametric solver provides results for a range of frequencies. The software computes integrals in the power expressions using boundary integration coupling variables and it plots the resulting attenuation versus frequency.

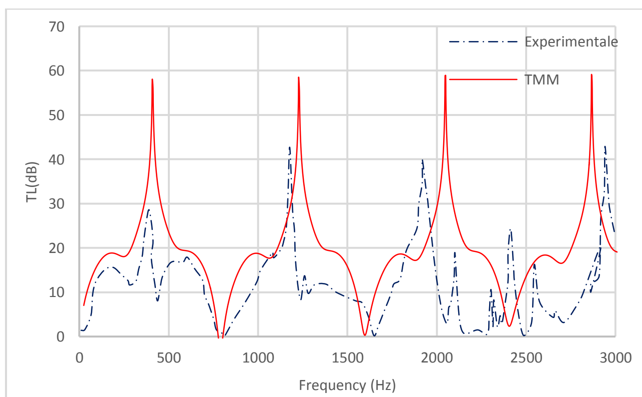



**Figure 3:** Performance curves of STL, comparison between TMM theoretical model and experimental values of simple expansion chamber muffler with extended tube [3].

## 3.6 Objective Function

The accuracy of the TA optimization depends on two control parameters: the cooling rate (*CR*) and the number of iteration ($I_{max}$). The optimization process with respect to objective functions (*Obj1*, *Obj2* and *Obj3*) is performed by varying these parameters. From formula (7), the objective functions and their ranges are reduced and set for the three mufflers respectively as following:

$$Obj_1(X_1, X_2, X_2) = STL(D_1, D_2, L_2) \tag{11}$$

$$Obj_2(X_1, X_2, X_3, X_4, X_5, X_6, X_7) = STL(D_1, D_2, D_3, L_1, L_2, L_3, L_5) \tag{12}$$

$$Obj_3(X_1, X_2, X_3, X_4, X_5, X_6, X_7, X_8, X_9) = STL(L_1, L_2, L_3, L_5, L_7, D_1, D_2, D_3, D_4) \tag{13}$$

## 4 Results and discussion

For the three studied muffler, the shape optimization is performed by testing various sets of parameters with respect to the pure tone of 300Hz. After this step, the STL is calculated with respect to various pure tones (300Hz, 500Hz, 700Hz and 800Hz) by using the optimal design obtained by optimization method.

Following the optimization process, the muffler is analyzed using a FEM and simulation analysis. The simulation analysis start by applying the required boundary conditions and then the meshing with a coarse predefined mesh sizes of 0.25mm on the x direction scale.

### 4.1 First case: Expansion-chamber muffler with inlet extended tube

The optimization process for the expansion chamber muffler with inlet extended tube using various sets of TA parameters is performed. The result is shown in table 2. The optimal design data is obtained at the cooling rate $CR$ = 0.99 and iteration number $I_{max}$ = 2500. This result reveals that the minimal state is achieved at the higher cooling rate.

Figure 4 plots the STL with respect to frequency in various design case. It shows that the STL values are roughly maximized at the desired frequencies and that the highest values of the $CR$ and $I_{max}$ parameters gave the highest STL. Therefore, the method of variation of these two parameters play essential role in TA optimization and using this method to find the better design solution is reliable.

The second step is to measure the STL of the optimized muffler with respect to various pure tones. Table 3 gives the obtained results of STL and this result is displayed in figure 5. This result reveals that increasing the pure tone expands the frequency bandwidth and the STLs are precisely maximized at the desired frequencies.

The 3D analysis of propagation modes is performed on the related optimal muffler's size with respect to pure tone of 2000 Hz. Figure 6 displays the internal sound pressure distribution at 2000 Hz. The pressure field varies primarily with the y direction while it is nearly constant in the z direction. The reason is that 2000 Hz is just higher than the cutoff frequency for the first symmetric propagating mode excited by the incoming wave.

We observe also that the selected frequencies how the sound pressure level distributions near the muffler inlet and outlet is important.

**Table 2:** Sound Transmission Loss of a single expansion-chamber muffler with targeted tone of 300 Hz and various $CR$ and $I_{max}$.

| Case | TA parameters | Results | | | | |
|---|---|---|---|---|---|---|
| | | R1 | R2 | R3 | R4 | STL (dB) |
| 1 | $CR$ = 0.90 $I_{max}$ = 250 | 0,213580756 | 0,202902009 | 0,795782821 | 0,781371602 | 20,56 |
| 2 | $CR$ = 0.93 $I_{max}$ = 250 | 0,202115091 | 0,201711434 | 0,754368947 | 0,784125799 | 20,84 |
| 3 | $CR$ = 0.96 $I_{max}$ = 250 | 0,201675088 | 0,201648677 | 0,759723156 | 0,799370566 | 20,96 |
| 4 | $CR$ = 0.99 $I_{max}$ = 250 | 0,200000412 | 0,200000004 | 0,799999932 | 0,799993991 | 21,39 |
| 5 | $CR$ = 0.99 $I_{max}$ = 400 | 0,200000297 | 0,2000003 | 0,799999879 | 0,798941011 | 21,38 |
| 6 | $CR$ = 0.99 $I_{max}$ = 800 | 0,200002602 | 0,200875503 | 0,730584904 | 0,759254063 | 20,70 |
| 7 | $CR$ = 0.99 $I_{max}$ = 1500 | 0,200090707 | 0,200030674 | 0,798613086 | 0,799459378 | 21,37 |
| 8 | $CR$ = 0.99 $I_{max}$ = 2500 | 0,2 | 0,2 | 0,799999989 | 0,8 | 21,39 |
| 9 | $CR$ = 0.99 $I_{max}$ = 6000 | 0,200000024 | 0,20000003 | 0,799999982 | 0,799999802 | 21,38 |
| 10 | $CR$ = 0.99 $I_{max}$ = 10000 | 0,20010711 | 0,200000046 | 0,799996472 | 0,799834974 | 21,38 |

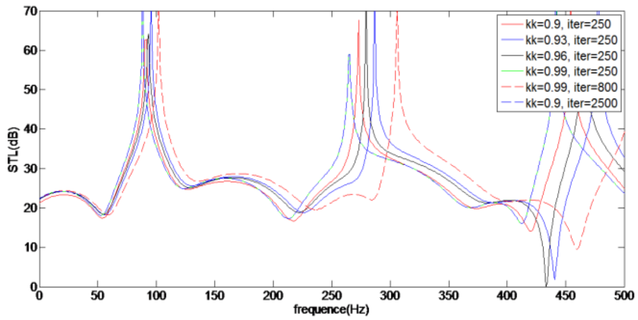



**Figure 4:** Performance curves of STL with respect to various maximal iterations ($I_{max}$) by TA [To = 0.2].

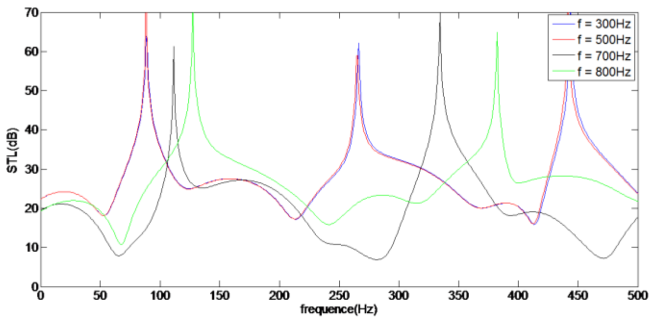



**Figure 5:** STL with respect to frequencies of a Single expansion-chamber for various pure tones [Targeted frequency: 300, 500, 700 and 800 Hz].

Figure 7 plots the theoretical transmission loss based on the TMM (blue line) and the FEM solution (red line) as a function of frequency. It's noticeable that the FEM solution has an upper frequency limit for its validity. This limit is the cut-on frequency which defines the frequency range where only plane waves can propagate. Above this frequency, also higher modes can propagate.

At frequencies higher than approximately 1400 Hz, the plot's behavior is more complicated and there is generally less damping. This is because, for such frequencies, the tube supports not only longitudinal resonances but also cross-sectional propagation modes. We notice that a discrepancy exists between the theoretical and the FEM solution even below the cut-on frequency. This discrepancy is due to that the elementary transfer matrices depend on the element which is modelled. The sound transmission loss is independent of the source and requires an anechoic termination at the downstream end. Thus, it does not involve neither the source nor the radiation impedance of the termination whereas the sound depend only on the sound source and does not allow the transfer matrices of the acoustic system to be obtained.

