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Editor's note: the sound of our country
Éditorial : le son de notre pays

The sound of our country

Dear reader, welcome to the second issue of our 46th year. I would like to open this editorial with my warm thank to three guest editors, that have helped me in attracting a series of contributions from Vancouver (and the province of British Columbia at large): please, join me in acknowledging Sasha Brown, Maureen Connelly, and Roberto Racca. This current year "regional" section JCAA aims to offer an opportunity to individuals, groups, and companies located around major Canadian cities, to showcase their activity. Although, the contributions for this section of this issue have been limited, we hope you will read three successful stories of three large consulting firms.
The issue is also enriched by technical research papers accepted for the categories of aeroacoustics, architectural acoustics, and engineering acoustics/noise control.

This issue contains a rich series of contributions. I personally thank all the authors and I hope we will attract more good papers in the future too.

We have also received new interesting articles for the Practitioners corner; it was late for this issue, but and next issue we will surely publish them.

Moving forwards, I am glad to stress that the 176th Meeting of the Acoustical Society of America (ASA) will be held jointly with the Acoustics Week in Canada 2018 of the Canadian Acoustical Association (CAA) in Victoria, BC, Canada, on 5-9 November 2018. We hope that you will consider presenting a paper or attending the meeting to participate in the exchange of ideas and the latest research developments in acoustics and to meet with your colleagues in the ASA. All members of the Canadian Acoustical Association are invited to submit a 2-pages conference proceedings paper that will be published in Canadian Acoustics in December 2018. Authors are encouraged to submit directly by November 1st directly in Canadian Acoustics journal.

I wish you a pleasant reading.

Umberto Berardi,
Editor-in-chief.

Le son de notre pays

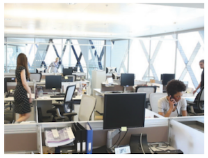
Chère lectrice, cher lecteur, bienvenue au deuxième numéro de notre 46ème année. Je voudrais commencer cet éditorial par remercier chaleureusement les trois rédacteurs invités, qui m'ont aidé à attirer plusieurs contributions de Vancouver (et de l'ensemble de la Colombie-Britannique): joignez-vous à moi pour remercier Sasha Brown, Maureen Connelly, et Roberto Racca. La section «régionale» de la revue vise à offrir une opportunité aux individus, groupes et entreprises situés autour des grandes villes canadiennes, de présenter leur activité. Bien que les contributions pour la section de ce numéro aient été limitées, nous espérons que vous lirez ces trois histoires réussies de trois grandes firmes de consultants.

Le numéro est également enrichi par des articles de recherche techniques dans les catégories de l'aéro-acoustique, de l'acoustique architecturale et de l'ingénierie acoustique / contrôle du bruit. Ce numéro contient une riche série de contributions. Je remercie personnellement tous les auteurs et j'espère que nous attirerons encore plus d'articles de qualité à l'avenir. Nous avons également reçu de nouveaux articles intéressants pour le Coin des praticiens; il était tard pour ce numéro, mais ils seront assurément publiés dans le prochain.

En allant de l'avant, je suis heureux de souligner que la 176e réunion de la Société américaine d'acoustique (ASA) se tiendra conjointement avec la Semaine canadienne d'acoustique 2018 de l'Association Canadienne d'Acoustique (ACA) à Victoria, C.B., Canada. Nous espérons que vous envisagerez de présenter un article ou d'assister à la conférence pour participer aux échanges d'idées et des derniers développements de la recherche en acoustique et de rencontrer vos collègues de l'ASA. Tous les membres de l'Association Canadienne d'Acoustique sont invités à soumettre un article de conférence de 2 pages qui sera publié dans la revue Acoustique Canadienne de décembre 2018. Les auteurs sont encouragés à soumettre d'ici le 1er novembre directement via la revue.

Je vous souhaite une agréable lecture.

Umberto Berardi,
Rédacteur en chef

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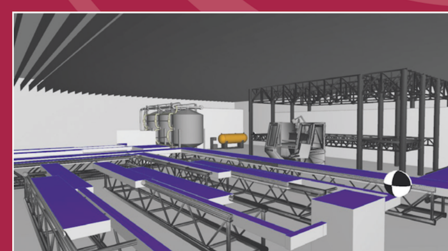

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BKL – ENGINEERING A BETTER SOUNDING WORLD

Mark Bliss*[1], Paul Marks†[1] et Tiberiu Spulber‡[1]
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Résumé

BKL Consultants Ltd. est une firme de génie-conseil indépendante qui travaille exclusivement dans le domaine de l'ingénierie acoustique depuis 1966. L'équipe travaille avec constructeurs, propriétaires, architectes, consultants, gouvernements, gestionnaires et autres professionnels afin d'offrir des solutions pratiques dans le domaine de l'acoustique architecturale et environnementale et du contrôle du bruit et des vibrations. Suivant les plus récentes normes et directives, BKL fournit des systèmes de surveillance, des diagnostic, des prévisions, des évaluations, de la gestion et des conseils en acoustique et en contrôle du bruit et des vibrations, en utilisant une gamme d'instruments spécialisés—incluant plus de 25 moniteurs de son et de vibrations—et d'installations informatiques.

Mots clefs : acoustique, bruit, vibration, consultation, services de conception, proposition de mesures palliatives, contrôle du bruit, isolation vibratoire, acoustique architecturale et du bâtiment, évaluation du bruit et des vibrations dans l'environnement, LEED

Abstract

BKL Consultants Ltd. is an employee-owned independent consulting firm that has worked exclusively in the field of acoustical engineering since 1966. BKL's team works with builders, owners, architects, consultants, governments, managers, and other professionals to provide practical architectural and environmental acoustics, noise, and vibration solutions. BKL provides sound and vibration monitoring, diagnostics, predictions, assessments, management, and mitigation advice to the latest standards and guidelines, using an array of specialized instrumentation—including more than 25 sound and vibration monitors—and computing facilities.

Keywords: acoustics, noise, vibration, consulting, design services, remedial advice, noise control, vibration isolation, architectural acoustics, building acoustics, environmental noise and vibration assessments, LEED

1 Introduction

BKL has been solving acoustical problems for over 50 years. It all started when company co-founder Ken Barron designed a Leo Beranek–inspired muffler to hush the gas-fired ovens at Dino's Pizza on Broadway in Vancouver. Today BKL serves clients on projects large and small across every sector—residential, transportation, healthcare, marine, cultural, industrial, education, and more.

BKL provides independent reviews and advice for mitigating sound and vibration-related risks. The company's services mainly relate to the design or renovation of buildings and infrastructure, delivered through P3, design-build or traditional methods; environmental assessments of major infrastructure projects; and addressing specific acoustical problems inside buildings or in surrounding environments.

Continue reading to learn more about BKL's expert contributions to three recent highlight projects in British Columbia: the Jim Pattison Outpatient Care and Surgery Centre, Port Mann / Highway 1 Improvement Project, and Telus Garden.

Figure 1: Jim Pattison Outpatient Care and Surgery Centre

2 Jim Pattison Outpatient Care and Surgery Centre

BKL joined the BC Healthcare Solutions team for this P3 project, a new four-storey, 188,000 square-foot facility. The centre is LEED Gold certified, and is the first in BC to bring together more than 50 services and programs, including day surgery, exclusively for outpatients.

First, BKL established a comprehensive list of criteria to guide all acoustics-related decisions. Next, BKL evaluated the impacts of traffic noise on the building and made recommendations to upgrade the building envelope's sound-isolation performance.

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Consulting on the interior acoustics, BKL reviewed interior sound isolation requirements, partition construction, and door and seal selection. The team calculated reverberation time in the main lobby, waiting areas, and other spaces, and recommended acoustical treatments to control reverberation and reduce overall noise levels.

To control noise from mechanical and electrical services, BKL recommended silencers for air-handling units, which delivered excellent results in controlling background noise from mechanical equipment in occupied spaces. In addition, BKL assessed noise from the emergency generators and made recommendations for sound isolation enclosures to mitigate the impacts of generator noise outside. The team analyzed vibration levels due to mechanical and electrical equipment and recommended vibration isolation that, when installed, limited floor vibration to levels lower than those required by the MRI manufacturer.

For this project, BKL took a hands-on approach and ensured the highest standards of quality were met. The team attended design meetings, reviewed shop drawings and carried out site visits to inspect and confirm the correct implementation of the acoustical recommendations. Upon completion, BKL tested noise, vibration, reverberation, sound isolation, confirmed compliance with the project criteria, and obtained LEED point IDc1 as a result.

Figure 2: Port Mann / Highway 1 Improvement Project

3 Port Mann / Highway 1 Improvement Project

Highway 1 in Metro Vancouver is BC's busiest transportation corridor. The $3.3 billion Port Mann / Highway 1 Project involved widening 37 kilometres of highway, and reconstructing 15 interchanges. The project also included the construction of the new Fraser Heights Connector linking Highway 1 with South Fraser Perimeter Road and the Golden Ears Crossing.

During the bid process, the design-build contractor asked BKL to join their team to carry out noise impact and mitigation assessments. BKL provided all acoustics-related information during the bid process and the pre-construction, construction, and post-construction phases.

To assess noise impacts and noise mitigation against the Ministry of Transportation and Infrastructure's policy and the design-build contract requirements, BKL developed a 3D noise model of existing and future road traffic noise for the entire project. This included over 450 lane kilometres of roadways and more than 1,000 residences and schools.

BKL monitored construction noise and vibration, and consulted on the design of more than 35 noise walls. BKL also monitored post-construction noise to confirm compliance with project criteria, and tested noise barrier insertion loss to ANSI S12.8.

Figure 3: Telus Garden

4 Telus Garden

Telus Garden features two main buildings: a 22-storey office tower and a 47-storey residential tower. When it opened in 2015, the office tower was the first building in Vancouver to earn LEED Platinum certification. The residential tower, opened in 2016, is certified LEED Gold.

For Telus Garden, BKL's project engineers provided comprehensive acoustical engineering services, establishing acoustical design guidelines for ambient noise (HVAC, traffic), sound isolation (speech privacy and transmission loss), and room acoustics (reverberation time) in open-plan and cellular office spaces, meeting rooms, and common areas. The team reviewed mechanical and electrical equipment, and developed noise control and vibration isolation for AHUs, elevators, generators, and the cooling tower. BKL also reviewed shop drawings, conducted testing in a mock-up suite, and provided acoustical partition design and STC reviews.

BKL also designed the internal acoustics for the television studio and support space that occupies almost 10,000 square feet of the office tower's second floor. Addressing the needs of acoustically disparate spaces including an encoding room, an audio suite, editing suites, meeting areas, and an open-plan office with 18 workstations, BKL worked with the interior designers to optimize sound isolation and room acoustics for noise-sensitive areas, like the studio and audio suite, while ensuring an appropriate level of speech privacy among individual workspaces in the open-plan office.

5 Conclusion

These three projects are examples of BKL's recent success in sharing its expertise in acoustics and vibration while contributing to projects that are important to British Columbians. If you're interested in learning more about BKL's services or project portfolio, visit www.bkl.ca/what-we-do

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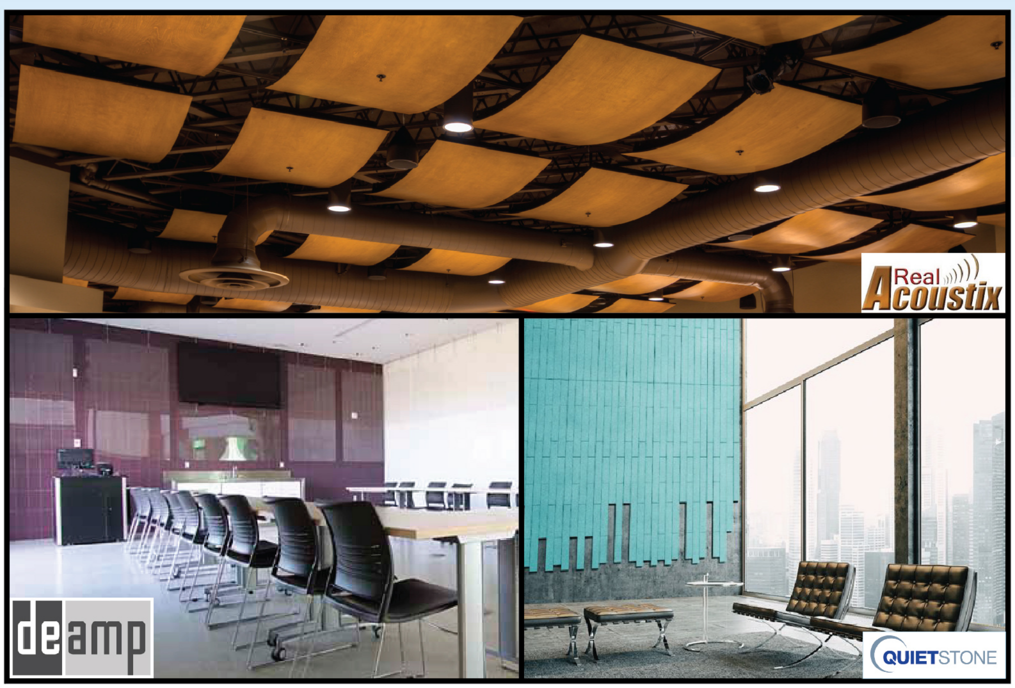

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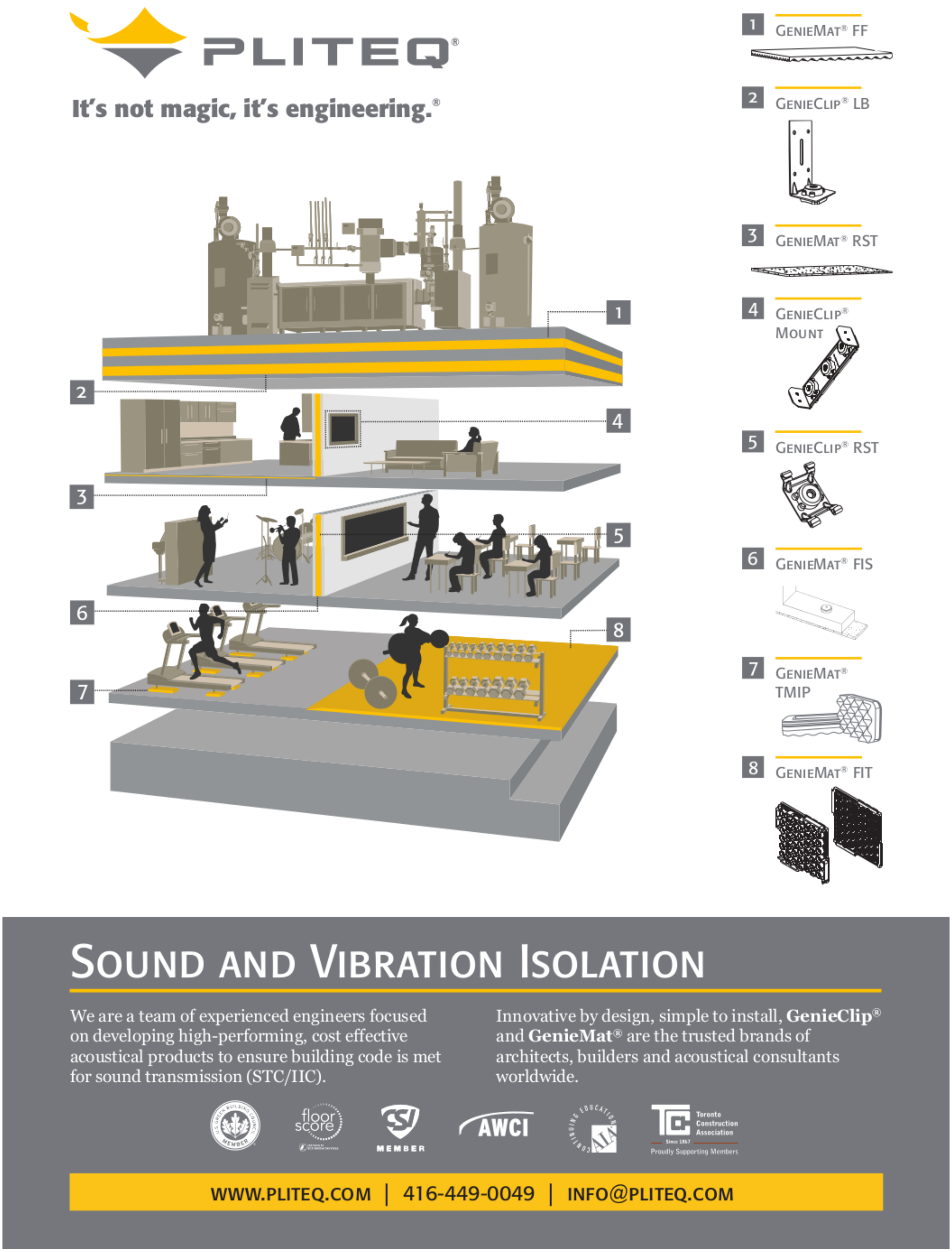
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Résumé

SLR est un bureau-conseil multidisciplinaire, fournissant des services environnementaux spécialistes à travers un réseau de bureaux aux États-Unis, Canada, Europe, Australie-Pacifique et en Afrique. Notre équipe mondiale en acoustique est le résultat de l’acquisition de deux entreprises, chacune avec plus de 35 ans d’histoire, l’ancien Heggies en Australie et HFP Acoustical Consulting en Amérique du Nord. Nous employons maintenant environ 90 conseillers spécialisés en acoustique à travers le monde. Une grande force de notre équipe est la capacité de tirer parti de notre expérience globale. Notre équipe d’acoustique Canadienne est au service des clients partout au Canada, aux États-Unis et à l’étranger.

Mots clefs : acoustique, bruit, vibration, environnementale, ferroviaire, transport, global, faune

Abstract

SLR is a multi-disciplinary consultancy providing specialist environmental services through a network of offices in the USA, Canada, Europe, Australia-Pacific, and Africa. Our global acoustic team came on board via the acquisition of two companies each with over 35 years of history, the former Heggies in Australia and HFP Acoustical Consulting in North America. Worldwide, we now employ around 90 specialist acoustic consultants. A great strength of our team is the ability to draw on our global experience. Our Canadian acoustics team services clients across Canada, the US and internationally.

Keywords: acoustics, noise, vibration, environmental, rail, transportation, global, wildlife

1 Introduction to SLR

SLR Consulting (Canada) Ltd. is a multi-disciplinary consultancy providing worldwide environmental sciences, engineering expertise and high-value advisory services through a network of offices supporting approximately 1200 environmental professional staff globally. With 17 offices and more than 210 employees in Canada, SLR is recognized as a leader in the provision of Environmental Services.

SLR’s global acoustic team came on board predominantly through the acquisition of two companies each with over 35 years of history providing acoustic services, the former Heggies in Australia (acquired in 2010) and HFP Acoustical Consulting in North America (acquired in 2014). Worldwide, we now employ around 90 specialist acoustic consultants, with almost a third of these located across North America. Our Canadian locations offering acoustic services are Vancouver and Calgary, supported when needed by field staff from our distributed offices shown in Figure 1.

Our global experience gives us a unique perspective on environmental noise in British Columbia. This paper provides some examples of situations where we have been able to apply experience of best environmental noise practices elsewhere to BC, Canadian and North American projects.

Figure 1: SLR’s Canadian office locations.

2 Overview of acoustics services

Our team’s expertise can be employed in a variety of ways, from providing guidance on architectural acoustics to carrying out full environmental noise impact assessments for major industrial facilities and infrastructure projects. We offer advice on mitigating measures to balance potential environmental issues with the need for development. SLR also undertakes design and implementation of noise monitoring schemes to verify site compliance.

- Industrial facility assessments and noise control design
- Transportation noise and vibration assessments
- Architectural acoustical design

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- PA speech intelligibility/audibility
- Underwater acoustics assessments
- Structural dynamics investigations and mitigation
- Mining / quarrying noise and blasting assessments
- Oil refinery and gas plant noise control
- Construction noise and vibration
- Wind farm noise assessments
- Environmental noise compliance surveys

3 Selected examples

3.1 Railway noise and vibration

In Canada, rail noise is federally regulated and noise complaints are adjudicated by the Canadian Transport Agency. Legislation requires freight companies to transport goods, and effectively means rail operators are not responsible for addressing any increases in noise and vibration arising from increases in rail traffic. For new projects, Environmental Impact Assessments during project planning stages are often the only point at which rail noise or vibration impacts are assessed.

Elsewhere in the world more stringent requirements for assessment and mitigation of noise from both rail freight and rail transit are common. These requirements have driven considerable research and development in Europe and elsewhere into best practices for rail noise and vibration control and railway noise policy. SLR's Vancouver team has internationally recognized experience of current best practices in areas including:

- Rail grinding and roughness implications for noise
- Transit source noise and vibration control
- Curve squeal noise mitigation
- Rail dampers for in-car and guideway noise mitigation
- Mitigation for new developments near rail
- Noise from freight rail yards
- Freight locomotive noise

3.2 Noise management plans

Existing mines, ports and industrial sites in British Columbia sometimes find they are increasingly receiving noise complaints. In some cases, they may have expanded or made changes to their operations over time resulting in increased noise emissions. In other cases, new residential developments increase the population near the site. Or, sometimes neighbours' expectations can change over time.

For many industries and jurisdictions there are no clearly defined noise limits. Local bylaws are often "nuisance" based and require a subjective interpretation of what noise is reasonable.

In this situation, SLR recommends development of a facility specific noise management plan. This plan can be developed in consultation with local government stakeholders and provincial regulators to define measurable noise goals, acceptable levels at different times of day, and procedures for ongoing monitoring. A noise management plan eliminates subjectivity, provides clearly defined responsibilities and sets a pathway to demonstrate best practices for balancing noise emissions with impacts to neighbours. Recent projects involving noise management plans include:

- Sand and gravel mine noise management plans
- Blast noise management plans
- Artic drilling noise management
- A coordinated plan to manage cumulative noise from multiple adjacent industrial facilities

Figure 2: Arctic noise monitoring.

3.3 Noise and wildlife (land and underwater)

The effect of noise from human activities on fauna is increasingly a subject of concern in the community when proposing developments such as new infrastructure, mines or industrial developments. This is particularly the case for projects along the BC coastline or in remote areas.

There are often no Canadian policies or accepted guidelines for noise levels or thresholds that may have an adverse effect on wildlife. The lack of policies is understandable when considering that responses to noise disturbance cannot be generalized across species.

In assessing noise impacts to wildlife, SLR combines acoustics and ecology expertise to determine appropriate precautionary parameters for each situation and to identify extents of habitat that may be affected. Assessments undertaken recently have included:

- Underwater overpressure and vibration due to blasting in and near fish-bearing inland water bodies
- Quarry noise impacts to birds
- Underwater noise impacts to marine mammals and fish
- Linear infrastructure construction noise and airblast impacts to bats and bumblebees

For more information

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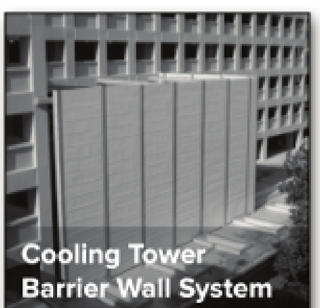

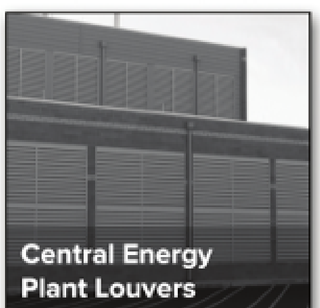

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RWDI: ACOUSTICS, NOISE, AND VIBRATION CONSULTANTS

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Résumé

RWDI est une firme de consultation multidisciplinaire canadienne ayant des bureaux partout au Canada (y compris des bureaux à Vancouver et à Victoria) qui offrent des services de consultation en acoustique. Dans cet article, nous présentons le large éventail d’expériences, de projets et de services réalisés par RWDI. La majeure partie de notre expertise concerne l'acoustique environnementale et architecturale et le contrôle du bruit et des vibrations mais inclue également des projets d’envergure dans le domaine des constructions en bois massif, du contrôle de la conformité aux codes et aux règlements, ainsi que l'élaboration de politiques. L’ensemble des projets entrepris par RWDI contribue à supporter diverses communautés grâce à notre soutien technique en conception et en construction.

Mots clefs : acoustique, bruit, vibration, architecture, environnement, bois massif, codes, règlements, politique, mesure, essai, conception

Abstract

RWDI is a Canadian multi-disciplinary consulting firm with offices across Canada that provide acoustic consulting services, including offices in Vancouver and Victoria. The broad range of projects, services, and experience are highlighted. The core of our work is in environmental and architectural acoustics, noise, and vibration projects, but is further detailed in highlights of our work in mass timber construction, code and bylaw testing, and policy development. It is concluded that the work we do ultimately contributes to helping improve our communities through technical support to the design and construction community.

Keywords: acoustics, noise, vibration, architecture, environment, mass timber, codes, bylaws, policy, measure, test, design

1 Introduction

RWDI is a Canadian owned and operated specialty consulting firm that began in 1972 as a snow loading consultancy out of Guelph, ON, and has expanded to include many specialty science and engineering consulting services with offices across Canada and around the world. In 2013 the Vancouver office began a focused growth in acoustics with the hire of Steve Meszaros (previously with RWDI in Guelph from 2000-2009). RWDI then acquired Dan Lyzun & Associates Ltd. of Vancouver (est. 2002) in 2014, and Wakefield Acoustics Ltd. of Victoria (est. 1988) in 2016.

Since its inception, RWDI’s BC acoustics team has had continuous growth and is supported by an even larger team that includes staff in RWDI’s Calgary, Toronto, and Guelph offices, as well as staff as far afield as Denmark.

The acoustics team is a strong support to RWDI’s promise of being ‘exceptional without exception’ while also striving to be the consultant of choice within our local communities. RWDI’s influence on our communities reaches far beyond the walls of our offices, and we feel a sense of pride when living, working and moving within the peace, quiet and enjoyment of spaces and places we’ve worked hard to help create. The subsequent sections are a summary of some of the work being done by our team.

2 Environmental noise and vibration

Major infrastructure projects (highway, transit, civil and energy works, marine ports, airports), industrial plants, commercial developments and even recreational facilities can create significant noise and vibration impacts at neighbouring residential and other noise-sensitive land uses. Such impacts may occur during both the construction and ongoing operation of these facilities.

Federal, Provincial, Regional and Municipal requirements dictate that such impacts should be assessed and mitigated. RWDI conducts such assessments independently in situations where noise and/or vibration are the primary concerns (such as facility construction or operations), or commonly as part of a multi-disciplinary team when a proposed project is subjected to a formalized environmental, social, and economic review process.

In carrying out such assessments, RWDI would typically conduct field measurements of pre-project (baseline) noise and/or vibration levels in potentially-affected communities. A large inventory of precision instrumentation and state-of-the-art modelling software is at

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our disposal to assess existing environments and predict future noise and vibration levels at locations of interest. This allows us to provide mitigation solutions that are customised to each project's specific requirements.

The experience of the RWDI team includes many major infrastructure projects in southwestern BC over the past two decades. Some of these include the Sea to Sky Highway Improvement Project, the Port Mann/Hwy 1 project, and the George Massey Tunnel Replacement Project. Currently, RWDI is involved in assessing community noise and vibration levels for the Roberts Bank Terminal 2 Project, Trans Mountain Expansion Project, Site C Clean Energy Project, Capital Regional District Sewage Treatment Project, Pattullo Bridge Replacement Project and the Hwy 1/Mountain Highway Interchange Project.

Figure 1: Measuring noise from the Pattullo bridge

3 Architectural acoustics and vibration

Architectural acoustics and vibration work deals with buildings of all type and scale with a broad range of issues that are interconnected. This section provides insight into some of the acoustics and vibration work done for building design with a focus on the intended use and the experience of the end user in mind.

Noise control and zoning bylaws drive most requirements for noise ingress controls for residential land-uses situated in noisy areas. The interior sound isolation requirements for party walls are dictated by the BC Building Code.

Mechanical noise emissions to the exterior are also dictated by local bylaws but internal mechanical noise is non-regulated. In these cases we look to various published guidelines such as those from ANSI, ASHRAE etc., as well as best practice guidelines from other jurisdictions as appropriate.

Room acoustics projects range from board rooms and offices (private and open plan), to classrooms, music rooms, studios, performance spaces, pools, arenas, and more. These are always rewarding projects to work on as they allow for a degree of artistic creativity. RWDI has strong relationships with local and international architects, where we collaborate to specify surface finishes and geometry for spaces that are both acoustically effective and aesthetically pleasing.

Figure 2: Room acoustics measurements in a concert hall with an omni-directional source

Only residential partitions are mandated by the BC Building Code; however, designing for indoor noise targets and creating comfortable working spaces requires in-depth knowledge of sound transmission control. Space constraints of high density living creates non-compatible adjacencies that are unavoidable. Some such complex adjacencies we've worked on include a generator room adjacent to a bedroom, music rehearsal spaces adjacent to classrooms, noisy MRI equipment adjacent to a lecture theatre, and a noisy roadway adjacent to a hearing testing facility, to name a few. Creating some of the quietest spaces and/or attenuating some of the loudest sounds requires ingenuity when combined with normal building design constraints.

Isolating machinery from building structures is paramount in all cases to minimize transmission of mechanical vibration into a building's structure where it can then efficiently propagate and radiate as structure-borne noise throughout the building. Elevators are a common source that requires careful isolation to avoid complaints from penthouse occupants.

Impact noise in residential spaces, is a common complaint in multi-family buildings. This is exacerbated by the trend to replace carpet with hard flooring. Like impact noises from gyms (weights dropping and treadmills), these are problems that are difficult to address without significant upgrades.

Vibration in buildings can be problematic where vibrations could adversely affect occupants or vibration sensitive equipment. RWDI models vibration from both internal (e.g., mechanical, footfall) and external (e.g., road, rail) sources and their propagation through soil, soil-structure, and building structures. Our experience in this type of work has enabled us to help design structures that meet very stringent research and occupant comfort requirements.

Wind-induced noise or aeroacoustics is a specialty service of RWDI, combining our wind engineering, climate statistics, fluid dynamics, and acoustics abilities. This service focuses on exterior building elements and design details that have the potential for creating noise when the wind blows. Desktop reviews lead to identification and risk ratings of various elements. Recommendations for design changes are provided for those elements at greater risk of making noise in the wind, and critical elements are tested in the wind tunnel to quantify their noise generating potential and to test the performance of solutions.

None of the above services stand alone. In combination they provide spaces that are comfortable, functional, and meet the needs of their occupants. We have many notable and unique projects that include a parkade rooftop daycare, the tallest wood structure building in the world, world-class research facilities, and many recreational, school, residential, healthcare, institutional, and commercial facilities.

4 Mass timber construction

RWDI has been involved in many projects using Nail Laminated Timber (NTL), Cross Laminated Timber (CLT) and hybrid constructions; the most notable being the "tallest wood frame construction building in the world", the Brock Commons student residence at UBC.

Architects are increasingly opting to use exposed wood finishes as driven by the "wood first" provincial mandate. CLT and NLT provide attractive options for exposing the structural members, which reduces the need for and cost of furring walls and finishes. CLT walls are also becoming increasingly common on projects in the Lower Mainland. However, wood structural elements such as NLT and CLT, while they may be constructed to have the strength and durability akin to concrete, have markedly different acoustic properties.

Airborne noise isolation

Airborne sound isolation in CLT walls is similar to basic insulated-cavity wall construction if there is at least one side of the CLT covered with furring and a layer of gypsum wallboard, or similar. This allows the other side to remain exposed in areas that do not require high levels of sound insulation. However, in order to meet building code requirements (currently STC 50 for party walls) either both sides of the CLT need to be covered in furring walls, or one side needs to be structurally isolated from the CLT. In such cases, exposed CLT does not seem to be the prudent choice. For CLT ceilings, the underside of the CLT should generally be covered to meet code requirements. As such, exposed CLT ceilings are only typically seen in penthouse units where there are no occupants or noisy equipment above.

Impact noise isolation

The driving factor in the construction of party floor separations using mass timber tends to be impact noise isolation. Mass timber floors provide poor impact noise isolation when compared to their concrete counterparts. Typically, an insulated suspended acoustic barrier ceiling (e.g., GWB) is required below with a high quality acoustic underlay or carpet above to reduce impact noise to reasonable levels. Even given careful consideration, however, it is still difficult to achieve impact noise isolation levels that are acceptable to occupants of multi-family residential units without the use of a concrete topping to increase the total mass of the floor assembly. Encouragingly, we are seeing increased innovation on the part of architects and developers to provide acoustically effective construction without foregoing the benefits of mass timber construction.

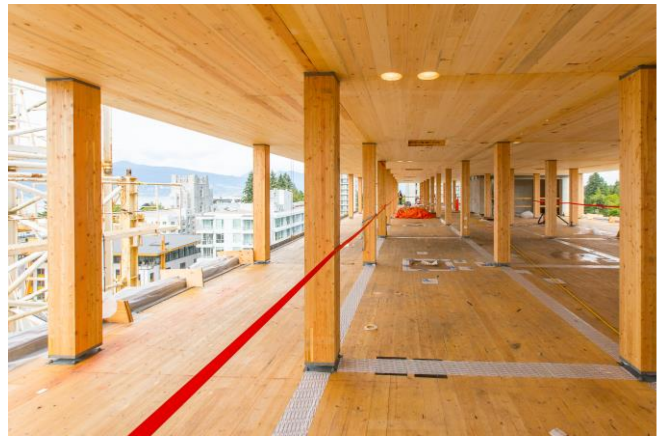

Figure 3: Brock commons, before wood was covered

5 Codes and bylaw testing

Building code testing (ASTC, AIIC)

RWDI provides acoustical testing services to validate building designs in terms of the sound transmission class (STC) of party walls and floors, the impact insulation class (IIC) of party floors, and to verify that exterior noise emissions from building services will not exceed municipal noise bylaw limits.

In BC, the building code currently requires that party wall partitions within multi-family dwellings provide a minimum of STC 50. To verify that the air-borne sound insulation provided by partitions meet this requirement, RWDI conducts STC testing in accordance with ASTM E336 "Test Method for Measurement of Airborne Sound Insulation in Buildings". An STC test conducted "in-situ" (as opposed to within a laboratory) is referred to as an apparent sound transmission class (ASTC) test. Unlike laboratory-tested STC ratings, ASTC ratings include secondary sound transmission paths due to sound flanking along common floors, ceilings and side walls. In cases where the ASTC is deficient, RWDI works with the owners to find solutions that will increase the air-borne sound insulation provided by the partition to acceptable levels.

While the B.C. Building Code does not include a requirement for party floor IIC ratings, it does recommend that bare floors (tested without carpet) should achieve IIC 55. RWDI conducts IIC tests in accordance with ASTM E1007 "Standard Test Method for Field Measurement of Tapping Machine Impact Sound Transmission Through Floor-Ceiling Assemblies and Associated Support Structures". Similar to the case of STC field-tests, measured IIC performance in buildings is referred to as the apparent impact insulation class (AIIC). The test procedure is similar to that for an ASTC test, with the primary difference being that the sound source is a standardized tapping machine, which uses metal cylinders to rhythmically strike the surface of the floor being tested.

In addition to providing AIIC test services, RWDI provides recommendations to architects and builders to maximize the AIIC of party floors. These recommendations often include evaluation of acoustical underlayment for use beneath finished floors.

Both ASTC and AIIC testing have been done on numerous residential projects in early construction to verify that a design meets Code, in post-construction to assess final performance, and in litigation cases where occupants are unsatisfied with the performance of their residence.

Noise bylaw testing

Certain municipalities in Canada, including the City of Victoria and those in the Lower Mainland, have noise bylaws that include quantitative noise level limits. Many of these municipal bylaws divide the city into “noise districts” which have different allowable noise levels based on land use.