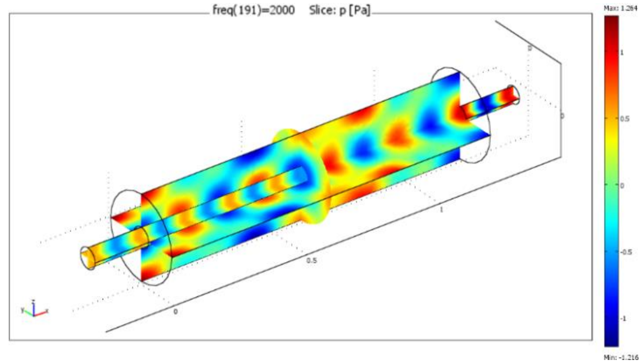



**Figure 6:** Optimized FEM model of the single expansion-chamber muffler with extended inlet tube and internal sound pressure distribution at 2000 Hz.

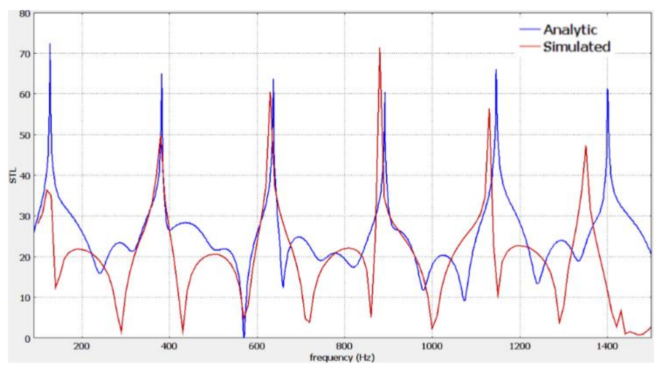



**Figure 7:** Muffler transmission loss versus frequency of single expansion-chamber muffler with inlet tube: Theoretical solution (blue line) and simulated solution (red line).

The differences obtained in the results may also be attributed to the finite element formulation used in simulation method which is Lagrange elements method or to the computational mesh method (density and refinement).

## 4.2 Second case: Expansion-chamber muffler with outlet extended tube

The shape optimization of a single expansion chamber with outlet extended tube with various sets of TA parameters with respect to the pure tone of 300Hz is performed. The result is shown in table 4.

**Table 3:** Sound Transmission Loss of a single expansion-chamber with respect to various targeted frequencies ($CR$= 0.99, $I_{max}$ = 2500).

| Case | Target frequency | Results | | | | |
|---|---|---|---|---|---|---|
| | | R1 | R2 | R3 | R4 | STL (dB) |
| 1 | 300 Hz | 0,20010935 | 0,20002373 | 0,79999829 | 0,79634202 | 22,11 |
| 2 | 500 Hz | 0,2 | 0,2 | 0,8 | 0,8 | 25,68 |
| 3 | 700 Hz | 0,23791322 | 0,46730982 | 0,78741544 | 0,64434888 | 115,50 |
| 4 | 800 Hz | 0,240834641 | 0,200736224 | 0,798729653 | 0,555818811 | 119,69 |

**Table 4:** Optimal STL for a Double expansion-chamber muffler at various *CR* and $I_{max}$ (Targeted tone of 300 Hz).

| Case | TA parameters | Results | | | | |
|---|---|---|---|---|---|---|
| | | R1 | R2 | R3 | R4 | STL (dB) |
| 1 | $CR$ = 0.90, $I_{max}$ = 250 | 0,203970327 | 0,201331047 | 0,558524911 | 0,543257189 | 45,88 |
| 2 | $CR$ = 0.93, $I_{max}$ = 250 | 0,200021583 | 0,200056361 | 0,799954532 | 0,792637646 | 49,31 |
| 3 | $CR$ = 0.96, $I_{max}$ = 250 | 0,202802328 | 0,201578411 | 0,743689134 | 0,582243506 | 47,86 |
| 4 | $CR$ = 0.99, $I_{max}$ = 250 | 0,200001586 | 0,200065207 | 0,799935022 | 0,799861406 | 49,34 |
| 5 | $CR$ = 0.99, $I_{max}$ = 400 | 0,200181749 | 0,200032438 | 0,774090819 | 0,710186438 | 48,81 |
| 6 | $CR$ = 0.99, $I_{max}$ = 800 | 0,200259589 | 0,200027709 | 0,725326708 | 0,798266388 | 48,77 |
| 7 | $CR$ = 0.99, $I_{max}$ = 1500 | 0,200000723 | 0,200001383 | 0,794550818 | 0,79040532 | 49,27 |
| 8 | $CR$ = 0.99, $I_{max}$ = 2500 | 0,20000004 | 0,200366626 | 0,799441375 | 0,799332283 | 49,31 |
| 9 | $CR$ = 0.99, $I_{max}$ = 6000 | 0,200014056 | 0,20000011 | 0,799806204 | 0,798318742 | 49,33 |
| 10 | $CR$ = 0.99, $I_{max}$ = 10000 | 0,200001007 | 0,200000002 | 0,799986738 | 0,799974136 | 49,34 |

As indicated, the optimal design data can be obtained at the cooling rate $CR$ = 0.99 and iteration number $I_{max}$ = 10000. This result indicate that the minimal state is achieved at the higher cooling rate. The acoustic performance of STL (with respect to frequency in various design case) is presented and plotted in figure 8. Obviously, the results revealed that the highest values of *CR* and $I_{max}$ gave the highest STL. At higher frequency, the plots behavior is more complicated and the STL are roughly maximized at the desired frequencies.

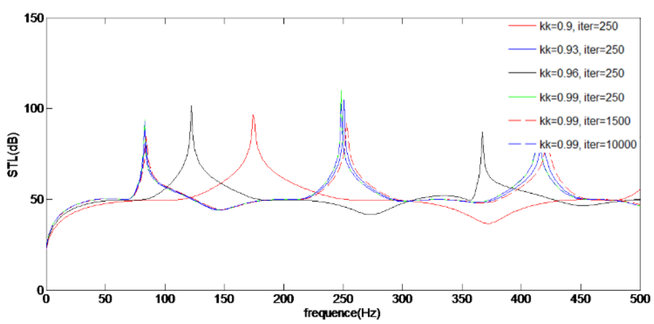



**Figure 8:** Performance curves of STL with respect to various maximal iterations ($I_{max}$) by TA [To = 0.2].

After this optimization step, the optimal design with respect to various pure tones is measured and summarized in table 5. The optimal STL curves with respect to targeted frequencies are plotted and depicted in figure 9.

The levels of the STL increase in the low frequency range whereas it decreases at high frequencies. It shows that the STLs are precisely maximized at the selected frequencies.

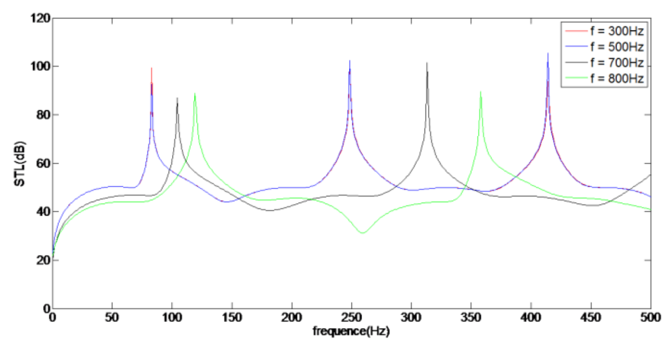



**Figure 9:** STL with respect to frequencies of a Double expansion-chamber muffler for various pure tones [Targeted frequency: 300, 500, 700 and 800 Hz].

**Table 5:** Optimal STLs for a double expansion-chamber with respect to various targeted frequencies (*CR* =0.95, $I_{max}$ =50000).

| Case | Targette frequency | Results | | | | |
|---|---|---|---|---|---|---|
| | | R1 | R2 | R3 | R4 | STL (dB) |
| 1 | 300 Hz | 0,200006541 | 0,200000986 | 0,799564675 | 0,799695519 | 50,09 |
| 2 | 500 Hz | 0,2 | 0,2 | 0,8 | 0,8 | 53,64 |
| 3 | 700 Hz | 0,236345991 | 0,20385864 | 0,653122383 | 0,776829314 | 141,22 |
| 4 | 800 Hz | 0,212167477 | 0,271589061 | 0,797512162 | 0,55666418 | 167,92 |

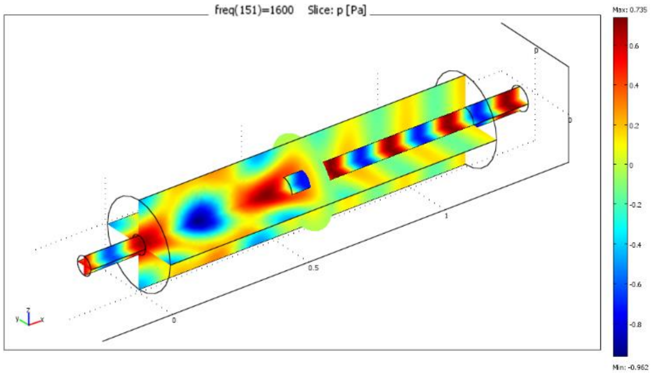



**Figure 10:** Optimized FEM model of the one expansion-chamber muffler and internal sound pressure distribution at 1600Hz (absolute pressure).