RWDI is often requested to evaluate compliance with noise bylaws. Our experience with measurement, modelling, and design help us with interpretation of bylaws and in solving noise problems in an efficient and effective way.

6 Policy development

As a result of RWDI’s extensive experience in the area of environmental noise and acoustics, RWDI has been instrumental in the development of noise policies and guidelines widely used within BC. RWDI has been tasked by various governing and regulatory bodies such as the B.C. Ministry of Transportation and Infrastructure (MoTI), B.C. Ministry of Energy, Mines and Petroleum Resources (MoEMPR), Vancouver Fraser Port Authority (VFPA), and the City of Vancouver (CoV) to develop or update noise policies. These policies reflect local conditions while generally aligning with national and/or international noise control objectives. RWDI’s experience includes the development of the following:

- Policy for Assessing and Mitigating Noise Impacts from New and Upgraded Numbered Highways (MoTI, 2014)
- Best Practice for Wind Power Project Acoustic Assessment (MoEMPR, 2012)
- Project & Environmental Review: Guidelines – Environmental Noise Assessment (VFPA, 2015)
- Soundsmart: City of Vancouver Noise Control Manual (CoV, 2005)

Further, RWDI has provided feedback on updates to the B.C. Oil and Gas Commission’s Noise Control Best Practice Guideline. RWDI personnel were also integral in the development of the Nail-Laminated Timber Canadian Design Construction Guide which provides best practice techniques for controlling acoustic issues that arise in timber framed construction.

Policy Objectives

The MoTI policy was developed to establish community noise objectives and provide a broad range of mitigation options for projects involving numbered highways. The goal is to avoid unacceptable noise levels at noise sensitive locations near highways.

The MoEMPR document was developed to recommend a best practice for conducting sound assessments of wind power projects that meets requirements and intent of the Land Use Operational Policy – Wind Power Projects, the requirements of the BC Clean Energy Project Development Plan Information Requirements, and provides sufficient technical analysis for reviewers to evaluate wind energy projects.

The VFPA policy is intended to assist applicants for projects on lands and waters managed by VFPA in the assessment of potential noise impacts associated with the operation of their proposed projects.

CoV’s SoundSmart manual was developed to familiarize residents with urban noise and its sources, how it affects people and how both exterior and interior noise can be controlled. The manual also provides guidance to prospective homebuyers and tenants on how to avoid living situations that they may find too noisy.

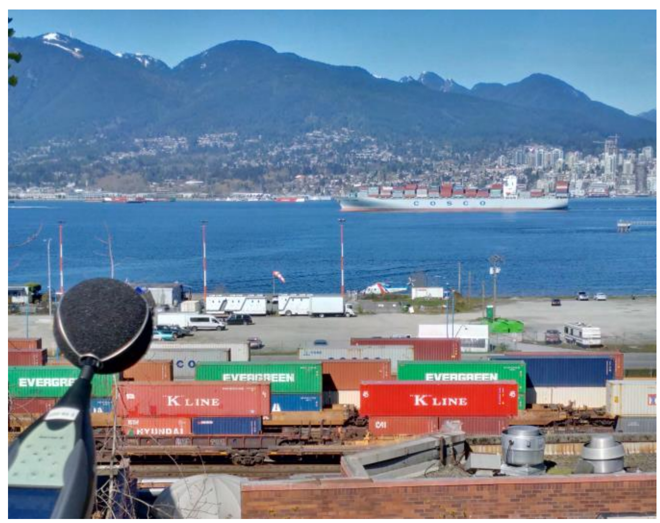

Figure 4: Rooftop noise measurements for CoV

7 Conclusion

We have provided a brief outline of some of the exciting work that RWDI has done in the fields of acoustics, noise and vibration in the province of British Columbia.

We are constantly applying our experience and knowledge in new and exciting ways to support the goals of our clients. Collaborating with the design and construction community allows us to improve the environment that surrounds us all, and to ultimately improve peoples’ lives.

RWDI’s acoustics, noise and vibration team is a small part of a greater company of world-leading experts. This unique position provides us with opportunities to work on projects from around the world and bring that experience back to our local projects.

Acknowledgments

The authors would like to thank our colleagues who make it a joy to come to work, our clients for bringing us onto their teams and entrusting us with their unique problems, and to the science and engineering community, past, present, and future, to whom we are indebted for the state of understanding from which we benefit.

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Experimental Validation Of An Acceleration Power Spectral Density Aircraft Panel Model Given Different Excitations

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Résumé

Le bruit et les vibrations dans une cabine d'avion pendant les conditions de croisière sont principalement causés par des excitations d'écoulement externes provenant de la couche limite turbulente (TBL). Le TBL fait vibrer les panneaux de fuselage de l'avion. Ces vibrations rayonnent l'énergie sonore sous la forme de bruit. Il est intéressant de pouvoir prédire la réponse de ces panneaux à différentes excitations à l'aide d'un modèle analytique, de sorte que les essais coûteux en soufflerie et en vol puissent être minimisés lors de la recherche sur le bruit. Deux modèles analytiques existants ont été modifiés pour tenir compte des différentes excitations: l'une avec des conditions aux limites simplement supportées et l'autre avec des conditions aux limites arbitraires. Ces modèles ont été programmés et validés par rapport à des données expérimentales, obtenues par les auteurs, pour un panneau mince rectangulaire avec des conditions aux limites entre des conditions simplement supportées et des conditions serrées. Le but de cette recherche est d'utiliser les modèles pour mener des études d'optimisation, simuler expérimentalement la réponse vibratoire résultante sur un panneau soumis à des fluctuations de pression TBL et utiliser un patch piézo-électrique pour simuler expérimentalement la même réponse de panel à partir d'un TBL excitation. On montre que les modèles analytiques modifiés prédisent avec précision la réponse du panneau pour une excitation de force ponctuelle, pour une excitation TBL et pour une excitation de patch piézoélectrique oscillant.

Mots clefs : couche limite turbulente, modèle analytique, densité spectrale de puissance, acoustique structurale, aéroacoustique

Abstract

The noise and vibration in an aircraft cabin during cruise conditions is primarily caused by external flow excitations from the turbulent boundary layer (TBL). The TBL causes the fuselage panels on the aircraft to vibrate. These vibrations radiate sound energy in the form of noise. It is of interest to be able to predict the response of these panels to different excitations using an analytical model, so that expensive wind tunnel and flight tests can be minimized when doing noise research. Two existing analytical models were modified to account for different excitations: one with simply supported boundary conditions and the other with arbitrary boundary conditions. These models were programmed and validated against experimental data, obtained by the authors, for a thin rectangular panel with boundary conditions between simply supported and clamped conditions. The goal of this research is to use the models to conduct optimization studies, experimentally simulate the resulting vibration response on a panel subjected to TBL pressure fluctuations and to use a piezo-electric patch as a means of experimentally simulating the same panel response from a TBL excitation. It is shown that the modified analytical models accurately predict the panel response for a point force excitation, for a TBL excitation, and for an oscillating piezoelectric patch excitation.

Keywords: turbulent boundary layer, analytical model, power spectral density, structural acoustics, aeroacoustics

1 Introduction

The noise and vibration in an aircraft cabin during cruise conditions is primarily caused by the external turbulent boundary layer (TBL) [1]. The TBL causes the fuselage panels on the aircraft to vibrate, which radiate sound energy in the form of noise in the cabin. In this context, the objective of this study is to validate an analytical model which predicts the behaviour of an aircraft panel, subject to different excitations, and with simply supported and arbitrary boundary conditions. The model will be given 1) a point force excitation from an impact hammer, 2) a turbulent boundary layer excitation caused by the flow on the outside of the panel, and 3) an excitation from a piezoelectric actuator bonded to the panel. The theoretical values, as predicted by the model, are then validated against experimental data for the three excitations.

Many researchers have studied the prediction of the response of a simple panel due to the TBL. Strawderman and Brand have some of the earliest simulated results for a turbulent flow excited panel vibration [2]. Others have modelled the response of the plate using wavenumber-frequency formulations, or have used finite element and

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boundary element methods where the plate is excited by a number of distributed forces having proper spatial and temporal correlations [3–6]. These methods tend to be very computationally intensive and as such are not a suggested approach when using recursive optimization routines or control algorithms, which are often involved in the reproduction of these types of responses. These types of models tend to be very robust for a variety of complex experimental conditions however, this makes them overanalyze simplified experimental conditions causing more calculations to be performed per iteration of an optimization routine.

One approach to calculate the radiated sound power (RSP) of vibrating structures is to use a modal analysis, as done by Roy and Lapi [7]. This approach is necessary when analyzing obscure shapes, but requires great computational power and time, making it difficult to iterate the calculations for optimization routines. Therefore, when looking at simple shapes, like that of a flat panel, analytical computational methods become a better choice. The analytical expressions for RSP can be derived for a given aircraft panel in terms of the displacement power spectral density (PSD) [1,8,9]. The acceleration PSD is calculated from the displacement PSD, which is proportional to the RSP [9]. The analytical models previously developed by Rocha were modified to account for other panel and enclosure combinations [10,11]. Berry also showed that the same type of analytical analysis was possible for panels with arbitrary boundary conditions [12].
Other studies have attempted to reproduce the TBL excitation using loudspeakers [13–19]. It was found that at low frequencies accurate reproduction can be obtained, however, the higher the frequency range the more loudspeakers are required and the more complex the control signals become. It has been predicted that using piezoelectric patches to excite the panel might require less actuators than loud speakers to obtain the same quality of reproduction at low frequencies and it might allow the response to be reproduced for higher frequency ranges because it removes the air gap in between the excitation device and the panel [20]. Piezoelectric patches also come in varying sizes allowing more to be bonded to the panel then the amount of loudspeakers that can be arranged in front of the panel. This is an additional reason why it is of great importance in this paper to prove that an accurate model exists for a panel with an excitation from a piezoelectric patch.

There have been many experimental setups used to try to replicate an aircraft panel. Some have attempted to reproduce a panel with simply supported boundary conditions, which allows the equations for the acceleration PSD response to be simplified [21–25]. However, these experimental setups are either very difficult to manufacture or are structurally weak for a thin aircraft panel. Additionally, a true aircraft panel, which were assumed to be simply supported for most of the tests outlined in this paper, are not actually simply supported as they often have boundary conditions in between simply supported and clamped conditions. Therefore, the experimental setup used to validate Berry’s model for arbitrary boundary conditions is the one used at DLR for their experimental work [19,26].

2 Methodology

Two main models have been used in this study, and further modified to account for different excitations. These are Rocha’s Model and Berry’s Model. Rocha’s Model is an analytical approach for a panel with simply supported boundary conditions and uses trigonometric spatial functions [1,8–11]. Berry’s Model is developed for a panel with arbitrary boundary conditions and uses polynomial spatial functions [12,27]. The following section describes briefly each model and different excitations used.

2.1 Rocha’s model

In this model, the panel is assumed to be flat and simply supported on all four sides. A panel, in the context of an aircraft, might not be defined as the boundary of a sheet of material, but instead as the enclosed area on that sheet, between the stringers and the formers. The connections of the material to the stringers and formers cause that section of material to act as a single, simply supported panel. The vibration of a single panel can be defined as [1]:

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

In which α_{m_x} and $\beta_{m_y}(y)$ are spatial functions that define the variation in vibration and can be defined as follows, for a simply supported plate [1]:

$$\alpha_{m_x}(x) = \sqrt{\frac{2}{a}}\sin\left(\frac{m_x \pi x}{a}\right)\,;\beta_{m_y}(y) = \sqrt{\frac{2}{b}}\sin\left(\frac{m_y \pi y}{b}\right) \tag{2}$$

Rocha’s research is able to reduce a “coupled system governing equations into the following matrix form” [1]:

$$\begin{bmatrix} M_{pp} & 0 \\ M_{cp} & M_{cc} \end{bmatrix} \begin{Bmatrix} \ddot{q}(t) \\ \ddot{r}(t) \end{Bmatrix} + \begin{bmatrix} D_{pp} & 0 \\ 0 & D_{cc} \end{bmatrix} \begin{Bmatrix} \dot{q}(t) \\ \dot{r}(t) \end{Bmatrix} + \begin{bmatrix} K_{pp} & K_{pc} \\ 0 & K_{cc} \end{bmatrix} \begin{Bmatrix} q(t) \\ r(t) \end{Bmatrix} = \begin{Bmatrix} P_{tbl}(\omega) \\ 0 \end{Bmatrix} \tag{3}$$

Where pp is used to denote the panel cc is to denote the enclosure and cp and pc are the interactions between the panel and enclosure. $q(t)$ defines the variation in $w(x,y,t)$ with respect to time and $r(t)$ is similarily defined for the enclosure. This matrix form assumes that the panel is simply supported, and encloses a cavity (like the panels surrounding the enclosed cabin of the aircraft). In this study, the author will assume that the cavity is not present and therefore the system equations can be reduced to:

$$H_w(\omega) = H(\omega) = \left[-\omega^2 M_{pp} + i\omega D_{pp} + K_{pp}\right]^{-1} \quad (4)$$

Where [1]:

$$M_{pp} = \; diag[\rho_p h_p] \quad (5)$$
$$D_{pp} = \; diag[2\rho_p h_p \omega_m \zeta_p] \quad (6)$$
$$K_{pp} = \; diag[\rho_p h_p \omega_m^2] \quad (7)$$

Each of these matrices are of size MxM. With this information, $S_{ww}(\omega)$ matrix can be defined as follows [1]:

$$S_{ww}(\omega) = H_w^*(\omega) S_e(\omega) H_w^T(\omega) \quad (8)$$

In this equation, $S_e(\omega)$ is a generalized PSD matrix of the different excitations. The * operator is used to denote the Hermitian conjugate and the T operator indicates the transpose of the matrix. With this displacement PSD matrix, the displacement PSD at a single point can be calculated for a given frequency as follows [1]:

$$S_{WW}(x_1, y_1, x_2, y_2, \omega) \quad (9)$$
$$= \sum_{m_{x_1}, m_{x_2}=1}^{M_x^2} \sum_{m_{y_1}, m_{y_2}=1}^{M_y^2} \begin{matrix} \alpha_{m_{x_1}}(x_1)\alpha_{m_{x_2}}(x_2) \\ \beta_{m_{y_1}}(y_1)\beta_{m_{y_2}}(y_2) S_{ww}(\omega)_{m_1,m_2} \end{matrix}$$

When $x_1 = x_2$ and $y_1 = y_2$ this calculates the autocorrelation at a single point and if these are not equal than it calculates the cross spectrum correlation between two different points. For the TBL excitation the autocorrelation is used and for the point force and the piezoelectric patch excitations the cross spectrum correlation is used. The equations required to calculate the velocity (S_{VV}) and the acceleration PSD (S_{AA}), at a single point on the panel are as follows [9]:

$$S_{VV} = \; \omega^2 \, S_{WW} \quad (10)$$
$$S_{AA} = \; \omega^4 \, S_{WW} \quad (11)$$

More information regarding Rocha's Model can be found in Appendix A.

2.2 Berry's model

The vibration of a single panel can still be defined as in equation (1), however, as Berry shows, the spatial functions used can be changed to polynomial functions [12].

$$\alpha_{m_x}(x) = \; \frac{2}{a} x^{m_x} \, ; \; \beta_{m_y}(y) = \frac{2}{b} y^{m_y} \quad (12)$$

A difference between the two methods is that Berry's model treats the panel mode indices as if they start at 0 instead of starting at 1. Similarly to equation (4), Berry defines the equation as follows [12]:

$$\left(-\omega^2 M_{mnpq} + \widetilde{K}_{mnpq}\right)\{a_{mn}\} = \{f_{mn}\} \quad (13)$$

Where:

$mnpq$ = $m_{x_1} m_{y_1} m_{x_2} m_{y_2}$ and a_{mn} = $q_{m_x m_y}$

a_{mn} is solved in equation (13) and is used to calculate the displacement PSD of the panel as follows:

$$w(x, y, \omega) = \; \sum_{m=1}^{Mx} \sum_{n=1}^{My} \alpha_m(x)\beta_n(y) a_{mn} \quad (14)$$
$$S_{ww}(\omega) = \; w(x, y, \omega) \;\; w(x, y, \omega)^* \quad (15)$$

2.3 Panel excitations and modified berry's model

For both of the models used, it is important to determine the correct way to represent the excitation. Rocha's model uses the excitation in its PSD form, whereas Berry's Model treats the excitation as a force spectrum. An impulse force can be represented as follows:

$$S_e(\omega) = \; f_{mn}(\omega) \, f_{mn}(\omega)^* \quad (16)$$
$$f_{mn}(\omega) = \; \alpha_m(x)\beta_n(y) f(\omega) \quad (17)$$

Here $f(\omega)$ is the frequency response of the force input as measured by the impact hammer. The spatial functions and mode numbering conventions change between the two models. However, using an impact force is the simplest excitation for both models.

The excitation from a TBL on the plate has previously been defined for Rocha's model. This work investigates the use of Rocha's model, for the TBL excitation, but using polynomial spatial functions [1]. An analytical equation has been defined for use with Berry's model of a TBL excitation with polynomial spatial functions. The following is the result of this derivation: the derivation starts with the Corcos model, which considers the cross power spectral density of the stationary and homogeneous turbulent boundary layer wall pressure field in a separable form in the streamwise, x-, and spanwise, y-directions, as follows [28,29]:

$$S(\zeta_x, \zeta_y, \omega) \quad (18)$$
$$= \; S_{ref}(\omega) e^{-\frac{\alpha_x \omega |\zeta_x|}{U_c}} e^{-\frac{\alpha_y \omega |\zeta_y|}{U_c}} e^{-i\frac{\omega \zeta_x}{U_c}}$$

Therefore the power spectrum from a turbulent boundary layer is defined as:

$$S_{tbl}(\omega) = \iint_{y_0}^{y_1} \iint_{x_0}^{x_1} \emptyset_m(x)\varphi_n(y)\emptyset_p(x')\varphi_q(y') \quad (19)$$
$$S(\zeta_x, \zeta_y, \omega) dx dy dx' dy'$$

Where the spatial separations in the streamwise and spanwise directions are $\zeta_x = x - x'$ and $\zeta_y = y - y'$. The polynomial spatial functions can be defined as:

$$\emptyset_m(x) = \alpha^m \, ; \; \varphi_n(y) = \beta^n \quad (20)$$

Where:

$$-1 \leq \alpha, \beta \leq 1 \text{ and } \alpha = \frac{2x}{a}\ ;\ \beta = \frac{2y}{b} \tag{21}$$

After substituting equation (18) and the polynomial spatial functions into (19) the following results:

$$\frac{S_{tbl}(\omega)}{S_{ref}(\omega)} = \iint_{x_0}^{x_1} \alpha^m \alpha'^p e^{-\frac{\alpha_x \omega |\zeta_x|}{U_c}} e^{-i\frac{\omega \zeta_x}{U_c}}\, dx dx' \iint_{y_0}^{y_1} \beta^n \beta'^q e^{-\frac{\alpha_y \omega |\zeta_y|}{U_c}} dy dy' \tag{22}$$

Let:

$$x_0 = \frac{a}{2}\alpha_0;\quad x_1 = \frac{a}{2}\alpha_1; \quad y_0 = \frac{b}{2}\beta_0;\quad y_1 = \frac{b}{2}\beta_1; \tag{23}$$

And $-1 \leq \alpha_0, \alpha_1, \beta_0, \beta_1 \leq 1$ therefore:

$$\frac{S_{tbl}(\omega)}{S_{ref}(\omega)} = \frac{a^2 b^2}{16} \iint_{\alpha_0}^{\alpha_1} \alpha^m \alpha'^p e^{-\frac{\alpha_x \omega a}{2U_c}|\alpha-\alpha'|} e^{-i\frac{\omega a}{2U_c}(\alpha-\alpha')} d\alpha d\alpha' \iint_{\beta_0}^{\beta_1} \beta^n \beta'^q e^{-\frac{\alpha_y \omega b}{2U_c}|\beta-\beta'|} d\beta d\beta' \tag{24}$$

This can be simplified to:

$$S_{tbl}(\omega) = S_{ref}(\omega) \frac{a^2 b^2}{16} S_\alpha S_\beta \tag{25}$$

Let:

$$\mu = \frac{\alpha_x \omega a}{2U_c};\ \varkappa = \frac{\omega a}{2U_c};\ \rho = \frac{\alpha_y \omega b}{2U_c} \tag{26}$$

Therefore:

$$S_\alpha = \int_{\alpha_0}^{\alpha_1} \alpha^m e^{-(\mu+i\varkappa)\alpha} \int_{\alpha_0}^{\alpha} \alpha'^p e^{(\mu+i\varkappa)\alpha'} d\alpha' d\alpha + \int_{\alpha_0}^{\alpha_1} \alpha^m e^{(\mu-i\varkappa)\alpha} \int_{\alpha}^{\alpha_1} \alpha'^p e^{-(\mu-i\varkappa)\alpha'} d\alpha' d\alpha = S_L + S_U \tag{27}$$

With a change of variables:

$$S_\beta \widehat{=} S_\alpha \tag{28}$$

This occurs when $\alpha_0 \to \beta_0$, $\alpha_1 \to \beta_1$, $\mu \to \rho$, $\varkappa \to 0$, $\alpha \to \beta$ and $\alpha' \to \beta'$. Further simplifying of these equations by setting $z = \mu + i\varkappa$ leads to:

$$S_L = \int_{\alpha_0}^{\alpha_1} \alpha^m e^{-z\alpha} \int_{\alpha_0}^{\alpha} \alpha'^p e^{z\alpha'} d\alpha' d\alpha \tag{29}$$

$$S_U = \int_{\alpha_0}^{\alpha_1} \alpha^m e^{\bar{z}\alpha} \int_{\alpha}^{\alpha_1} \alpha'^p e^{-\bar{z}\alpha'} d\alpha' d\alpha \tag{30}$$

$$S_U = \int_{\alpha_0}^{\alpha_1} \alpha'^p e^{-\bar{z}\alpha'} \int_{\alpha_0}^{\alpha} \alpha^m e^{\bar{z}\alpha} d\alpha d\alpha' \tag{31}$$

With a change of variables: $m \to p$ and $z \to \bar{z}$ then $S_U \widehat{=} S_L$. These equations can then be integrated to result in an analytical expression where $S_L = f_{m,p}$:

$$f_{m,p} = \frac{1}{z}\begin{bmatrix} m f_{m-1,p} - \alpha_1^m e^{-z\alpha_1} g_p + \\ \dfrac{\alpha_1^{m+p+1} - \alpha_0^{m+p+1}}{m+p+1} \end{bmatrix} \tag{32}$$

$$g_p = \frac{1}{z}\left[\alpha_1^p e^{z\alpha_1} - \alpha_0^p e^{z\alpha_0} - p g_{p-1}\right] \tag{33}$$

Where: $m, p \geq 0$ and $f_{-1,p} = 0$ and $g_{-1} = 0$

The piezoelectric actuator excitation has previously been defined for Berry's Model by Charette and Berry [27]. It treats the force from the piezoelectric patch as a point force located at its center. It shows how to incorporate the effects of the piezoelectric patch to the mass and stiffness matrices of the panel. These values often have very little effect on large panels due to the relative size of a single patch. However, it is important to include the patches impact in the model because the authors aim to modify the model to include multiple patches at different locations, which will have a more significant impact on the panels' mass and stiffness matrices. One major deviation in this work from Charette's paper, is that instead of using a piezoelectric patch on both sides of the panel it has only been used on one side. Therefore the equations defined are all divided by two.

3 Results

3.1 Impact hammer

The first goal was to use the impact hammer to strike the panel at one location and measure the acceleration using an accelerometer at a different location on the panel. The benefit of this test is that it is accurate and relatively simple to complete multiple configurations of hammer and accelerometer locations.

The first step was to compare Rocha's model to Berry's model. Therefore, boundary conditions of a simply supported panel were used in Berry's model and the same panel parameters were given to each code. Table 1 lists the panel parameters of the test panel used and Figure 1 shows the resulting acceleration PSD for a given impact excitation for each of the models and compares it to simulation results obtained from Ansys. The Ansys results were obtained from the same model defined previously by Misol using shell elements for the plate and torsional springs to describe the fixture at the edges [19]. Berry's model cannot accept infinity for the translational stiffness constant, therefore, a value of $5 * 10^7$was used [12]. The goal of this work is to test for a single point acceleration PSD not the overall acceleration PSD. This work is to show the accuracy of both Berry's and Rocha's model at predicting the panel response and it is not required to check the overall response however, multiple tests at different random panel locations have been taken for each test.

Figure 1 shows the models predict the same general shape for the acceleration PSD for a given point force excitation. However, there are two main differences in the outputs of the models: 1) the magnitudes of the peaks are different and 2) at higher frequencies the models appear to not agree. The difference in magnitudes of the peaks is likely due to the way the damping is entered into each model. In Rocha's model the damping is defined using the damping matrix, whereas in Berry's model the effects of the panel's damping are included as an imaginary component in the stiffness matrix. This might be the reason why Berry's model underpredicts the amount of damping compared to Rocha's model and FEM results. The divergence of Berry's model occurs around 600 Hz due to the number of panel modes used at these high frequencies. Berry's model runs into the error of the matrix appearing singular when the mode number is high, $(m, n) \geq (10,10)$. This error can be removed if frequencies of interest are small enough, or the panel's thickness is increased, requiring less panel modes. However, it appears that by selecting polynomial trial functions severely limits the frequency range that can be calculated.

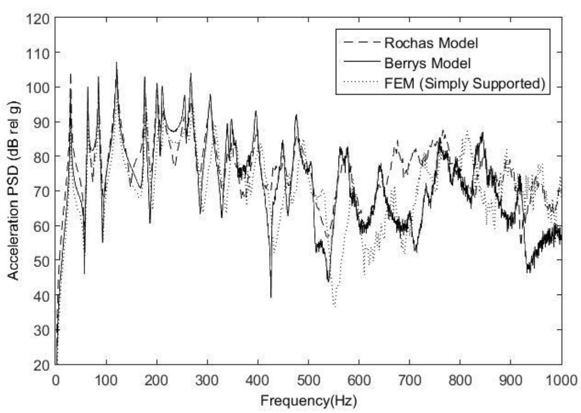

Figure 1: Comparison of acceleration PSD results for a panel with simply supported boundary conditions at a point x = 20.6 cm and y = 21.6 cm, with a point force applied at x = 7.7 cm and y = 3.8 cm for three different models.

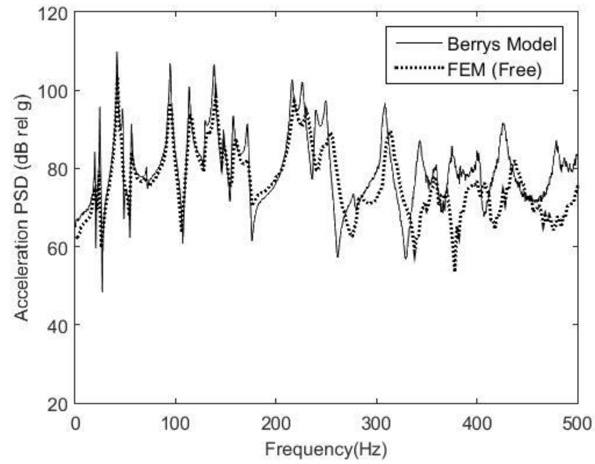

Figure 2: Comparison of acceleration PSD results for a panel with free boundary conditions at a point x = 20.6 cm and y = 21.6 cm with a point force applied at x = 7.7 cm and y = 3.8 cm for two different models.

Table 1: Physical properties of the test panel.

Variable	Description, Units	Value
a	Panel Length [m]	0.47
b	Panel Width [m]	0.37
ρ_p	Panel Density [kg m^{-3}]	2800
h_p	Panel Thickness [m]	0.0011
ν_p	Poisson Ratio	0.3
E_p	Panel Elasticity Modulus [Pa]	$6.5 * 10^{10}$
ζ_p	Damping Ratio	0.01
N_x	Panel Longitudinal Tension [Nm^{-1}]	0
N_y	Panel Lateral Tension [Nm^{-1}]	0

The next step was to check Berry's model given "free" boundary conditions. The rotational and translational stiffness constants were set to zero and the acceleration PSD at a single point was again compared to the results given from the Ansys model. The results can be seen in Figure 2.
The third step was to gather experimental data. Using the test setup at DLR, an accelerometer was placed on the panel using wax, and the panel struck using an impact hammer [19,26]. The impact hammer location and accelerometer location were measured and both the force data and the acceleration data were recorded. The accelerometer data was used as the experimental PSD values and were compared to the results given by Berry's model using the force data from the impact hammer as the input to the model. This test enabled the translational and rotational stiffness constants (c and k) to be found for the test panel. This was done by varying the values of c and k until the predicted plot most accurately matched the experimental data, as seen in Figure 3. The DLR test panel with test locations (F for Force Applied and M for Acceleration Measurement locations) can be seen in Figure 4. Some of the test instrumentation mounted on the panel can be seen in Figure 5

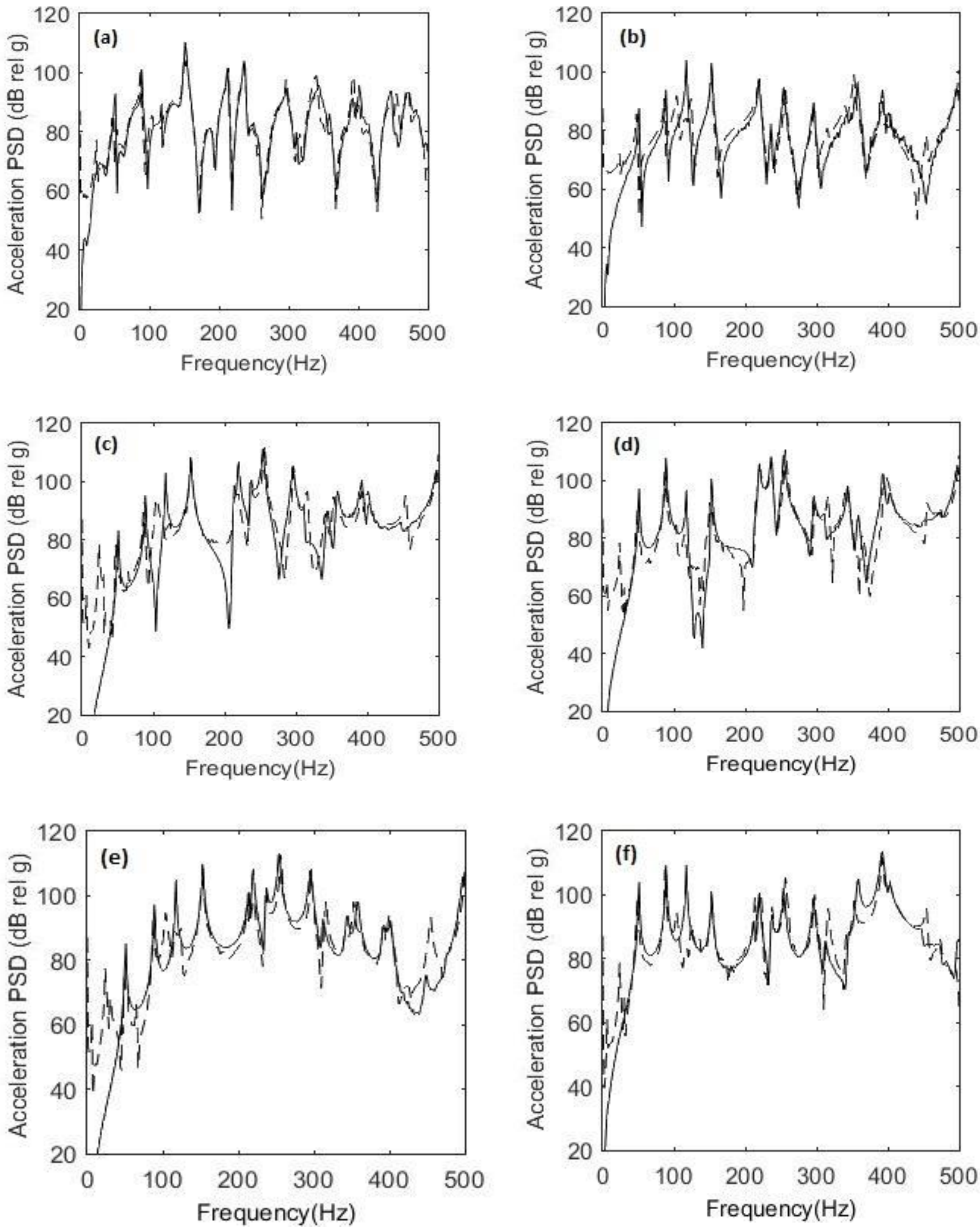

Figure 3: Comparison for predicted (solid line) vs. experimental (dashed line) of acceleration PSD results for a panel with arbitrary boundary conditions at: (a) measured at (M): x = 5.4 cm, y = 13.6 cm and force applied at (F): x = 5.4 cm and y = 13.6 cm, (b) M: x = 31.2 cm, y = 32.1 cm and F: x = 31.2 cm and y = 32.1 cm, (c) M: x = 31.2 cm, y = 32.1 cm and F: x = 4.5 cm and y = 3.9 cm, (d) M: x = 31.2 cm, y = 32.1 cm and F: x = 4.2 cm and y = 19.8 cm, (e) M: x = 31.2 cm, y = 32.1 cm and F: x = 4.2 cm and y = 32.2 cm, (f) M: x = 31.2 cm, y = 32.1 cm and F: x = 10.6 cm and y = 26.0 cm.

To ensure that the model was working accurately over the entire panel, 14 more hammer and accelerometer locations were measured experimentally. The experimental data was compared to the predicted values and they all resulted in similar plots. This indicated that the model worked over the entire area of the plate and that the values of c and k selected were accurate.

It is important to note that since Berry's model uses polynomial trial functions it takes many more modes to accurately predict the acceleration PSD than Rocha's model does. In the convergence equations, in order to get accurate predictions for Berry's model, F_{max} is set 5 times higher than when used for Rocha's model. This means it takes more modes to result in an accurate prediction of the acceleration PSD, being more computationally expensive.

3.2 TBL Excitation

To ensure the new derivation of a TBL excitation defined with polynomial spatial functions was correct, the results for Rocha's model and Berry's model were compared. Rocha's model has been previously verified compared to actual wind tunnel test data obtained at NASA, and this code has been validated against these results [1,30]. Berry's model was set with simply supported boundary conditions and run for the same flight conditions as Rocha's. Table 2 contains the flow conditions used to predict the TBL over the test panel. The results of this comparison can be found in Figure 6.

The accuracy of Berry's model given a TBL input is very sensitive to the number of panel modes used. Figure 7 and Figure 8 contain the results of using the same number of panel modes for each of the target frequencies. It shows that each target frequency requires a different number of panel modes to result in an accurate prediction of the acceleration PSD from a TBL excitation. They show that it is critical to make a preliminary study on Berry's model so reliable results can be obtained. These plots are limited to 400 Hz to highlight how sensitive Berry's model is to the number of panel modes used.

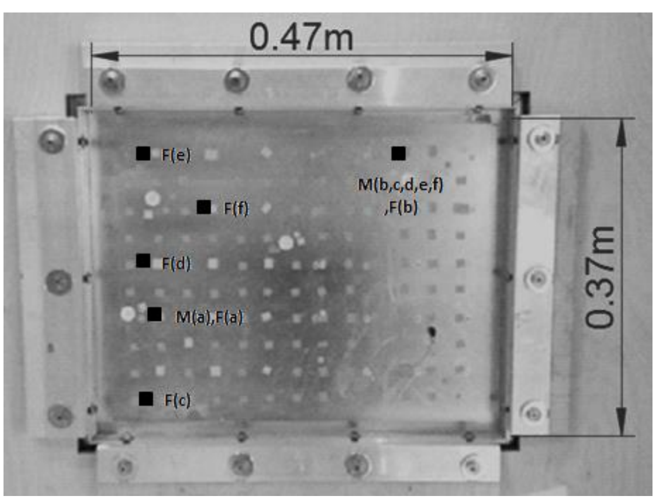

Figure 4: Front view of the DLR test setup with impact force test locations.