The 3D analysis of propagation modes is performed on the related optimal muffler's size with respect to pure tone of 1600 Hz. The result is shown in Figure 10.

We notice that for the selected frequencies how the sound pressure level distributions near the muffler outlet is important.

Figure 11 plots the theoretical transmission loss (blue line) and the numerical solution (red line) as a function of frequency. We observe a discrepancy between the analytic and the simulated plots is higher than those of muffler with inlet extended tube. Also we notice that the FEM solution present an upper frequency limit for its validity which is around 1500 Hz. This limit is the cut-on frequency defined previously.

### 4.3 Third case: Expansion-chamber muffler with inlet and outlet extended tube

The result of the shape optimization of a the expansion-chamber muffler with inlet and outlet extended tubes based on various sets of TA parameters and with respect to the pure tone of 300Hz is shown in table 6.

As indicated, the optimal design data can be obtained at the cooling rate $CR$ = 0.99 and iteration number $I_{max}$ = 10000. It reveals that the minimal state is achieved at the higher cooling rate. The acoustic performance of STL (with respect to frequency in various design case) is presented and plotted in figure 12.

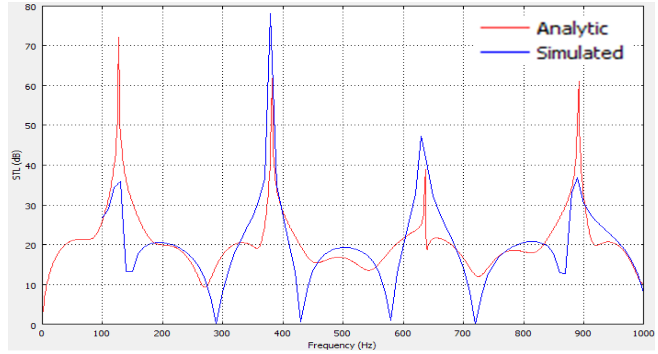



**Figure 11:** Single expansion-chamber muffler with outlet extended tube, transmission loss versus frequency (Theoretical solution (blue line) and simulated solution (red line)).

Obviously, the results reveal that the highest values of the maximum of ($CR$ and $I_{max}$) gave the highest STL. At higher frequency, the plots behavior is more complicated and the STLs are roughly maximized at the desired frequencies. The STL of the muffler's optimal sizes with respect to various pure tones are summarized in table 7. The optimal STL curves with respect to targeted frequencies are plotted and depicted in figure 13. We notice that the levels of the transmission loss increase in the low frequency range, whereas the levels decrease at high frequencies and it shows that the STLs are precisely maximized at the selected frequencies. After that the 3D analysis of propagation modes is performed on the related optimal muffler's size with respect to pure tone of 2820 Hz.

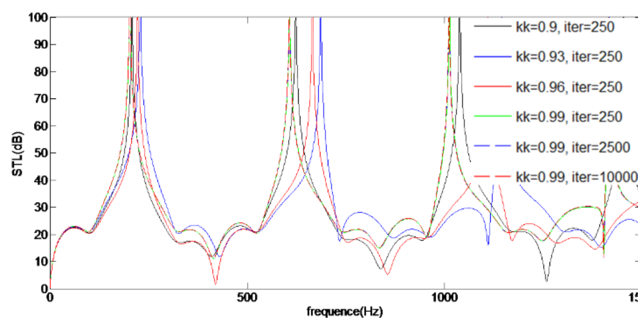



**Figure 12:** Performance curves of STL with respect to various maximal iterations ($I_{max}$) by TA [To = 0.2].

**Table 6:** Optimal STL for a Double expansion-chamber muffler at various $CR$ and $I_{max}$ (Targeted tone of 300 Hz).

| Case | TA parameters | Results | | | | |
|---|---|---|---|---|---|---|
| | | R1 | R2 | R3 | R4 | STL (dB) |
| 1 | $CR$ = 0.90, $I_{max}$ = 250 | 0,493648572 | 0,200000002 | 0,7995986 | 0,318039144 | 21,31 |
| 2 | $CR$ = 0.93, $I_{max}$ = 250 | 0,20040704 | 0,200000006 | 0,799314016 | 0,380509568 | 21,38 |
| 3 | $CR$ = 0.96, $I_{max}$ = 250 | 0,49057417 | 0,200938109 | 0,796665887 | 0,358804594 | 21,11 |
| 4 | $CR$ = 0.99, $I_{max}$ = 250 | 0,203700732 | 0,200000011 | 0,793509973 | 0,302309052 | 21,62 |
| 5 | $CR$ = 0.99, $I_{max}$ = 400 | 0,202774242 | 0,200670458 | 0,779143091 | 0,355654907 | 21,29 |
| 6 | $CR$ = 0.99, $I_{max}$ = 800 | 0,200927913 | 0,200172399 | 0,799162089 | 0,30012868 | 21,65 |
| 7 | $CR$ = 0.99, $I_{max}$ = 1500 | 0,200775342 | 0,2 | 0,79999581 | 0,301296387 | 21,67 |
| 8 | $CR$ = 0.99, $I_{max}$ = 2500 | 0,200003813 | 0,2 | 0,799782514 | 0,300023475 | 21,67 |
| 9 | $CR$ = 0.99, $I_{max}$ = 6000 | 0,200037701 | 0,200000069 | 0,799983124 | 0,30018866 | 21,67 |
| 10 | $CR$ = 0.99, $I_{max}$ = 10000 | 0,2 | 0,2 | 0,799999999 | 0,3 | 21,68 |

**Table 7:** Optimal STLs for a double expansion-chamber with respect to various targeted frequencies (*CR*=0.95, $I_{max}$ =50000).

| Case | Targette frequency | Results | | | | |
|---|---|---|---|---|---|---|
| | | R1 | R2 | R3 | R4 | STL (dB) |
| 1 | 300 Hz | 0,200378869 | 0,200005517 | 0,799887167 | 0,300417339 | 22,49 |
| 2 | 500 Hz | 0,200004147 | 0,200002545 | 0,61423464 | 0,300032686 | 23,14 |
| 3 | 700 Hz | 0,200428366 | 0,200205034 | 0,799994563 | 0,300055346 | 23,25 |
| 4 | 800 Hz | 0,20000481 | 0,20001244 | 0,799999553 | 0,300006166 | 28,62 |

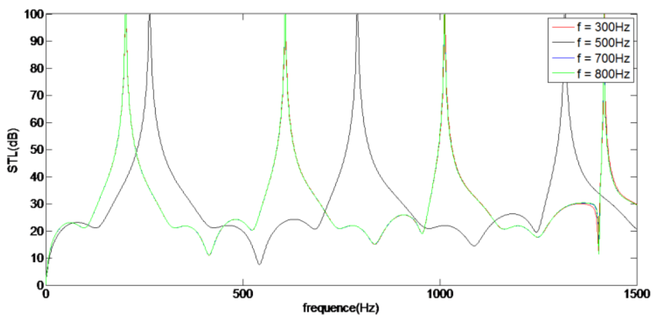



**Figure 13:** STL with respect to frequencies of the expansion-chamber muffler with inlet and outlet extended tube for various pure tones [Targeted frequency: 300, 500, 700 and 800 Hz].

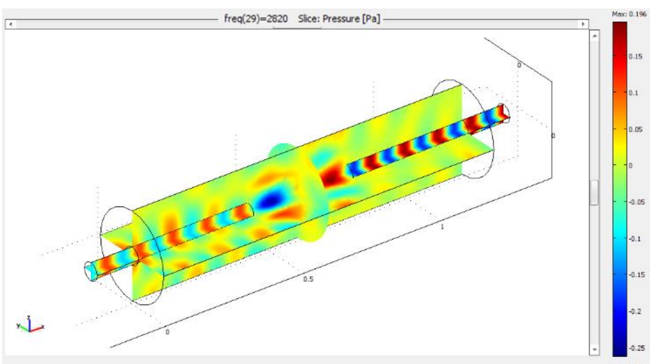



**Figure 14:** Optimized FEM model of the one expansion-chamber muffler and internal sound pressure distribution at 2820Hz (absolute pressure).

Figure 14 shows the internal sound pressure distribution at 2820 Hz of the single expansion chamber muffler, we observe how the sound pressure level is higher than those of muffler with inlet extended tube and outlet extended tube in high frequencies. We can also observe for the selected frequencies how the sound pressure level distributions near the muffler outlet is important.

Figure 15 plots the theoretical transmission loss (blue line) and the simulation solution (red line) as a function of frequency.

The FEM solution has an upper frequency limit for its validity. This limit is the cut-on frequency defined previously, we observe a discrepancy between the analytic and the simulated plots as the theoretical results based on the TMM depend on the element which is modelled and not on the source whereas the FEM solution is based on the sound acoustic power method and depend only on the sound source and do not allow the transfer matrices of the acoustic system to be obtained.

The discrepancy obtained in the results plot may also be attributed to the used formulation in finite element method which is Lagrange elements or to the computational meshed applied (density and refinement).

## 4.4 Comparison

The optimal muffler's design data for the three kinds of expansion-chamber mufflers with extended tubes (inlet, outlet and inlet/outlet tubes) with space constraint is summarized in table 8 and plotted in figure 16. As shown, it is obvious that the attenuation of the single expansion-chamber muffler with inlet and outlet tube is a little superior to the other mufflers. Consequently it gives the best acoustical performance.

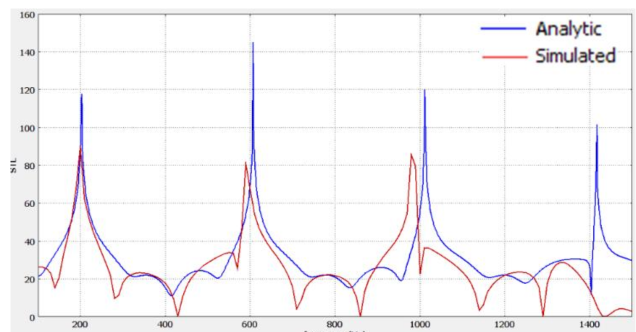



**Figure15:** Expansion-chamber muffler with inlet and outlet extended tube transmission loss versus frequency: theoretical solution (blue line) and simulated solution (red line).

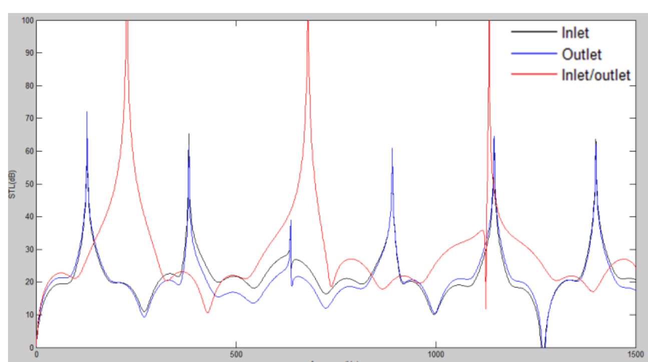



**Figure 16:** Comparison of the optimal STL level with respect to the three kinds of optimized mufflers within a same space-constrained space [broadband noise].

**Table 8:** Comparison of the acoustical performance with respect to three kinds of optimized mufflers within same space-constrained situation.

| Case | Type of muffler | Results | | | | |
|---|---|---|---|---|---|---|
| | | R1 | R2 | R3 | R4 | STL (dB) |
| 1 | Inlet extended tube | 0,24 | 0,2 | 0,79 | 0,55 | 119,69 |
| 2 | Outlet extended tube | 0,21 | 0,27 | 0,79 | 0,55 | 167,92 |
| 3 | Inlet & Outlet extended tubes | 0,2 | 0,2 | 0,79 | 0,3 | 28,62 |

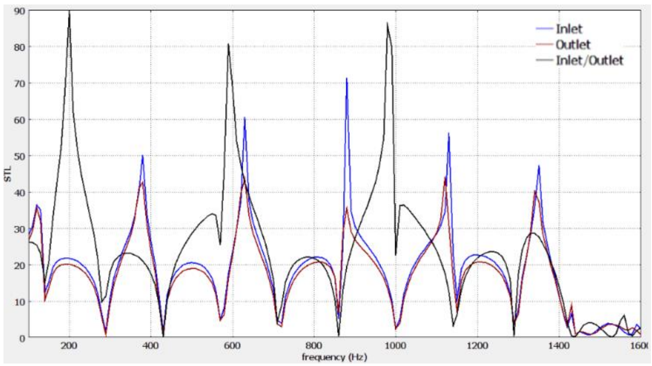



**Figure 17:** FEM solution of the three shape optimized mufflers: Inlet (Blue line), Outlet (Red line) and Inlet/Outlet (Black line).

For the FEM Solution, the sound attenuation of the three optimized mufflers' configurations as a function of frequency is plotted in figure 17. The plot shows that for the muffler with inlet and outlet extended tubes gives the highest acoustical performance than the two other mufflers.

## 5 Conclusion

The shape optimization of three kinds of reactive mufflers with extended tubes under space constraints is applied in this paper by using a novel scheme Threshold Acceptance coupled with 3D finite element analysis.

This numerical analysis using TA optimizer shows to be an efficient method to optimize reactive mufflers under space constraints.

The TA optimizer is based on the TMM method applying the plane wave theory as well as four-pole transfer matrices. This optimization method shows the importance of the two TA parameters (*CR* and $I_{max}$) in the optimization process also it reveals that this method is valid when the influence of high order modes can be neglected. For the FEM, the analysis is performed on the shape optimized mufflers obtained by the TA methods. This method depend on the sound acoustic power. The TMM depends on the element which is modelled and not on the sound source, whereas for the sound acoustic power depend on the sound source and does not allow the transfer matrices of the acoustic system to be obtained, also it depends on the finite element formulation and the computational mesh method.

The comparison between the numerical prediction based on the TMM and the FEM solution based on the sound acoustic power displays higher discrepancies in the curves for the muffler with outlet extended tube than the two other mufflers.

The TA optimization method and the FEM method showed that the single expansion-chamber muffler with inlet and outlet extended tube provides considerably better STL values than the two other mufflers.

Consequently, the approach of the optimal design of STL proposed in this study is quite efficient in dealing with the reactive mufflers within a space-constrained situation.

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## References

[1] E. Bécache, A.S. Bonnet-Bendhia, Numerical simulation of exhaust muffler, An homogenized finite element method, Cari'06, Vol. 1, (2006), pp. 1-10.

[2] S. Bilawchuk, K. R. Fyfe, Comparison and implementation of the various numerical methods used for calculating transmission loss in silencer systems, Appl. Acoust. , Vol. 64, (2003), pp. 903-916.

[3] S.N.Y. Gerges, R. Jordan, F.A. Thime, J.L. Bento Coelho, J.P. Arenas, Muffler Modeling by Transfer Matrix Method and Experimental Verification, J. Braz. Soc. Mech. Sci.& Eng., Vol. 27(2), (2005), pp. 132-140.

[4] M.L. Munjal, Plane wave analysis of side inlet/outlet chamber mufflers with mean flow, Appl. Acoust. , Vol. 52(2), (1997), pp. 165-175.

[5] M. L. Munjal, Acoustics of Ducts and Mufflers with Application to Exhaust and

Ventilation System Design, John Wiley & Sons, Inc., (1987), pp. 328.

[6] M. L. Munjal, Advances in the acoustics of flow ducts and mufflers, Vol. 15(2), (1990), pp. 57-72.

[7] K.S. Andersen, Analyzing Muffler performance using the Transfer Matrix Method, Proc.Comsol Conf., (2008).

[8] L.J. Yeh, M.C. Chiu, Computer aided design on single expansion muffler under space constraints, Proc. 19th Nat. Conf. Mech. Eng., Vol. C7, (2002) pp. 625-633.

[9] R.J. Bernhard, Shape Optimization of Reactive Mufflers, Noise Cont Eng J, Vol. 27(1), (1986) pp. 10-17.

[10] M.C. Chiu, L.J. Yeh, Shape Optimization of Single-Chamber Mufflers with Side Inlet/Outlet by Using Boundary Element Method, Mathematic Gradient Method and Genetic Algorithm, Tamkang J. Sc. Eng., Vol. 12(1), (2009) pp. 85-98.