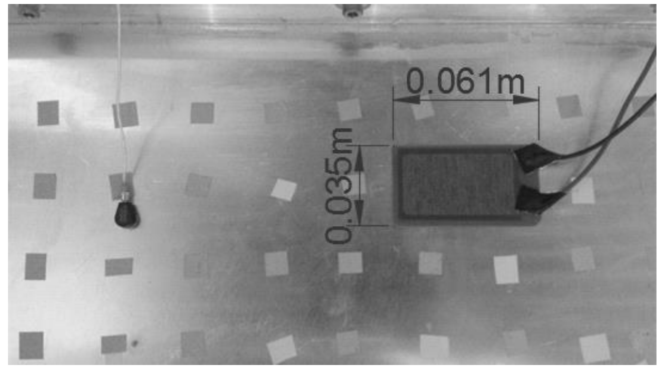

Figure 5: Excitation and monitoring system of the DLR test setup, using a piezoelectric patch and an accelerometer.

Table 2: Air parameters for determining TBL.

Variable	Description, Units	Value
ρ_0	Density of Air [kg m^{-3}]	1.225
c_0	Speed of Sound [m s^{-1}]	340
U_i	Freestream Velocity [m s^{-1}]	35.8
U_c	Convective Velocity [m s^{-1}]	23.3
M	Mach Number	0.105

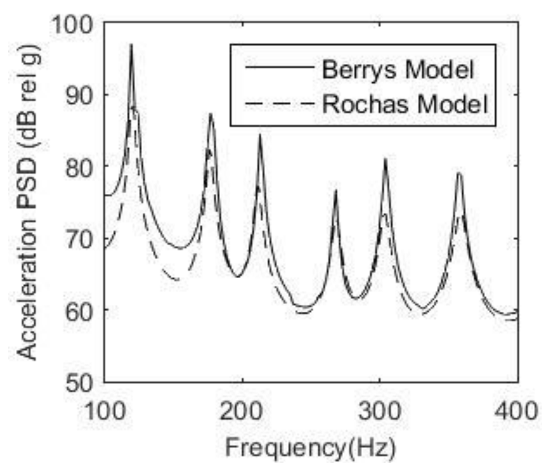

Figure 6: Comparison of acceleration PSD results for a panel with simply supported boundary conditions at a point x=a/4 and y = b/4 with a TBL excitation applied.

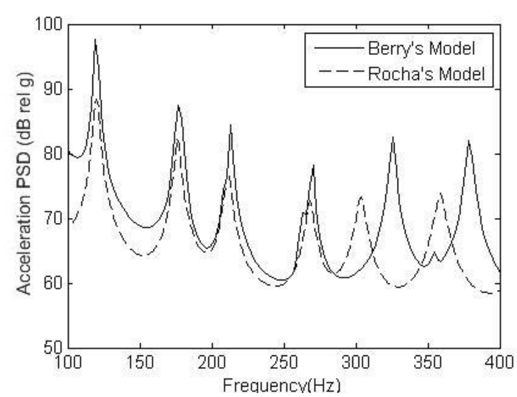

Figure 7: Comparison of acceleration PSD results for a panel with simply supported boundary conditions at a point x=a/4 and y = b/4, with a TBL excitation applied. Each target frequency is calculated with the same constant number of panel modes $(m, n) = (9,7)$.

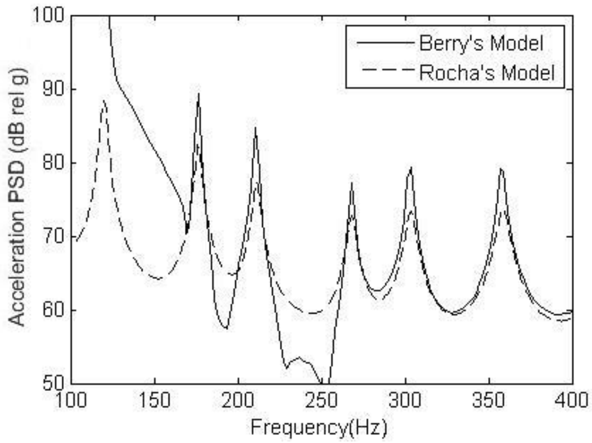

Figure 8: Comparison of acceleration PSD results for a panel with simply supported boundary conditions at a point x=a/4 and y = b/4, with a TBL excitation applied. Each target frequency is calculated with the same constant number of panel modes $(m, n) = (14{,}11)$.

Currently, in order to match Berry's model to Rocha's model, a trial and error approach is required to determine the number of panel modes used at each frequency. In order to get the results in Figure 6, the convergence test was used to calculate the number of panel modes needed, however, five times the target frequency was used as the input to the test in Berry's model.

3.3 Piezoelectric patch excitation

A piezoelectric patch has been attached to the test panel using double sided tape. This method is not as not as accurate as bonding it with glue, however, it allows for the patch to be moved and multiple tests to be run. The double sided tape has not proven to have a large impact on the results as can be seen from Figure 9. The piezoelectric patch is given a frequency sweep with a constant voltage swing and the acceleration measurements taken. Table 3 contains the parameters of the piezoelectric patch used to excite the panel. Using Charrete and Berry's piezo model for a piezoelectric patch on a panel with arbitrary boundary conditions, the acceleration PSD has been predicted and compared to the experimental data obtained at DLR by the authors. The comparison between the predicted response and the actual response is shown in Figure 9. Three other patches and accelerometer configurations have been tested and exhibit similar results. The locations of these tests on the panel can be seen in **Erreur ! Source du renvoi introuvable.**.

Figure 9 shows that the model predicts the panel response between 100 and 400 Hz. Below 100 Hz the mounting structure adds additional natural frequencies. This is why below 100 Hz the predicted model does not appear to give good results because it is the natural frequencies of the support structure that is being obtained. Also, above 400 Hz the polynomial spatial functions do not provide accurate results due to matrices appearing singular. The polynomial spatial functions at high modal numbers start to approach infinity at an exponential rate. The division by such a matrix causes the solution to appear singular. This means the prediction becomes less accurate as the modal number increases. The 400 Hz limiting frequency could be increased if the thickness of the panel is increased. However between the 100 and 400 Hz range the model appears to accurately predict the panels' response due to the piezoelectric patch excitation.

4 Conclusions

The objective of the present study is to validate models of an aircraft panel given different excitations on the panel. The models were given a point force excitation from an impact hammer, a turbulent boundary layer excitation caused by the flow on the outside of the panel, and an excitation from a piezoelectric actuator bonded to the panel. The theoretical values, as predicted by the models, are validated against experimental data from the three excitations. The models were modified to incorporate each of the excitations.

Rocha's model has previously been validated against experimental data. In this work, Rocha's model and Berry's model for a panel with simply supported boundary conditions, and the two models appear to agree for a range of frequencies (mostly low frequencies) for the different excitations. In order to study panels with arbitrary boundary conditions, Berry's model was considered for an optimization routine. The model has been modified using Rocha's power spectral density approach, and has been shown that Berry's modified model can be used to accurately predict a panel's acceleration PSD given a point force excitation, a TBL excitation and a piezoelectric patch excitation over a limited frequency range. Berry's current model has been found to only be valid for a finite number of panel modes due to the polynomial spatial functions. In the future, other spatial functions might be of interest to test to determine if a function exists that does not limit the frequency range as substantially. The different excitations have been validated against Ansys, Rocha's model and experimental data.

Since Berry's modified model has been proven to give an accurate prediction of the acceleration PSD, over a limited frequency range, for each of the excitations it can now be used to select the optimal positions of piezoelectric patches to reproduce the acceleration PSD caused by a TBL in constant cruise conditions.

Table 3: Piezoelectric patch parameters.

Variable	Description, Units	Value
L_x^{pz}	Length of piezoelectric patch [m]	0.061
L_y^{pz}	Width of piezoelectric patch [m]	0.035
L_z^{pz}	Thickness of piezoelectric patch [m]	0.0002
ρ_{pz}	Density of piezoelectric patch [kg m^{-3}]	7500
e_{31}^{pz}	Effective piezoelectric transverse coefficient (x-direction)	1.02
e_{32}^{pz}	Effective piezoelectric transverse coefficient (y-direction)	1.23
$\Delta\varphi^{pz}$	Applied voltage peak to peak [V]	8.5
V^{pz}	Applied voltage offset [V]	200

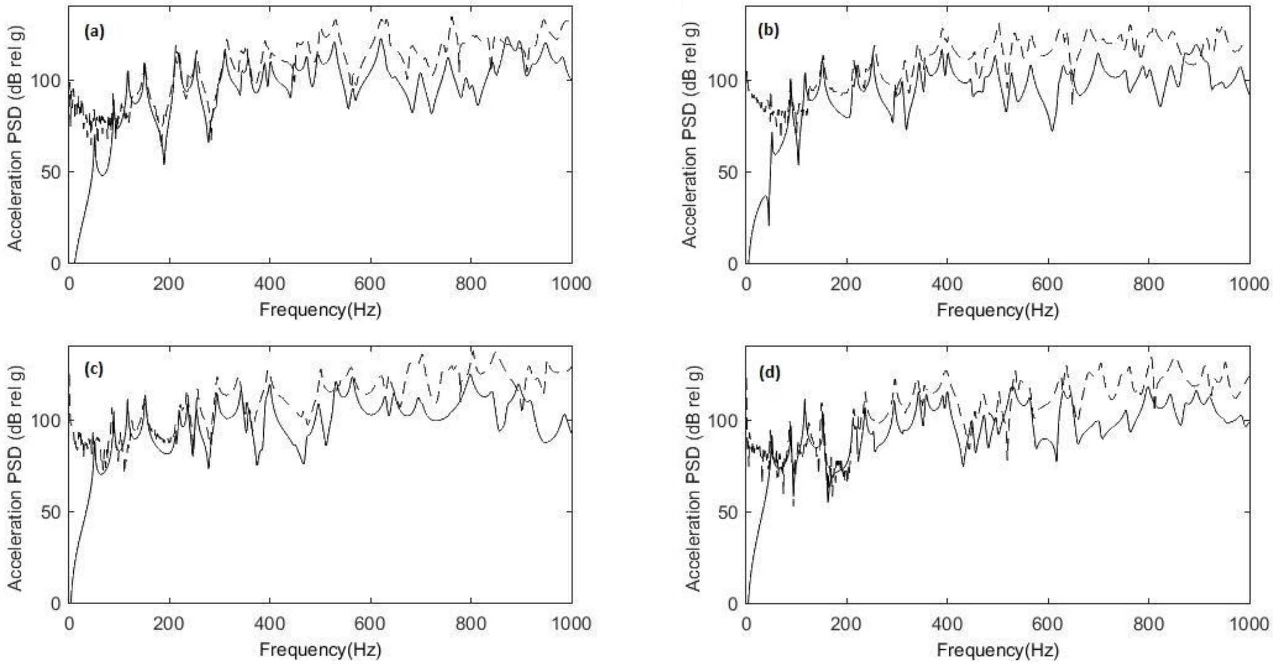

Figure 9: Comparison, using Berry's model, of predicted (solid line) vs. experimental (dashed line) of acceleration PSD results for a panel with arbitrary boundary conditions: (a) measured at (M): x = 26.2 cm, y = 8.6 cm with a piezoelectric actuator excitation applied at (F): x = 12.3 cm, y = 7.4 cm (b) M: x = 31.4 cm, y = 26.0 cm and F: x = 12.3 cm, y = 7.4 cm (c) M: x = 31.4 cm, y = 26.0 cm and F: x = 15.5 cm, y = 26.0 cm (d) M: x = 25.5 cm, y = 13.4 cm and F: x = 15.5 cm, y = 26.0 cm

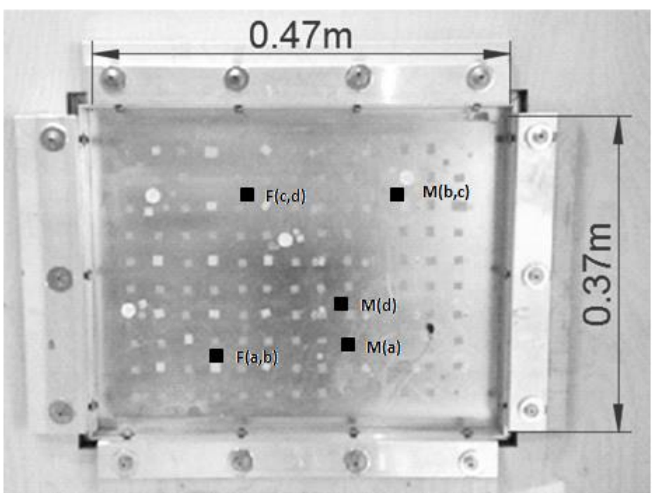

Figure 10: Front view of the DLR test setup with piezoelectric actuator excitation test locations.

This would allow for cost intensive flight and wind tunnel tests to be reduced and replaced by ground tests using a simple panel/piezoelectric patch experimental setup.

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Nomenclature

a	Panel Length [m]
b	Panel Width [m]
D_p	Panel Bending Stiffness [N m]
D_{pp}	Damping Matrix (Rocha's Model)
E_p	Panel Elasticity Modulus [Pa]
f_{mn}	Force Function Matrix based on the excitation
h_p	Panel Thickness [m]
$\widetilde{K}_{mnpq}$	Complex Stiffness Matrix (Berry's Model)
K_{pp}	Stiffness Matrix (Rocha's Model)
m_x, m_y	Plate Mode
M	Total Number of Plate Modes Considered
M_{mnpq}	Mass Matrix (Berry's Model)
M_{pp}	Mass Matrix (Rocha's Model)
N_x	Panel Longitudinal Tension [N m^{-1}]
N_y	Panel Lateral Tension [N m^{-1}]
$S_{ref}(\omega)$	Efimtsov's model of the TBL pressure spectrum
ρ_p	Panel Density [kg m^{-3}]
ν_p	Poisson Ratio
$\boldsymbol{\zeta_p}$	Damping Ratio

Appendix A – Rocha's model details

The first step to calculating the acceleration PSD is to determine the panel modes and the natural frequency that corresponds with each mode, as follows [10]:

$$\omega^P_{m_x m_y} = \sqrt{\frac{1}{\rho_p h_p}\left\{\begin{matrix} D_p\left[\left(\frac{m_x\pi}{a}\right)^2 + \left(\frac{m_y\pi}{b}\right)^2\right]^2 + \\ N_x\left(\frac{m_x\pi}{a}\right)^2 + N_y\left(\frac{m_y\pi}{b}\right)^2 \end{matrix}\right\}} \tag{34}$$

Where:

$$D_p = \frac{E_p h_p^3}{12(1-\nu_p^2)} \tag{35}$$

This equation can be simplified to assume that the panel is not under tension ($N_x = N_y = 0$) in either direction. This simplified equation can be seen below [10]:

$$\omega^P_{m_x m_y} = \sqrt{\frac{D_p}{\rho_p h_p}}\left[\left(\frac{m_x\pi}{a}\right)^2 + \left(\frac{m_y\pi}{b}\right)^2\right] \tag{36}$$

In order to determine how many modes are needed at a specific frequency, a convergence test must be completed. Convergence is reached when the distance between two nodes of the structural mode shape is less than or equal to one half-wavelength, $\lambda/2$, of the bending wave on the plate at the analysis frequency [10]. These values must be rounded to the next highest whole number, to coincide with a plate modal number [10]:

$$N_{Max} = \frac{2a}{\lambda};\ M_{Max} = \frac{2b}{\lambda}; \tag{37}$$
$$\lambda = 2\pi \left(\frac{D_p h_p}{\rho_p}\right)^{0.25} (\omega)^{-0.5}$$

The convergence test determines the point at which additional panel modes do not change the overall shape of the final plot, but instead, appear to make the plot slightly noisier. By running a convergence test at every target frequency, it allows the program to limit the number of panel modes used for lower target frequencies, speeding up the computational time to run the program.

Equation (3) can also be arranged as follows to better show how it can be reduced to equation (4) [1]:

$$Y(\omega) = H(\omega)X(\omega) \tag{38}$$

$$Y(\omega) = \begin{Bmatrix} W(\omega) \\ P(\omega) \end{Bmatrix} \tag{39}$$

$$X(\omega) = \begin{Bmatrix} P_{tbl}(\omega) \\ 0 \end{Bmatrix} \tag{40}$$

$$H(\omega) = \begin{bmatrix} -\omega^2 M_{pp} + i\omega D_{pp} + K_{pp} & K_{pc} \\ -\omega^2 M_{cp} & -\omega^2 M_{cc} + i\omega D_{cc} + K_{cc} \end{bmatrix}^{-1} \tag{41}$$


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ACOUSTIC NETWORK

ACOUSTIC PROJECT OF A CONFERENCE ROOM OF THE SECONDARY SCHOOL "AVENIR 33" (DELÉMONT, SWITZERLAND)

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Résumé

Cet article rapporte l'étude acoustique d'une salle de conférence dans une école secondaire en Suisse. L'approche architecturale s'est concentrée sur le choix de la forme et des matériaux de l'auditorium, tandis que les traitements acoustiques proposés devaient assurer une bonne isolation et de l'intérieur ainsi qu'une bonne intelligibilité de la parole. Les simulations acoustiques ont été effectuées avec le logiciel Odeon. Elles ont permis de prévoir les principaux indicateurs acoustiques (T_{30}, EDT, C_{80}, D_{50} et STI) et donc d'optimiser les surfaces et les positions des traitements acoustiques pour différentes utilisation de la salle (taux d'occupation). Ainsi, les traitements acoustiques proposés permettent une bonne propagation de la voix de l'orateur sans besoin d'un système de sonorisation. Une campagne de mesure acoustique réalisée à la fin de travaux de construction confirme la bonne qualité acoustique de l'auditorium.