[11] L.J. Yeh, Y.C. Chang, M.C. Chiu, Shape Optimal Design on Double-Chamber Mufflers Using Simulated Annealing and a Genetic Algorithm, Turkish J. Eng. Env. Sc., Vol. 29, (2005), pp. 207- 224.

[12] P. W. Jones, N. J. Kessissoglou, Experimental study of the Transmission Loss of Mufflers used in Reperatory Medical Devices, Vol. 38, (2010) , pp. 1 - 13

[13] M.L. Munjal, A. G. GALAITISIS, I. L. VER, Passive Silencer, Noise Vibration Control Engineering, J. Wiley & Son, Inc., (2005), pp. 279-343.

[14] Davies, P.O.A.L., Realistic Models for Predicting Sound Propagation in Flow Duct Systems, Noise Cont. Eng. J. , Vol. 40,(1993), pp. 135-141. [18]

[15] Beranek, L.L, Istvan, L, Noise and Vibration Control Engineering: Principles and Applications, John Wiley & Sons, Inc, (1992), ISBN 0-471-61751-2.

[16] O. Bräysyean, J. Berger, A Threshold Accepting Metaheuristic for the Vehicle Routing Problem with Time Windows, Appl. Math., (2003).

[17] A. Metropolis, Rosenbluth, Equation of static calculations by fast computing machines, J. Chem. Phys., Vol. 21(6), (1953), pp. 1087-1092.

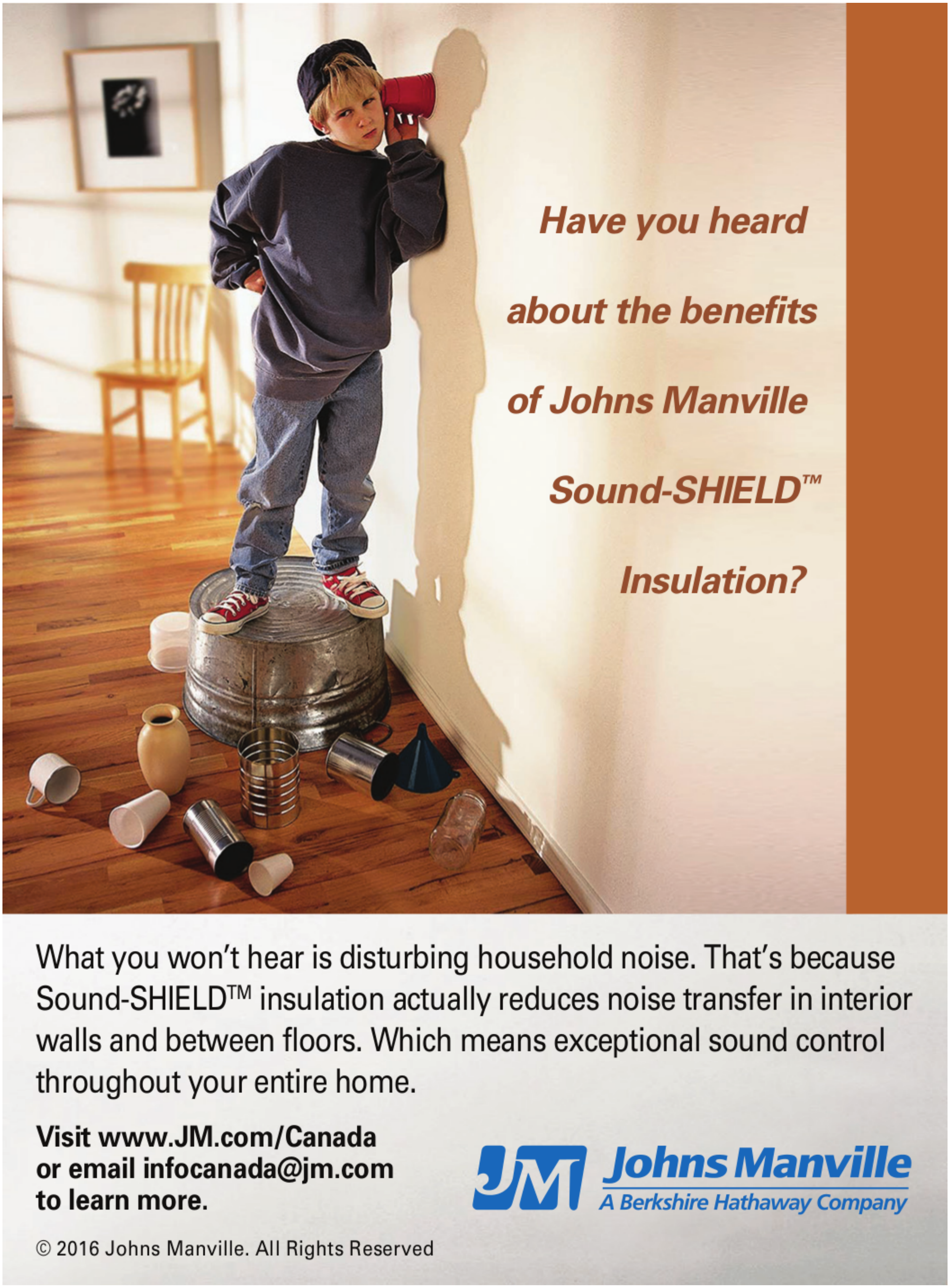
[18] G. Dueck, T. Scheurer, Threshold accepting: A general purpose optimization algorithm appearing superior to simulated annealing, J. Comput. Phy., vol. 90, (1990), pp. 161-175.

[19] M. Yagiura, T. Ibaraki, On metaheuristic algorithms for combinatorial optimization problems, Sys. Comp. Japan, vol. 32, (2001), pp. 33–55.

[20] COMSOL 3.4, Acoustics Module Model Library, COMSOL AB, (2007), pp. 74-85.

## Notation

| | |
|---|---|
| STL | Sound transmission loss (dB) |
| $w_i$ | time-averaged incident sound power |
| $w_t$ | transmitted sound power |
| q | dipole source |
| $p_0$ | incoming pressure wave |
| $k_1$ | wave vector |
| I | time-averaged sound intensity |
| $W$ | transmitted sound powers |
| $c_0$ | sound speed (m/s) |
| $D_0$ | diameter of the expansion chamber in the muffler (m) |
| $D_i$ | diameter of the i-th segment of the muffler (m) |
| $f_{01}$ | cut-off frequency (Hz) |
| $f$ | cyclic frequency (Hz) |
| $I_{max}$ | maximum iteration |
| j | imaginary unit |
| k | wave number |
| $CR$ | cooling rate in SA |
| $L_0$ | total length of the muffler (m) |
| $L_i$ | length at the i-th element (m) |
| $M_i$ | mean flow Mach number at the ith element |
| $Obj_i$ | objective function (dB) |
| $p$ | acoustic pressure (Pa) |
| $p_i$ | acoustic pressure at the ith node (Pa) |
| $u$ | acoustic particle velocity ($ms^{-1}$) |
| $u_i$ | acoustic particle velocity at the *ith* node ($m\ s^{-1}$) |
| $\rho_0$ | air density ($kg\ m^{-3}$) |
| $pb(T)$ | transition probability |
| $To$ | initial temperature (°C) |
| $Q$ | volume flow rate of venting gas ($m^3\ s^{-1}$) |
| $S_i$ | section area at the i-th node ($m^2$) |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**Canadian Acoustical Association**

**Minutes of the Board of Directors Meeting**
Held by video conference
1 June 2017

Present: Frank Russo (chair), Alberto Behar, Bryan Gick, Dalila Giusti, Michael Kiefte, Andy Metelka, Roberto Racca, Mehrzad Salkhordeh, Jérémie Voix; Umberto Berardi also participated as incoming Editor

Minutes taken by Jérémie Voix and Roberto Racca

Meeting called to order (after several technical glitches with video conference) at 14:32 EDT

**President's report (Frank Russo)**

Fall nominations to the Board of Directors

Frank presented the proposed slate of nominations to be put forward at the fall AGM by the past-president Christian Giguère:

- Umberto Berardi, Ryerson University
- Roberto Racca, JASCO Applied Sciences
- Dalila Giusti, Jade Acoustics
- Jérémie Voix, ÉTS
- Hugues Nélisse, IRSST
- Bryan Gick, UBC
- Bill Gastmeier, HGC Engineering
- Michael Kiefte, Dalhousie University
- Mehrzad Salkhordeh, dB Noise Reduction
- Alberto Behar, Ryerson University
- Andy Metelka, SVS Canada
- Joana Rocha, Carleton University

Award in Physical Acoustics

A potential award donation from Hugh Jones to the CAA which had been the subject of long-standing discussion due to funds administration issues has been withdrawn following his passing away because of complications from his last will, as communicated by his daughter. A motion to cease pursuing the idea of a prize in physical acoustics, which would have been funded by the donation, was made by Frank and seconded by Dalila; carried unanimously. Bill suggested that Frank write a note to the daughter about considering a donation in the future.