Mots clefs : acoustique de la salle, simulation, maquette virtuelle, auditorium.

Abstract

This paper reports the study of a conference room inside a secondary school in Switzerland. The architectural approach developed focused on the choice of the room's shape and materials, while the acoustic treatments, proposed had to provide a good speech intelligibility. Simulations run with the software Odeon allowed to foresee the main acoustic indicators (T30, EDT, C80, D50 and STI) and therefore optimize the dimensions and positions of the acoustic treatments for different audiences. Moreover, the acoustic treatment aims to enhance the vocal emission of the speaker without amplification systems. Acoustic measurements carried out at the end of the building construction confirmed the good acoustic quality of the conference room.

Keywords: room acoustics, simulation, virtual model, conference room.

1 Introduction

Large enclosed spaces have mainly evolved over time due to the need to accommodate people with common interests for activities and events [1, 2, 3]. The functions for which an environment is intended imply forms of visual and sound communication, it is therefore essential, for the acoustic design, to define the intended use as well as the appropriate choice of the acoustic parameters to be analysed such as reverberation time, definition, clarity and STI (Speech Transmission Index). The optimal acoustic conditions required are achieved by either means of appropriate acoustic correction interventions or inserting appropriate amounts of sound-absorbing material into the room [4]. The environment in question is the conference room of the secondary school "Avenir 33" in Delémont (Switzerland) in 2016. The project as a whole included a school canteen with 212 places, a 154-seat conference room, a library, a wing for the secretariat and teaching rooms, a chemistry lab, 4 computer rooms, 4 theoretical classrooms and 6 classrooms for the carrying out of practical lessons. Prefabricated wood caissons were used for the structures of the load-bearing facades and structural floors. The interior surfaces of the rooms (walls and ceiling) were covered in wood, with the choice of the material being connected to the need to obtain warmer and more domestic tones, which promote a feeling of calm, concentration and warmth. Particular attention was given to sound insulation (impact sound and plane noise) inserting a layer of sand (thickness 6.0 cm) inside the floors. Finally, given the size of the classrooms, the wooden panels were appropriately perforated to obtain a suitable sound-absorbing system to reduce the unwanted reverberation. Figure 1 shows the external view of the school. Furthermore, attention was given to the acoustics of the large spaces on the ground floor such as the conference room and school canteen. The requests of the client were that the walls and ceiling had to be covered in wood and, in this case, perforated wooden panels were chosen. The conference room is a regular volume of parallelepiped shape. Figure 2 shows the internal view of the conference room. While Figures 3, 4 and 5 show the section of the room with the main dimensions. The geometrical

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characteristics are summarized in Table 1. The shape of the room is bound to the entire structure, with a regular geometry and large windows on one side for lighting needs with natural light. The perimeter wall towards the outside is made of large transparent glass surfaces, covering about 33 m^2. The interior side walls and ceiling are covered with wooden panels and the floor is covered with a carpet. The choice to use wooden panels is due to aesthetic and functional requirements. The wooden panels can be either perforated, so as to give the walls suitable sound absorption characteristics or be unperforated and therefore reflect the incident sound waves to obtain suitable sound reflection characteristics.

Figure 1: Photo of the external view of the school (photo by Thomas Jantscher).

Figure 2: Photo of the conference room (photo by Corinne Cuendet).

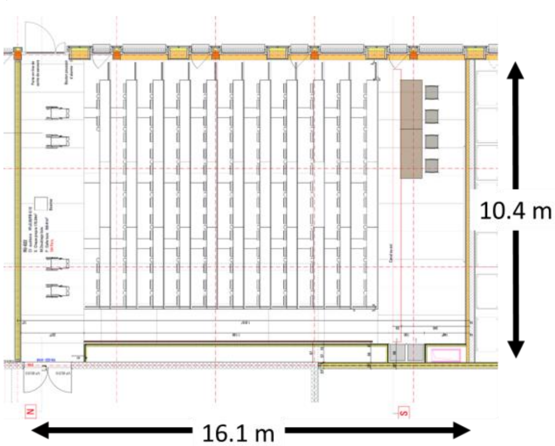

Figure 3: Plan of the conference room.

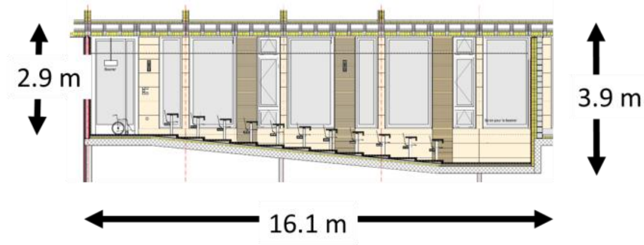

Figure 4: Lateral section of the conference room.

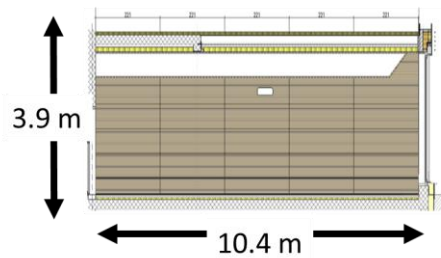

Figure 5: Frontal section of the conference room.

Table 1. Conference room dimensions.

Length, m	16.1
Large, m	10.4
Maximum height, m	3.9
Minimum height, m	2.9
Volume, m^3	570
Maximum capacity	154 seated
Glass surface, m^2	33

2 Acoustic project

For the purpose of a suitable acoustic design, along with a correct assessment of the characteristics of the materials to be used in the room, it was necessary to know in advance the intended use of the environment, from which the optimum reverberation time can be determined. For each use of the room, a suitable value of the optimal reverberation time as well as the other acoustic parameters was provided. The intended use of the school conference room was for conferences and screening films. The design of the room had to therefore be dedicated to the comprehension of speech, with the possible use of electroacoustic sound amplification systems. Therefore, it was necessary to estimate the value of the optimal acoustic parameters according to the intended use. For rooms intended to listening to speech, the reverberation time values required are shorter than those required for concert halls. The effect of the sound tail produced by excessive reverberation can lead to the superimposition of the sound impulses causing the consequent distortion of the perception of the speech. For example, a short reverberation time involves an acoustically too "dry" environment, where the comprehension of speech is insufficient due to the presence of direct sound and the absence of contributions to the sound field coming from the reflections of sound from the surfaces of the room. In this configuration, the voice emitted by a speaker arrives weak and feeble to the listener present, reducing the ability to understand speech. An excessive length of reverberation time or the presence of a long sound tail involves a reverberation environment where the understanding of speech is insufficient due to the overlap of the phonemes, with this effect being generated by multiple number of reflections of the waves. Several solutions were adopted during the design phases to reach the best listening conditions in the conference room. The wooden chairs and tables are periodic surfaces, that give a sound diffusion The floor was covered with a thin layer of carpet, with a low sound absorption. The wall claddings of the hall were made of wooden panels. This panel has good sound absorption characteristics if suitably perforated and installed at a suitable distance from the rigid rear wall, with a layer of sound-absorbing material being inserted into the gap to improve sound absorption. However, it can also be an acoustically reflecting panel if the holes are closed, thus creating a smooth acoustically reflecting panel. The ceiling was partly realized with acoustically reflecting wooden panels positioned in the area where the speaker sits. This solution was chosen so as to reinforce the direct sound component of the voice, while in the area away from the speaker, in the area where the listeners are, sound-absorbing wooden panels were installed. In addition, the wall to the right of the speaker's position is made of a large glazed surface, while the one on the left is completely made of wood. One of the acoustic problems is due to the rectangular hall plan with parallel flat walls. In environments with parallel flat surfaces, many reflections are generated, due to the reflections of the sound on the walls, with undesired acoustic effects such as "flutter echo". To reduce the effects of the multi reflections coming from the side walls, which generate acoustic defects, the wall on the left side of the speaker was made with wooden panels so as to create a smooth reflecting surface, interspersed with 1.0 metre wide stripes and equal height to that of the hall, of sound-absorbing wooden panels. This arrangement allows to reduce the effects of the sound multi reflections. The wall to the right of the speaker, the areas in which the windows are not present, were lined with sound-absorbing wooden panels. Similarly, for the ceiling, this was partly achieved with sound-absorbing wooden panels and partly with reflective wooden panels, with them reflecting the sound and reinforcing the speaker's voice which reflected properly from the ceiling propagates uniformly throughout the conference room. During the design phase, the acoustic study was drawn up using a 3D CAD model of the room, in order to evaluate the appropriate amount of sound-absorbing material to be inserted to obtain the desired reverberation times. Figure 6 shows the location of the wooden panels installed in the conference room on the vertical walls (60 m^2). While Figure 7 shows the location of the sound-absorbing wooden panels under ceiling in the conference room (35 m^2). The total area covered is 95 m^2.

3 Acoustic parameters

One of the most used acoustical parameters for evaluating the quality of a room is the reverberation time (measured in seconds). This parameter was introduced at the end of the 1800s by Clement W. Sabine and is defined as the time interval in which the sound energy decreases by 60 dB after switching off the source [5].

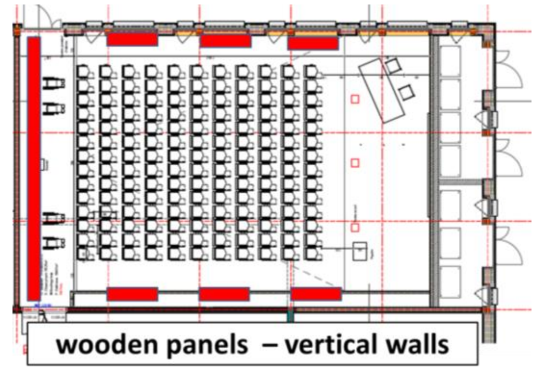

Figure 6: Location of the sound-absorbing wooden panels on the vertical walls in the conference room.

Figure 7: Position of the sound-absorbing wooden panels under ceiling in the conference room.

In a room, the longer the reverberation time, the greater the contribution of the components of the reflected sound compared to the direct one. The reverberation time value is a function of the room volume and the total sound absorption of its internal surfaces. The absorption of the materials changes with the variation of the frequency, the reverberation time also varies at the different frequencies considered. In architectural acoustics, for the acoustic correction of closed environments, the frequencies in octave bands from 125 Hz to 4.0 kHz are considered. Frequencies above 4.0 kHz are not considered because sound absorption by air prevails. Frequencies below 125 Hz are not considered because the human voice produces very little sound energy below 125 Hz that would be useful or important for speech intelligibility.

The acoustic parameters, for the purposes of the study and understanding of architectural acoustics, are reported in the ISO 3382-1 (2012) standards "*Acoustics - Measurement of room acoustic parameters"* [6]. EDT, (Early Decay Time) defined as the time taken for the sound level to decrease by 10dB. It is a time that takes into account the direct sound and approaches that which is the subjective perception of the time of decay. C_{80}, index of clarity, measures the goodness of listening to music in a hall. It is also a function of the reverberation time and the distance of the listener from the orchestra and therefore is linked to the subjective intensity of the direct sound. D_{50}, index of definition, is related to the understanding of speech. It represents the ability to distinguish sounds that follow one another over time. Speech Transmission Index (STI) represents the degree of amplitude modulation in a speech signal and refers to the distortion in speech signals caused by reverberation, echoes, and background noise.

In particular, typical suggested values of the different monaural acoustic parameters for both the speech comprehension and music listening are: the reverberation time T_{30} should assume values below 1 s for a clearer perception of **speech**, while it may assume greater values, around 2 s for music listening preference; the definition D_{50} may assume values from 0 to 1.0, but for a good speech comprehension should have values above 0.50; the indices STI can take values between 0 and 1.0, being greater than **0.6** for favorable speech conditions. Table 2 shows the optimal values of the acoustic parameters for the different listening conditions, from which it is noted that for an environment intended for speech an optimal reverberation time of less than 1.0 seconds is required [7, 8].

Table 2: 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	> 2	> 0.5

4 Acoustic virtual model simulations

During the design phase, the architectural acoustics software "Odeon" was used to study the conference room acoustics as well as choose the type of sound-absorbing materials and relative surfaces to be covered [9, 10, 11]. The software Odeon uses the principles of geometrical acoustics and adopts a hybrid calculation method that combines two classical methods, the image source method and the ray-tracing method. The software imports 3D CAD models, modelled as flat surfaces so that the ray-tracing technique or images can be used. The transition order (TO) at which the software changes from the early image source method to the late ray-radiosity method was set equal to 2 in order to consider some efficient early reflections coming from the theatre surfaces. The impulse response length equal to 3,000ms, with a resolution of 3.0ms, and number of rays equal to 100,000. The presence of the chairs is simulated with a box the size of the seating area (lx=9.0, ly=10.0, lz=0.50). It is then necessary to evaluate the presence of diffusing surfaces to which the scattering value must be assigned (for chairs s = 0.7) [12, 13, 14]. The area corresponding to the seating area is equal to 90 m^2. Every material present in the numerical model is assigned a value of the acoustic absorption coefficient in octave bands from 125 Hz to 4.0 kHz [15, 16, 17]. The absorption coefficient values were obtained from current technical literature, with the materials in use being absorbent wooden panels and reflective wooden panels, glazed surfaces, carpet for the floor, wood for the chairs and tables. The area corresponding to the wooden panels absorbent is equal to 95 m^2. Table 3 shows the absorption coefficient values assigned to the different materials using computer-aided simulation. These values are taken from the specifications.

Table 3: Sound absorption coefficient values of the materials using computer-aided simulation

Freq. , Hz	125	250	500	1 k	2 k	4 k
wooden panels rigid panels	0.03	0.03	0.03	0.04	0.05	0.07
wooden panels abs. panels	0.30	0.40	0.80	1.00	0.80	0.60
Wooden chairs unoccupied	0.06	0.10	0.10	0.20	0.30	0.20
Wooden chairs occupied	0.30	0.50	0.80	0.80	0.90	0.80

The numerical simulations, based on the absorption coefficient values assigned to each surface of the room, provided values of the acoustic characteristics during the design phase. For example, according to these evaluations, the reverberation time at the frequency of 1.0 kHz is about 1.0 s, this value represents the final goal of acoustic design, i.e. to obtain a room in which the comprehension of speech is the main objective. Numerical simulations were also performed to evaluate the presence of the audience in the hall. This simulation was performed by replacing the absorption coefficient value of the empty chairs, with the absorption coefficient value of the seats occupied by the public [18, 19]. Figures 8, 9, 10 and 11 show, respectively, the trends of the values of the predicted without audience and predicted with audience of the acoustic parameters (EDT, T_{30}, C_{80} and D_{50}). The values of the T_{30} and EDT at medium frequencies are about 1.0 s. In the simulations as shown in the graph, the effect of the presence of the audience manifests itself in a significant way on the parameter D_{50} that from 0.50 of the empty hall increases to 0.6 with an occupied hall. The frequencies of 500 Hz and 1.0 kHz are important for the purpose of understanding speech. The acoustic simulations show that at the frequencies of 500 Hz and 1.0 kHz the room, also with the presence of the audience, has good features for speech understanding.

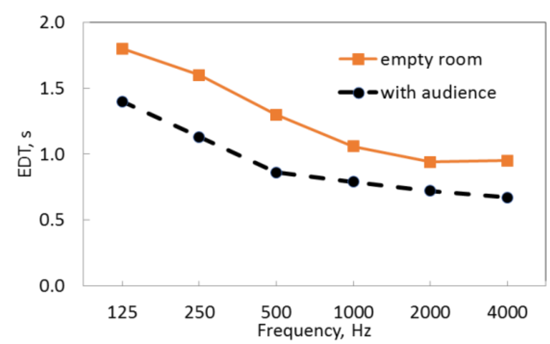

Figure 8: Average calculated values of EDT.

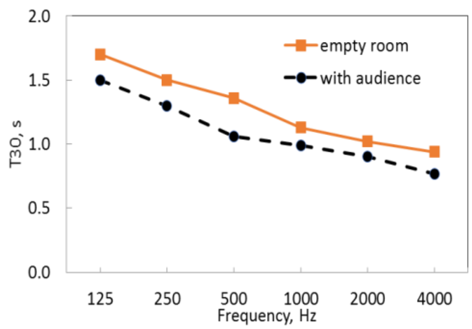

Figure 9: Average calculated values of T_{30}.

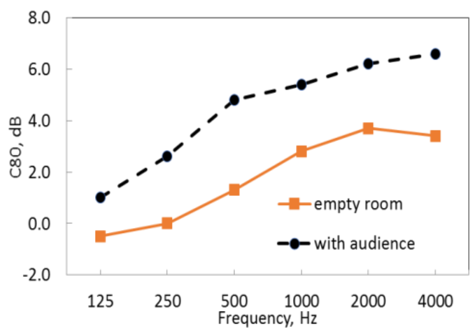

Figure 10: Average calculated values of C_{80}.

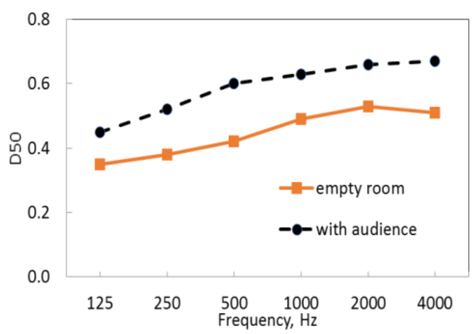

Figure 11: Average calculated values of D_{50}.

5 Acoustic measurements

After the room was designed according to the design guidelines and following the indications of the processing performed with the Odeon software, measurements of the acoustic characteristics were carried out to evaluate the acoustic conditions of the project. The acoustic measurements were carried out in an empty room with an average internal temperature of about 20 °C and a relative humidity of 50%. There were no noisy activities in the school, and the noise of vehicular traffic was negligible considering the distance from the nearest road. During the acoustic measurements the sound level was less than 35dBA. To reduce the background noise, the measurements were taken in empty conditions without students, so the impulse responses were all recorded under empty conditions. An ambisonics microphone was used for recording the impulse responses. Toy balloons, inflated with air, were used as sound sources. A balloon popping produces an impulse that excites the sound field, furthermore the background noise is very low due to it being far from any intrusive noises. A balloon popping gives a sufficient signal to noise ratio (SNR). The sound sources were positioned, at the height of 1.6 m from the floor (the height of the orator) in the position where the orator would be standing, the microphone was located on the seats, at a height of 0.8 m in different positions, to simulate the possible positions of the listeners. Eight measuring microphone points were identified (Figure 12) for the evaluation of the spatial average values of the acoustic parameters. The position of the sound source as well as the measuring points coincides with the points considered in the numerical simulation with the software Odeon.

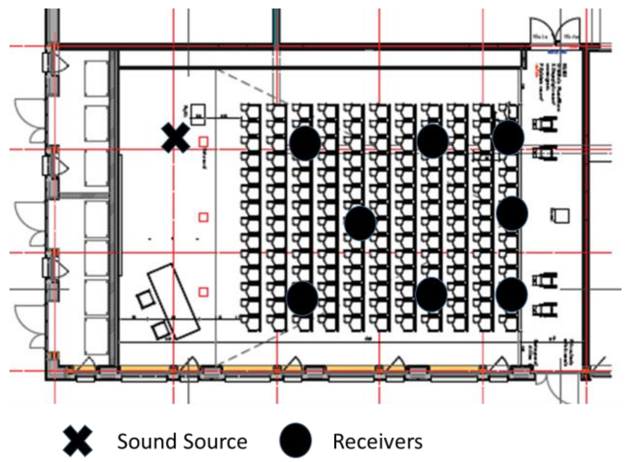

Figure: 12. Position of the sound source and the receivers.

The recorded impulse responses were elaborated with the software Dirac 4.0, analysing the acoustic parameters defined in the ISO 3382-1, such as reverberation time (T_{30}), EDT, clarity (C_{80}), definition (D_{50}), and sound transmission index for speech intelligibility (STI). The acoustic procedure and post processing methodology were similar to those used in other spaces, as in many other theatres and rooms [20, 21].

6 Discussion

Figures 13, 14, 15 and 16 report the average measured values of the acoustic parameters EDT, T_{30}, C_{80}, and D_{50}, together with the intervals of the standard deviation. These values are averaged among the eight receiver locations for each octave band frequencies from 125 Hz to 4.0 kHz, while with an overall average STI=0.62 (+/- 0.03). All the measurements were carried out under unoccupied conditions. Obviously, the audience would have had an absorbing effect with a consequent reduction of the reverberation as well as the EDT and the increase of D_{50} and STI. The parameters C_{80} and D_{50} are sensible to the early part of the impulse response, because there are large spatial variations. While for the parameter EDT at the frequencies of 125 Hz and 250 Hz, there are sensible variations of the standard deviation. For the parameter T_{30}, there are no sensible variations of the standard deviation.

The average values of the measured acoustic parameters coincide with the values obtained from the numerical prediction using the software Odeon. The value of T_{30} is almost uniform in the considered frequency range with it varying from 1.18 s to 125 Hz to 0.9 s at 4.0 kHz. Moreover, T_{30}, at the frequencies of 500 Hz and 1.0 kHz, obtained through the measurements in situ is about 1.0 s. The parameter D_{50}, which expresses in percentage the number of phonemes actually included, assumes values of about 0.5, also confirming the prediction made with the software, the value of D_{50} = 0.5 confirms that the room has good conditions for listening to the spoken. The values of C_{80}, measured experimentally post work, deviates slightly from the values obtained by the prediction made with Odeon 11. The Speech Transmission Index (STI) was considered, with this parameter representing the degree of amplitude modulation in a speech signal and refers to the distortion in speech signals caused by reverberation, echoes, and background noise. This index can assume values between 0 and 1, being greater than 0.6 for favourable speech conditions. The measured average value, with an empty room, of the STI = 0.62, confirms that there are good characteristics for listening to speech. The presence of sound absorbing lateral surfaces prevented the formation of possible acoustic defects due to the multiple reflections such as flutter echo. Therefore, the insertion of the sound-absorbing and partly reflective wooden panels allowed to obtain good acoustic comfort conditions inside the conference room.

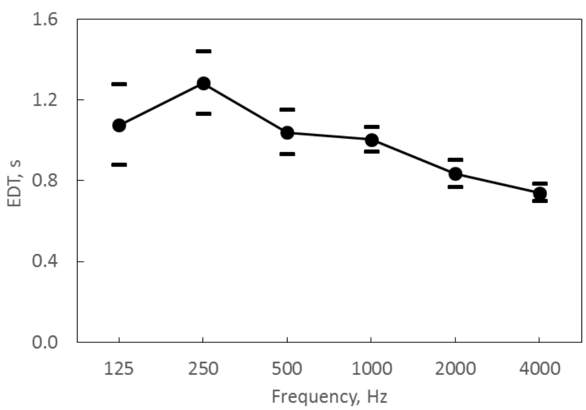

Figure 13: Average measured values of EDT.

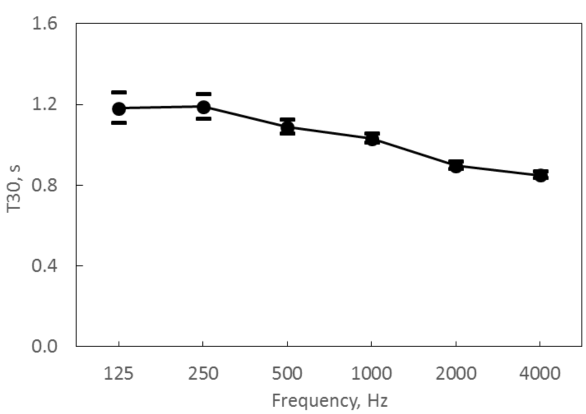

Figure 14: Average measured values of T_{30}.

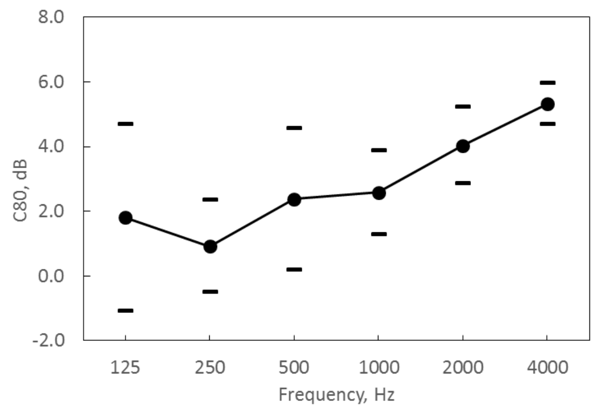

Figure 15: Average measured values of C_{80}.

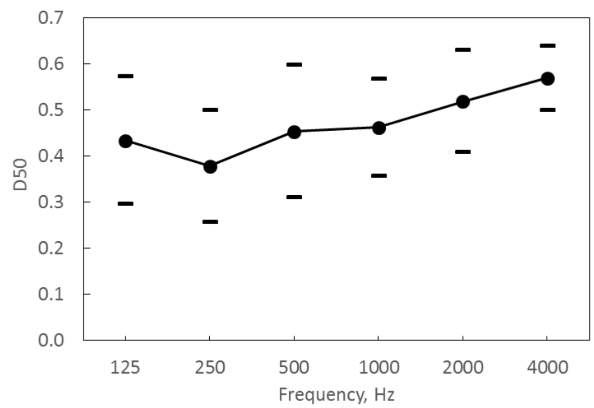

Figure 16: Average measured values of D_{50}.

7 Conclusions

This work reports the acoustic simulation of a conference room. The study was performed in a first phase by choosing the materials to be used to obtain an adequate value of the acoustic characteristics for the speech understanding. The conference room was built, and after acoustic measurements were carried out to check if the acoustic characteristics correspond to those required by the project. Simulation techniques, using the software Odeon, provided useful information for understanding how to improve the acoustics of the conference room. Firstly, simulations were used to obtain realistic data about the current acoustics of the conference room under unoccupied conditions. The acoustic treatments that were proposed guaranteed an optimal value of the reverberation to about 1.0 s at middle frequencies. The insertion of the sound-absorbing and reflective wooden panels allowed to obtain good acoustics of the conference room. The estimated values of the reverberation time (T_{30}) and definition (D_{50}) with the numerical prediction using the Odeon software coincided with the values of the experimental measurements. In conclusion, the results of this study attest to the effectiveness of the planned intervention with the acoustic simulation and therefore the reliability of the prediction carried out using the software.

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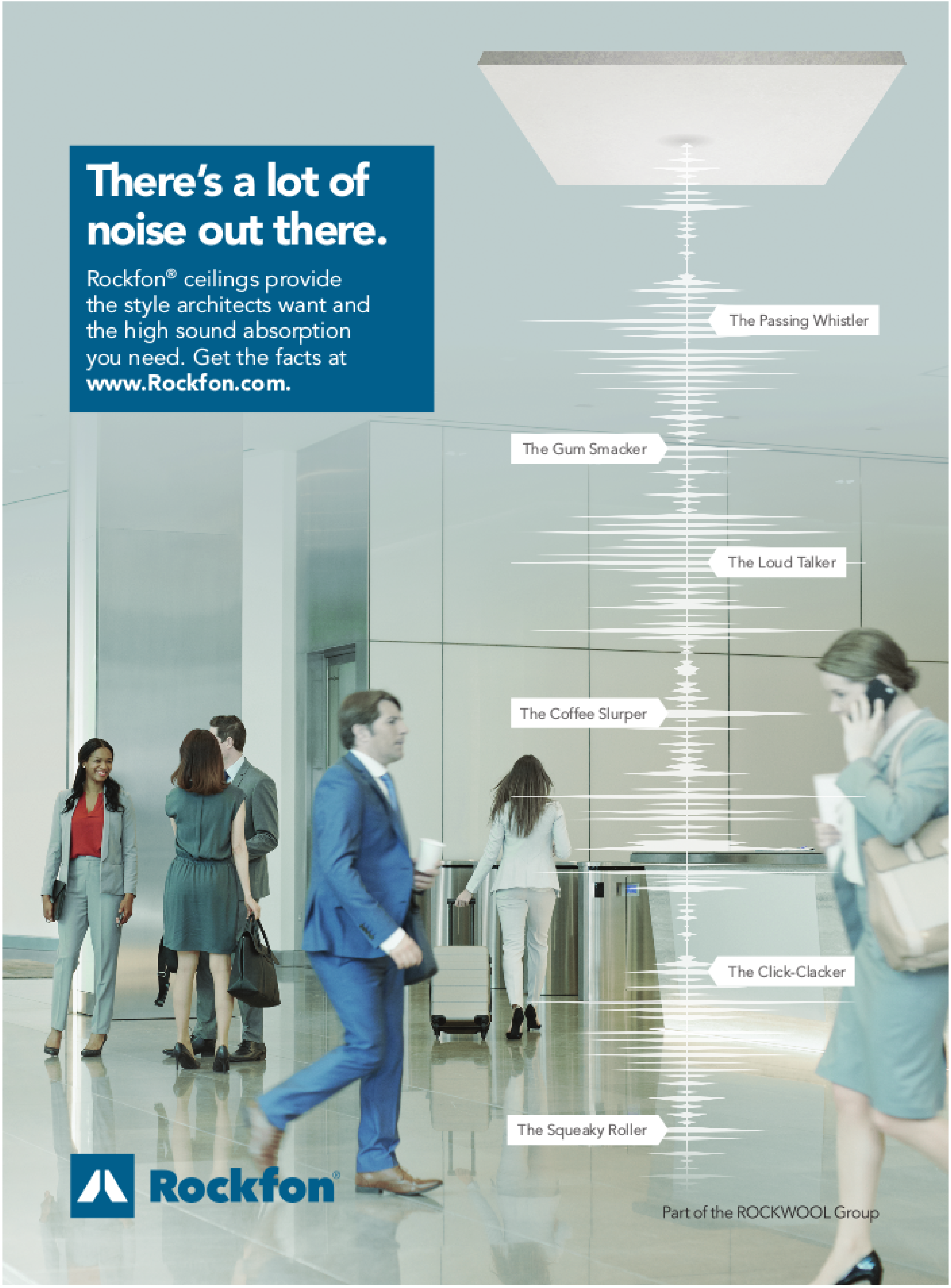

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LIVING WALL AND ACOUSTIC COMFORT – A CASE STUDY

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Résumé

Le confort des occupants dans un environnement intérieur inclut non seulement les qualités thermiques de l'espace mais s'étend à d'autres attributs de performance spatiale. Par exemple, la performance acoustique des pièces intérieures est perçue comme ayant un effet direct sur les niveaux de productivité des occupants de l'espace. L'intégration d'un système de verdure tel qu'un mur vivant peut être une stratégie d'isolation acoustique possible. Le but de la recherche actuelle était d'obtenir des mesures acoustiques in situ à quatre endroits avec des murs vivants, et d'évaluer leur potentiel global d'isolation acoustique. De plus, l'étude comprenait une évaluation du potentiel d'intégration du mur vivant à la galerie Paul Cocker, au département des sciences de l'architecture de l'Université Ryerson. Les résultats ont démontrés que les chutes d'eau, les pompes et les systèmes de ventilation des modules de mur vivant généraient des niveaux de bruit élevés. Les résultats de l'étude de cas sont présentés dans cet article.

Mots clefs : Conception durable; acoustique intérieure; systèmes de verdure verticale; simulation acoustique.

Abstract

Occupant comfort in an indoor environment includes not only thermal qualities of the space but expands to other space performance attributes. For instance, acoustic performance of indoor rooms is seen to have a direct effect on the productivity levels of the occupants of the space. The integration of greenery system such as a living wall can be a possible sound absorption strategy. The aim of the current research was to obtain in situ acoustic measurements at four locations with living walls, and evaluate the overall potential of sound absorption of these living walls. In addition, the investigation included an assessment of the potential of integrating the living wall within the Paul Cocker Gallery, at Ryerson University's Department of Architectural Science. The results showed that the living wall modules generated high levels of running waterfall, and pump and fan system noise. The results of the case study are presented in this paper.

Keywords: Sustainable design; indoor acoustics; vertical greenery systems; acoustic simulation.

1 Introduction

Buildings are constructed essentially to provide an indoor environment entirely separated from the outdoor atmosphere, creating an enclosure that caters for the well-being of its occupants. Indoor enclosed spaces are a function of the construction assemblies and the enclosed volume within. The current 'green' and 'sustainable' industry incorporates factors that are an index to comfort of the occupants and users of the space. Therefore, the various aspects of building physics are integrated in the design of buildings that provide energy savings in their construction and occupation phases. Previous studies evaluating the effect of indoor environment in terms of human comfort and their work performance demonstrated a significant reduction in work performance of the occupant due to their discomfort within the space [1]. Comfort for occupants of the space extends beyond thermal qualities. Where other aspects come into consideration, the indoor air quality (IAQ) extends to encompass a broader range of attributes of a space, attaining an overall indoor environment quality (IEQ).

A significant index of the indoor environment quality for occupant comfort and productivity levels is the acoustic performance of the space [2]. The acoustics of a space is influenced by airborne and structure borne sounds, transmitted from outdoors and adjacent spaces. The requirement for noise control strategies is recognized in sustainable building design, in addition to the more common parameters of air quality and thermal control [3].

The necessity of attaining well-defined acoustic qualities pertains to the tasks carried out within the space, to avoid disruption of concentration of users or the undesired transmission of conversation. Additionally, noise levels above a certain threshold could lead to discomfort, and with longer exposures and higher noise levels, a possible partial or total hearing loss [2].

Current practices incorporate living walls within interior spaces for their indoor air quality properties. Studies have shown that these living walls can also be utilized for sound absorption, where they have the ability to absorb some of the noise within the space and reduce the overall sound levels [3].

Living walls are generally constructed as panels of geotextile felts with pre-cultivated plants, which are fixed to a vertical support or on the wall structure. A variation of this

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panel form integrates a module box with a substrate, structurally held onto the wall.

The focus of this paper, therefore, is to evaluate the effectiveness of the integration of living walls within interior spaces to improve the acoustic comfort for the occupants. In-situ measurements of four spaces with installed living walls were carried out to determine the acoustic conditions. Living walls were, then, integrated in a 350 cu.m Paul Cocker Gallery space located in the Department of Architectural Science building of Ryerson University in Toronto. Acoustic simulations were used to assess the potential of applying living walls as a passive sound absorption mechanism. Absorption coefficients of the living walls as well as that of the gallery envelope space were obtained from the literature. The results of the measurements and acoustics simulations are presented below.

2 Background

The necessity of noise absorption in interior spaces is linked with occupant comfort within the space. Noise levels, above a certain threshold, has been shown to reduce people's efficiency in carrying out tasks, as well as affect the sense of balance, raise blood pressure and reduce blood flow volume, as verified by laboratory studies [3].

Surfaces within an internal space would absorb, reflect or transmit sounds. Lower absorption coefficient values of the room surfaces results in higher sound pressure levels in the space, and longer reverberation time, affecting speech intelligibility and sound perception [4]. Hence, criteria have been set to create a guideline for architects and engineers to follow when designing spaces when acoustic performance is considered.

2.1 Acoustic comfort

In order to determine the acoustic comfort within interior spaces, the criteria and guidelines are considered according to the typology of the spaces and the tasks carried out in them. Design values to be achieved in different spaces, through the use of Noise Criterion, Room Criterion, and weighted sound pressure levels dBA and dBC are available in the literature [4]. The following acoustic metrics were used to evaluate the suitability of living walls.

Room criteria contours

The Room Criterion curve (RC) defines the background noise level within a space using a single number, determined from the measured octave-band sound pressure levels. RC curves provide the character of the sound in addition to the single number, defining the sound within a room as either rumbly or hissy, if they fall within the range shown on the RC graph [4]. In addition, RC Contours include low frequency bands (16 Hz and 31.5 Hz) so as to determine the impact of noise induced vibrations.