Outreach activities

Frank informed the BoD that with the ongoing valuable assistance of social media coordinator Huiwen Goy (earlier assisted by Steven Sonnenberg) the CAA has close to 600 followers on LinkedIn and 200 on Twitter. Promotion of acoustics related events including the CAA annual meeting is an important component of the information distributed through these channels.

Creation of a local chapter

Frank informed the BoD that he had been approached by John Swallow with a proposal to set up a Toronto chapter of the CAA and a request for some seed money for room rental and catering for its initial activities. After discussion of the matter the BoD agreed to sanction this local branch but with no ongoing financial commitment. Umberto indicated that he could have Ryerson University provide space for the chapter's meetings, and it was suggested that the CAA contribute up to $250 for refreshments at the first event. The BoD expressed willingness to offer the same initial monetary support to any group interested in opening a local CAA chapter. Andy suggested that that CAA publicise and promote this through the journal, website and social media. Motion to endorse the proposed Toronto chapter and to provide one-time support of up to $250 for room and refreshments for this and any other new chapters across the country was made by Frank and seconded by Michael; carried unanimously.

**Treasurer's report (Dalila Giusti)**

Dalila summarized the Treasurer's report that had been submitted in advance to all BoD members. She pointed out that apart from three overdue advertisers accounts, receivables were all current and the Association's finances were in good shape. All investments tied up until the following year or 2019 in funds performing satisfactorily. Dalila informed the BoD that she had not been able to file the annual financial report because of not having received the final report for the 2016 AWC conference (Vancouver), and the situation was becoming critical. Frank agreed to follow up with the organizers to prompt immediate release of the conference financials at least. Motion to accept the Treasurer's report was made by Frank and seconded by Andy; carried unanimously.

**Incidental business**

Jérémie made a brief incidental announcement prior to presenting the Editor's report: he and Christian Giguère had been contacted about hosting the International Conference on Sound and Vibration (ICSV) of 2019 (or 2020) in Montreal. He indicated that they were interested in organizing the event and were seeking endorsement and some organizational support by the CAA, with no commitment to any financial backing and indeed a potential revenue stream in case of profit. The BoD expressed no objections to this endorsement. Andy asked whether, in the same vein as the 2018 CAA annual conference partnering and combining with the ASA fall conference in Victoria, a similar arrangement might be sought with ICSV in 2019. Jérémie replied that the ICSV normally is held in July, a few months earlier than the CAA annual conference, so there was no reason to consider combining the two and no logistic conflicts would arise from holding the events separately.

**Editor's report (Jérémie Voix and Umberto Berardi)**

Umberto Berardi, new Editor in chief, reported that the June special issue covering Halifax acoustics activities had garnered very low interest – only one submission to date – and there were concerns about the viability of that issue. The idea was considered of expanding the city-specific issue to a broader region. Umberto noted that articles for that type of issue were not very hard to write; they would be 1-2 page summaries of activities of a company or institution.

Umberto followed up with the related broader matter of low interest in the journal by authors of scientific and technical papers. He indicated that only a handful of submissions (Jérémie gave an up to date count of 9) in the past 12 months. He suggested the approach to request that members of the Board submit articles and/or try to generate interest in their employees or students to submit. Another possibility voiced by Frank would be to drop the number of issues per year, but that would have its own problems with advertising etc. Umberto commented that a lower number of issues would mean an even longer period for submitted articles to be published, which would discourage authors. He indicated that having two regular issues per year plus the conference issue and a special interest issue (region or topic) was still a good paradigm but more contributions had to be encouraged. The editorial board would be looking at ideas to achieve this.

The September issue (annual CAA conference issue) will be the first to be produced entirely under Umberto's leadership as Editor in Chief, replacing Jérémie. Other changes in the editorial team include a new Deputy Editor, Romain Dumoulin, new Journal Manager, Cécile Le Cocq, and new Copyeditor, Olivier Valentin.

Jérémie pointed out several changes and developments underway with the Journal:

a) Potential change in publication process by becoming associated with a large publishing group; matter would be taken under advice by the new Editorial Advisory Committee.
b) Support for Digital Object Identifier (DOI) cataloguing of articles appearing in Canadian Acoustics, which will enable instant electronic referral. Annual fee would be US$275 moving forward for DOI listing of upcoming articles. Optional archival listing of existing articles back to the inception of the journal could be costed separately.
c) A publishing agreement with EBSCO was still not executed pending resolution of some matters with the CAA's official postal address; now potentially in question because of possible change in publication process.
d) An application for indexing of the Journal by the Emerging Sources Citation Index (ESCI) was rejected for technical reasons that the new Editor and Editorial Advisory Committee will consider countering.

### Secretary's report (Roberto Racca)

Roberto provided and reviewed the current tally of Association members and Canadian Acoustics subscribers as summarized in the table below, which shows by comparison the numbers for 2016 reported at the September and April meetings.

| **Category** | **Paid-up 2017 (as of 1 June)** | **Paid-up 2016 (as of 18 Sept)** | **Paid-up 2016 (as of 1 April)** |
|---|---|---|---|
| Regular member | 145 | 148 | 133 |
| Emeritus | 1 | 1 | 1 |
| Student | 34 | 33 | 18 |
| Sustaining subscriber | 28 | 26 | 26 |
| Indirect subscribers | | | |
| - Canada | 6 | 6 | 5 |
| - USA | 4 | 4 | 4 |
| - International | 4 | 4 | 3 |
| Direct subscribers | 1 | 3 | 3 |
| **Total** | **223** | **225** | **193** |

Roberto noted that the numbers now appeared to be stable which might indicate a more consistent cycle of renewals with fewer issues with the on-line system and PayPal payments. He noted the encouraging consistency of renewals for the sustaining subscribers, whom he made a point of supporting by personal contact and assistance in taking advantage of benefits such as direct on-line access for all employees to current contents of the Journal. Roberto pointed out that handling of paper-based renewals has been improved by a more streamlined forwarding of postal deliveries and that the Association had joined a fully electronic renewals notification and payment system implemented by one of the major subscription agencies, which would reduce processing delays that led in the past to journal delivery miscues.

Dalila recommended that the secretary and treasurer jointly monitor this new electronic agreement to ensure that all relevant notification of payments are received.

**Other business**

<u>PRUAC (Pacific Rim Underwater Acoustics Conference) support</u>

Roberto gave a quick background of the conference, that was last held in 2015 in Vladivostok; the 2018 event is to be held in Taipei and organizers asked for endorsement from the CAA with no financial obligations. A motion to endorse was made by Frank and seconded by Dalila; carried unanimously.

<u>2017 CAA annual conference</u>

Frank reported that the organizing of the Guelph meeting is on track and already on solid financial footing with good sponsorship commitment.

**Adjournment**

Motion to adjourn at 16:15 EDT by Mehrzad; seconded by Michael.

# REGISTER NOW !

## ACOUSTICS WEEK IN CANADA 2017 – GUELPH, ONTARIO

Your participation is invited for Acoustics Week in Canada 2017 to be held **October 11-13, 2017**. The Canadian Acoustical Association is inviting registrations for the conference hosted at the Delta Hotel and Conference Centre in Guelph Ontario.

This conference highlights the state of the art in acoustics by bringing together leaders, researchers and thinkers.

A **conference schedule** is being drafted and will be published on the web site http://awc.caa-aca.ca

Highlights include:

| | |
|---|---|
| Tuesday October 10 | Standards meetings |
| Wednesday October 11 | Keynote Speaker **Elliott Berger**:<br>Bang! Damage from impulse noise and the effectiveness of hearing protection |
| | **Reception** with evening Trade Show |
| Thursday October 12 | Keynote Speaker **John Bradley**:<br>A Rationale for a National Classroom Acoustics Standard |
| | **National Research Council workshop** about tools and data available to consultants and designers to address acoustic changes in the National Building Code |
| | **Trade Show** during daytime |
| | **Banquet** and Awards Ceremony |
| Friday October 13 | Keynote Speaker **Samir Ziada**:<br>Flow-Excited Acoustic Resonances |
| | Student Presentation Awards |

## Key Dates

- Submit a **two-page paper and register for the conference by August 1, 2017** for publication in the conference proceedings.
- **Register by September 11, 2017** for the lower early registration rates. Prices will go up.
- **Reserve rooms** from the block of reduced-rate hotel rooms **by September 11, 2017.**


# REGISTER NOW !

## Registration

| | Member | Non-member |
|---|---|---|
| Full conference registration | $ 495 | $ 615 |
| Student conference registration | $ 275 | $ 335 |

For further details please refer to the conference website. Rates go up after September 11, 2017.

## Venue and Accommodation

The conference will be held at Marriott's Delta Hotel and Conference Centre in Guelph.

Enjoy the convenience of staying at the conference venue. The hotel has set aside a block of rooms for the conference at a special rate of **$129 per night** for reservations made until September 11th. Claim the special rate by reserving at 519-780-3700 or 1-800-268-1133 and identifying the group Canadian Acoustical Association (Group Code CAN101017_001). Extend your stay and enjoy the local area at the same special rate. Guests have free access to the large Movati Athletic fitness centre next door.

## Exhibition

You will have opportunities to look for new products and try the latest in equipment from key suppliers to the acoustic community. Look for them at the conference exhibition for in-person and hands-on interaction!

Suppliers who have not yet reserved their spot should do so promptly, as spaces are filling up quickly. A few sponsorship opportunities are also still available. Contact the Exhibits and Sponsorship coordinator – Bernard Feder.

## Key Contacts:

Conference Chair: Peter VanDelden (conference@caa-aca.ca)

Technical Chair: Christian Giguère (technical-chair@caa-aca.ca)

Exhibits and Sponsorship: Bernard Feder (bfeder@hgcengineering.com)

Registration: Dalila Giusti (treasurer@caa-aca.ca)

Conference Web Site: Kyle Hellewell (kyle.hellewell@rwdi.com)

**http://awc.caa-aca.ca**

# INSCRIVEZ-VOUS DÈS MAINTENANT

## SEMAINE CANADIENNE DE L'ACOUSTIQUE 2017 - GUELPH, ONTARIO

Vous êtes invités à participer à la Semaine canadienne de l'acoustique 2017 qui aura lieu du **11 au 13 octobre 2017**. Vous pouvez dès maintenant vous inscrire au congrès qui se tiendra au à l'Hôtel et Centre de Congrès Delta Guelph en Ontario.

Ce congrès fera état des dernières connaissances en acoustique en réunissant des leaders, chercheurs et autres acousticiens.

Un **calendrier complet** du congrès est en voie de préparation et sera affiché sur le site http://awc.caa-aca.ca

Les points saillants:

| | |
|---|---|
| Mardi 10 octobre | Réunions sur les normes |
| Mercredi 11 octobre | Conférencier invité **Elliott Berger**:<br>Bang! Dommages causés par les bruits impulsifs et l'efficacité de la protection auditive |
| | **Réception de bienvenue** et lancement de l'Exposition technique en soirée |
| Jeudi 12 octobre | Conférencier invité **John Bradley**:<br>Vers une norme nationale en acoustique de salle de classe |
| | **Atelier du Conseil national de la recherche du Canada** sur les nouveaux outils et données disponibles aux consultants et concepteurs à tenir compte des derniers changements au Code national du bâtiment |
| | **Exposition technique** ouverte toute la journée |
| | **Banquet** et remise des prix |
| Vendredi 13 octobre | Conférencier invité **Samir Ziada:**<br>Résonances acoustiques excitées par les fluides |
| | Prix pour meilleures présentations étudiantes |

## Dates importantes

- **Soumettre votre article de 2 pages et vous inscrire au congrès d'ici le 1er août** afin de rencontrer les exigences de publication pour les actes du congrès.
- **Vous inscrire au congrès d'ici le 11 septembre 2017** pour bénéficier du tarif hâtif.
- **Réservez votre chambre d'ici le 11 septembre 2017** pour bénéficier du bloc de chambres à tarif réduit.

# INSCRIVEZ-VOUS DÈS MAINTENANT

## SEMAINE CANADIENNE DE L'ACOUSTIQUE 2017 - GUELPH, ONTARIO

### Inscription

| | Membre | Non-membre |
|---|---|---|
| Inscription complète (3 jours) | $ 495 | $ 615 |
| Inscription pour les étudiants (3 jours) | $ 275 | $ 335 |

Pour plus de renseignements, veuillez consulter le site internet du congrès. Les taux augmentent après le 11 septembre 2017.

### Lieu et logement

Le congrès se tiendra à L'Hôtel Delta Guelph et Centre de congrès par Marriott.

Profitez-en pour être directement sur le site du congrès. L'hôtel a mis à disposition un bloc de chambres pour les participants au tarif réduit de **$129 par nuit** pour les réservations effectuées jusqu'au 11 septembre. Réclamez le tarif réduit en réservant au 519-780-3700 ou au 1-800-268-1133 et indiquez que vous faites partie du groupe « Association canadienne d'acoustique » (Code de groupe CAN101017_001). Prolongez votre séjour et profitez de l'occasion pour découvrir la région au même tarif réduit. Les clients de l'hôtel ont accès gratuit au centre d'entrainement « Movati Athletic » situé juste à côté.

### Exposition technique

Les fabricants et fournisseurs qui n'ont pas encore réservé leur table sont priés de le faire dans les plus brefs délais car les places sont limitées. Quelques commandites sont encore disponibles. Veuillez communiquer avec le coordonnateur des exposants et commanditaires – Bernard Feder.

### Personnes contacts:

Président du congrès: Peter VanDelden (conference@caa-aca.ca)

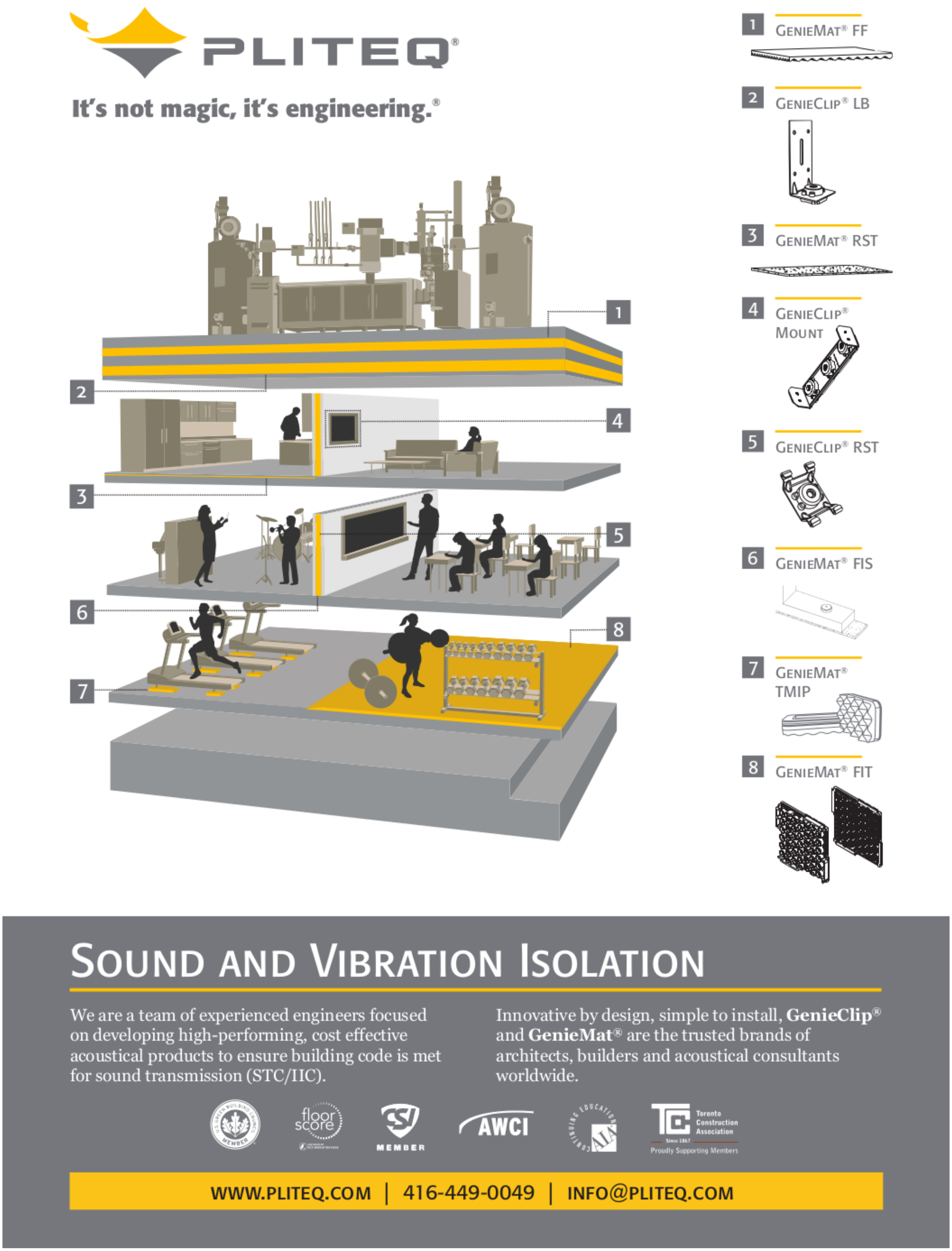
Directeur scientifique: Christian Giguère (cgiguere@uottawa.ca)

Exposants et commanditaires: Bernard Feder (bfeder@hgcengineering.com)

Inscription: Dalila Giusti (treasurer@caa-aca.ca)

Site internet du congrès: Kyle Hellewell (kyle.hellewell@rwdi.com)

**http://awc.caa-aca.ca**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# 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*

**CAA is now social!**

Canadian Acoustical Association is moving to the social media!

Find us on social media: - Twitter: CanAcoustical - Facebook: facebook.com/canadianacousticalassociation

*December 14th 2015*

**Acoustics exhibition at the Musée de la Nature et des Sciences in Sherbrooke (QC)**

The Musée de la Nature et des Sciences in Sherbrooke will feature an exhibition on Acoustics.

A group from the Université de Sherbrooke led by Dr. Olivier Robin approached the CAA with a request for support of a new exhibition they are planning entitled 'Musée de la Nature et des Sciences'. This exhibit is intended to disseminate knowledge about acoustics to the general public, including information about sources of sound, sound propagation and sound reception. This initiative aligns well with our interest as an Association in education and outreach, as well as with activities being planned by the International Commission of Acoustics (of which the CAA is a member) for the proposed UNESCO "Year of sound" in 2019 (http://www.sound2019.org). The board agreed to commit up to $2000 of financial support along with promotional assistance. We expect that the group will be able to present elements of the exhibition in the journal and at an upcoming Acoustics Week in Canada.

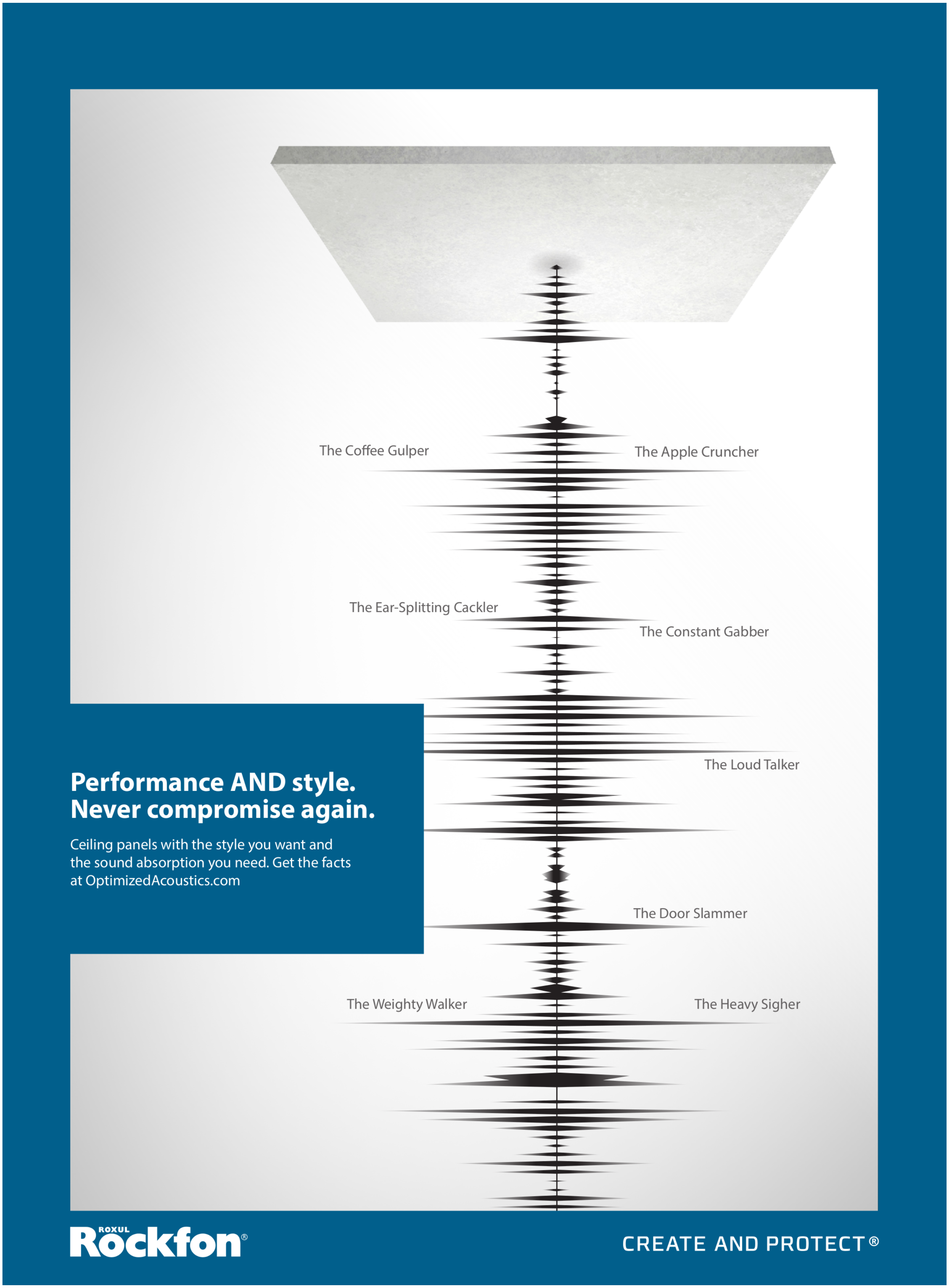
*November 5th 2016*

**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*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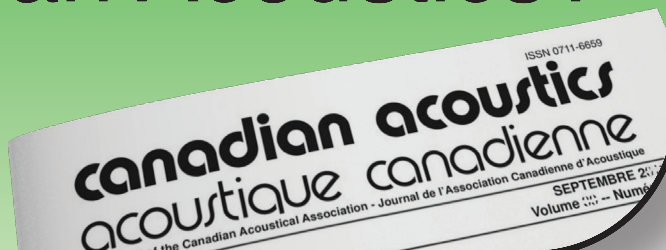
# Why publish in Canadian Acoustics?




## 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

# Canadian Acoustical Association
# Association canadienne d'acoustique

## Application for Membership

CAA membership is open to all individuals who have an interest in acoustics. Annual dues total $100.00 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 $100.00 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*

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**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 $ 100.00
[ ] CAA Student Membership $ 50.00

Corporate Subscriptions (4 issues/yr)
[ ] $100 including mailing in Canada
[ ] $108 including mailing to USA,
[ ] $115 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, The Canadian Acoustical Association, PO Box 74068, Ottawa, Ontario, K1M 2H9, 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 100.00$ pour les membres individuels, et de 50.00$ pour les étudiants. Tous les membres reçoivent ***l'Acoustique Canadienne***, la revue de l'association. Les nouveaux abonnements reçus avant le 31 août s'appliquent à l'année courante et incluent les anciens numéros (non-épuisés) de ***l'Acoustique Canadienne*** de cette année. Les nouveaux abonnements reçus après le 31 août s'appliquent à l'année suivante.

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

Les abonnements pour la revue ***Acoustique Canadienne*** sont disponibles pour les compagnies et autres établissements au coût annuel de 100.00$. Des compagnies et établissements préfèrent souvent la cotisation de membre bienfaiteur, de 475.00$ par année, pour assister financièrement l'ACA. La liste des membres bienfaiteurs est publiée dans chaque issue de la revue ***Acoustique Canadienne***. Les nouveaux abonnements reçus avant le 31 août s'appliquent à l'année courante et incluent les anciens numéros (non-épuisés) de ***l'Acoustique Canadienne*** de cette année. Les nouveaux abonnements reçus après le 31 août s'appliquent à l'année suivante.

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

***Pour correspondance administrative et financière:***

Nom / Organisation ______________________________

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Nom / Organisation ______________________________

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**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
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