Reverberation time

Reverberation time of an enclosed space is the time that it takes the measured sound pressure level to decrease by 60 dB. It defines the level of acoustic absorption within the space [5]. The optimum reverberation time values, from Reference 6, at 500 Hz is shown in Figure 1, based on the volume and the use of the given space.

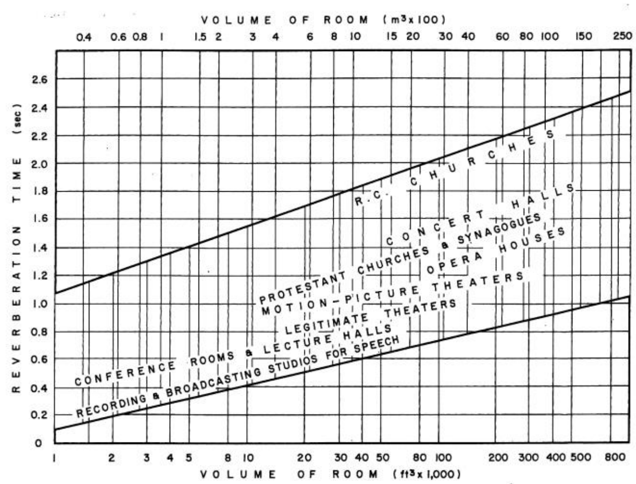


Figure 1: Range of acceptable reverberation time [6]

Clarity

Clarity and Reverberation time have an inversely proportional relationship, where increasing reverberation time would lead to a lower clarity value, and to obtain a higher clarity, the reverberation time has to be decreased.

Clarity (C80) is defined as the difference between the sound energy received at a listener in the first 80 milliseconds and the reverberant energy in dB [5].

An acceptable range of value for clarity is between –3 to +10 dB, where increasing the reverberant energy could lead to a decrease in clarity (more negative value) and thus decreasing the definition of sound and obtaining a "muddy" sound [5]. It is essential to maintain the design RT values to ensure that the sound reaching the listener has a well-defined clarity.

2.2 Living walls

Vegetation can reduce sound levels through the reflection, scattering and absorption by plant elements. Azkorra et al. have shown that the sound absorption of living walls is dependent on both the soil (substrate) and the vegetation itself [7]. The vegetation rooting and the presence of fibers leads to an acoustically very soft soil, due to the porosity created by the plant elements, hence, significantly influencing the absorption properties of the soil. With greater vegetation coverage, the absorption coefficient of the wall increases with increasing frequencies. Therefore, it can be determined that the substrate (soil) performs well in low frequency by absorbing the acoustic energy, and the plants perform better in high frequencies through scattering the sound [8, 9].

Azkorra et al. evaluated a modular based living wall system, where measurements were carried out in a reverberant chamber and the sound absorption and sound reduction index were calculated. The measured sound reduction index ranged from 9.7 dB to 17.1 dB across one-third octave frequencies centered at 100 Hz to 5 kHz. The sound absorption coefficient ranged between 0.35 and 0.51 across the frequency bands.

Experiments carried by Wong et al. conclude that the sound absorption coefficients of living walls under investigation are higher than those of other building materials, therefore, representing an enhanced noise attenuation mechanism [9].

Davis et al., tested living wall modules solely with substrate and densely planted with ferns and showed that the most prominent outcome found was the increase in sound absorption caused by ferns for frequencies higher than 400 Hz [3]. The weighted random incidence sound absorption coefficient of the modules densely planted with ferns equaled 1.0.

Perez et al. evaluated the effect of sound absorption of two in-situ vertical greenery systems, a Green Wall and a Green Façade [10]. The Green Wall was a pre-cultivated modular based system, while the Green Façade was made with a 2-mm wire mesh parallel to the cubicle façade wall, located 25 cm away by means of metallic supports anchored to the wall. Their results agreed with those obtained by Wong et al. [9] and Azkorra et al. [7]. The acoustic performance of the two Vertical Greenery Systems (VGS) demonstrated different frequency spectra, where the Green Façade exhibited a profile much more irregular than the Green Wall.

The results obtained from the work of Horoshenkov et al. show that the absorption coefficient of plants is controlled predominantly by the leaf area density and angle of leaf orientation [8]. However, absorption coefficient for the living wall was not determined.

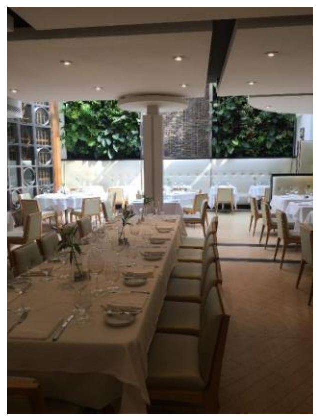
Lacasta et al. found the absorption coefficient of in-situ green walls to be measured at approximately 0.65, using an experimental prototype [11]. The intensity of vegetation density in the wall used was at an intermediate stage, and the noise absorption values demonstrated can be observed as average values.

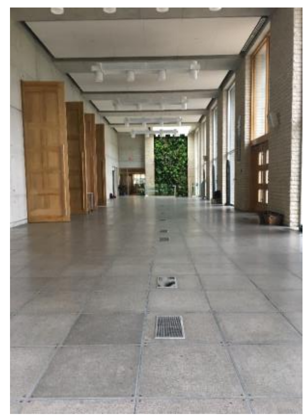
Kang et al. conducted a series of measurements in a reverberation chamber to examine random-incidence absorption coefficients and scattering coefficients of vegetation by considering soil depth, vegetation coverage and leaf size, and soil moisture content [12]. Outcome attained solidified findings of References 7, 8, and 9, concerning the absorption of acoustic energy by the soil and the scattering by the plant, in addition to the increase in coverage of the vegetation and its impact on the overall absorption coefficient of the living wall across the frequency range.

Fernandez-Bregon et al. assessed the effect on sound mitigation by measuring sound levels across a bare concrete wall and one with a living wall installed by fasteners onto the concrete block wall [13]. The results demonstrated that the average decrease in dB levels was around 2-8% compared to a bare concrete block wall, providing minimal acoustic benefits for the assembly.

3 Measurements

Acoustic measurements were conducted in four chosen locations with living walls. The photos of the four locations are shown in Figure 2. Each location was visited during hours with least or no occupants using the space, for the purpose of conducting measurements to evaluate the performance of the space without human interference. Background sound levels were conducted using a Quest meter. The different acoustic metrics such as reverberation time were measured using sine-sweep impulse responses with a Bruel & Kjaer dodecahedron speaker system. **NOTE:** It must be pointed out that the measurement of RT in the 125 Hz band was dependant on the existing background levels and hence it was not possible to obtain the RT values at some of the four locations.


Location 1


Location 2*


Location 3

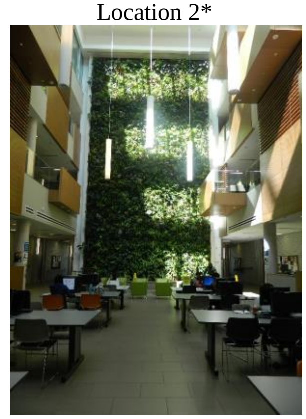

Location 4

Figure 2: Measurement locations. *NOTE: Location 2 - Living Wall – St. Gabriel's Passionist Church, Toronto, Ontario, Canada, The Passionist Community of Canada, Owner. Larkin Architect Limited, Architect.

Location 1 is a restaurant with the living wall installed in the dining room space. The floor area of the dining room is 176 m^2, and the living walls cover a total area of 20 m^2 of the end wall. An interior waterfall is integrated between the two portions of the living wall.

Location 2, the church common area (Northex of the St. Gabriel's Passionist Church), has a floor area of 246 m^2. A

21 m^2 living wall was built on an end wall. The living wall is not accompanied by a waterfall. The overall building strives towards sustainability in the built environment.

Location 3 is a hub for promoting healthy practices for the community. The 14 m^2 of the living wall is located in the main lobby atrium of the center, with a floor area of 150 m^2 on the main level. The living wall has an adjacent waterfall.

Location 4 is the largest of all locations in floor area and total covered living wall area, with 740 m^2 and 126 m^2, respectively. The living wall is installed in the atrium of a college library. The living wall soars through four floors of the library.

3.1 In-situ acoustic measurements

All the four living walls chosen for the current research have similar construction and installation techniques. The living wall is composed of a structure that is mounted on the constructed wall, in which layers of felt are attached. The plants are inserted into the growth medium, constitute the felt with a hydroponic mechanism. A water pipe system runs behind the wall and a basin positioned at the bottom collects and recirculates the water for a continuous flow to water the plants.

The plants used are composed of mainly the large leaf type, and a mixture of different species is utilized to resemble a naturally organic ensemble.

3.2 Results and discussion

Location 1

The layout of Location 1 and the measurement locations are shown in Figure 3 below.

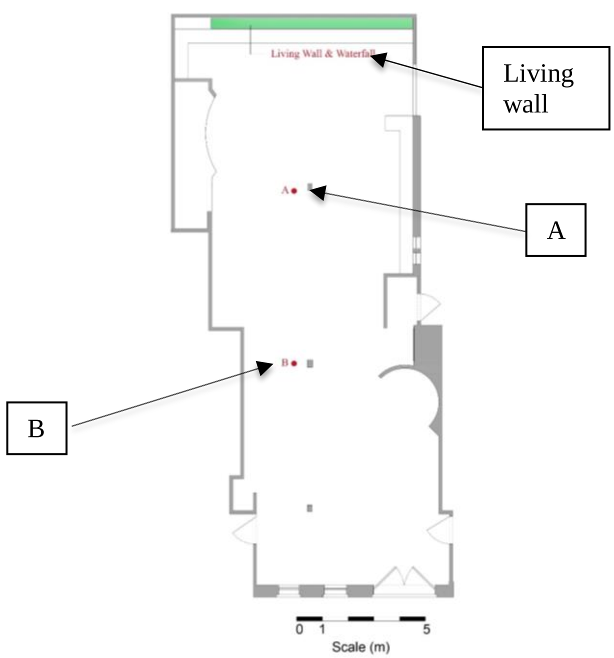


Figure 3: Location 1 floor plan.

The A-weighted average for the measured background noise level in the space at Point A is 53.7 dBA and Point B is 46.4 dBA. The impulse response test measurements were carried out with the microphone at Points A and B. Two sets of measurements were done for each point location.

The noise from the waterfall further affected the reverberation time measurements at lower frequency ranges. However, other than the lower frequency values, the reverberation time obtained from the measurements have reasonably close values to the recommended design values for the volume of the restaurant space. The reverberation results are sown in Figure 4 below.

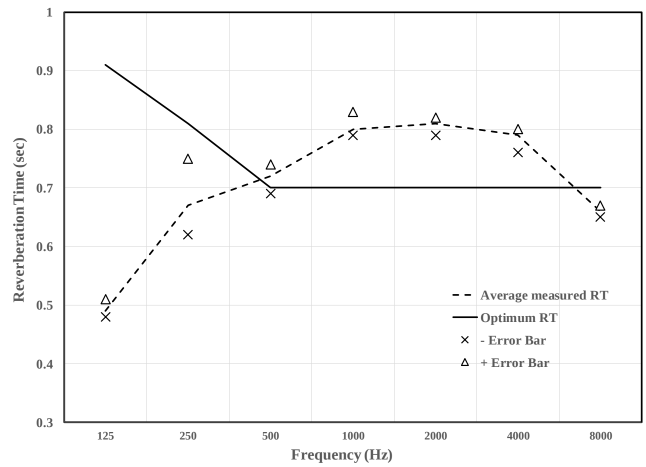


Figure 4: Reverberation time results for location 1

The obtained measurements of the clarity index (C80) show values that reach the extreme end of the acceptable range (-5 to +10 dB), where the values are 9.0 to 11.6 dB. This indicates that the quality of clarity of the sound reaching the listener is slightly "muddy".

The background noise levels across the octave band center frequencies measured in Location 1 are averaged and the RC number of the space was evaluated. The Room Criteria rating for Location 1 is RC-49 and is higher than the recommended RC-40 for a restaurant dining room space. The results are summarized in Table 1 below.

Table 1: Ambient sound levels of the living wall, location 1

Band Frequency, Hz	63	125	250	500	1 K	2 K	4 K
Location 1 SPLs, dB	61	56	48	46	45	42	34

The results of Table 1 show that the sound character is 'hissy' as the high frequency levels much higher than the acceptable RC-40.

The acoustic performance of the dining room is summarized in Table 2 below.

Table 2: Acoustic results of location 1

Acoustic Metric	Ambient sound	RT@500 Hz	C_{80}, dB
Condition	RC49 Hissy	0.7	9 to 12
Acceptability	NO	No	Yes

Location 2

The layout of Location 2 and the three measurement locations are shown in Figure 5 below.

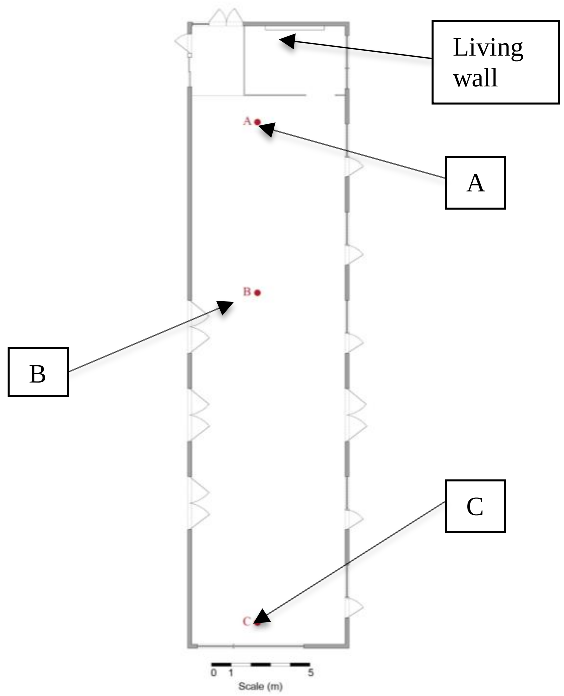


Figure 5: Location 2 floor plan - Narthex – St. Gabriel's Passionist Church, Toronto, Ontario, Canada, The Passionist Community of Canada, Owner. Larkin Architect Limited, Architect.

The background noise level was measured at 3 points (A, B and C) within the space and the average results are summarized in Table 3. The A-weighted average for Point A is 46.4 dBA, Point B is 45.4 dBA, and Point C is 45.6 dBA.

Table 3: Ambient sound levels of the living wall, location 2

Band Frequency, Hz	63	125	250	500	1 K	2 K	4 K
Location 2 SPLs, dB	56	52	47	42	40	38	27

The impulse response test measurements were carried out with the microphone at Points A and B. Two sets of measurements were done for each point location. The reverberation time results are shown in Figure 7. While the measured reverberation time of the space is higher than the optimum values for the volume and use identified for the space, the clarity of sound lies within the ideal range of -5 to +10 dB, verifying good speech intelligibility between the listener and the sound source.

As the living wall in this location was accessible for covering, two additional sets of impulse response testing were executed, to obtain values that facilitate the calculation of the absorption coefficient of the living wall. Quarter-inch MDF boards were used to cover 3.70 m^2 portion, more than 15% of the living wall's total area.

An increase in the reverberation time of the space is observed when covering a small portion of the living wall, justifying its effective acoustic absorption. Hence, it can be predicted that covering the entire wall or replacing its surface area would ultimately affect the acoustic performance of the space, further increasing the reverberation time and reducing the sound absorption.

Figure 6: Measurements with covered portion – Location 2

*NOTE: Location 2 - Living Wall – St. Gabriel's Passionist Church, Toronto, Ontario, Canada, The Passionist Community of Canada, Owner. Larkin Architect Limited, Architect.

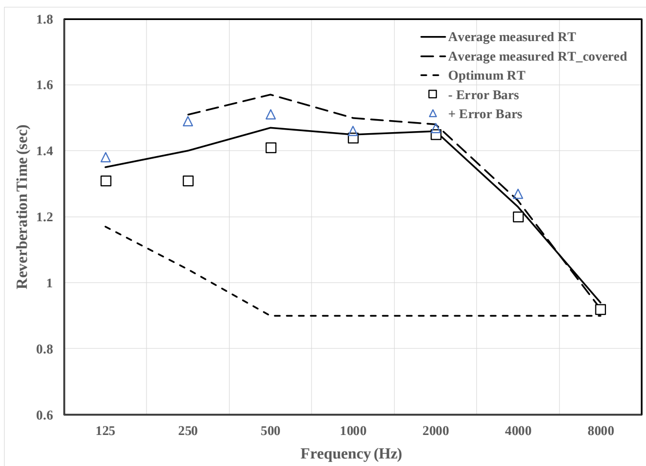


Figure 7: Reverberation time results for Location 2 (error bars for the uncovered case only are shown)

The Room Criteria rating for the church common area is identified as RC-42 and is much higher than the recommended RC-35 for this location. Because of the hard or sound reflective surfaces, such as concrete, glass, brick and floor tiles, the reverberation within the space is higher. The installed living wall area is not as significant when compared to the other surface areas, having only a minor effect on the acoustic properties of the space. The results of Table 3 show that the sound character is 'hissy' as the high frequency levels are much higher than the acceptable RC-35.

The acoustic performance of the lobby area is summarized in Table 4 below.

Table 4: Acoustic results of location 2

Acoustic Metric	Ambient sound	RT@500 Hz	C_{80}, dB
Condition	RC42 Hissy	1.45	-1 to 8.3
Acceptability	NO	➢ than 0.9 No	Yes

A portion of the living wall was covered with a hard, sound-reflecting board to assess the effect on the overall reverberation time due to the sound absorption of the living wall. The outcome demonstrated an increase in reverberation time in the experimentation with the covered portion of the living wall, verifying the sound absorption capacity of the living wall. However, due to difficulty covering the entire wall, the overall sound absorption of the living wall was not possible to determine.

Location 3

The layout of Location 3 and the three measurement locations are shown in Figure 8 below.

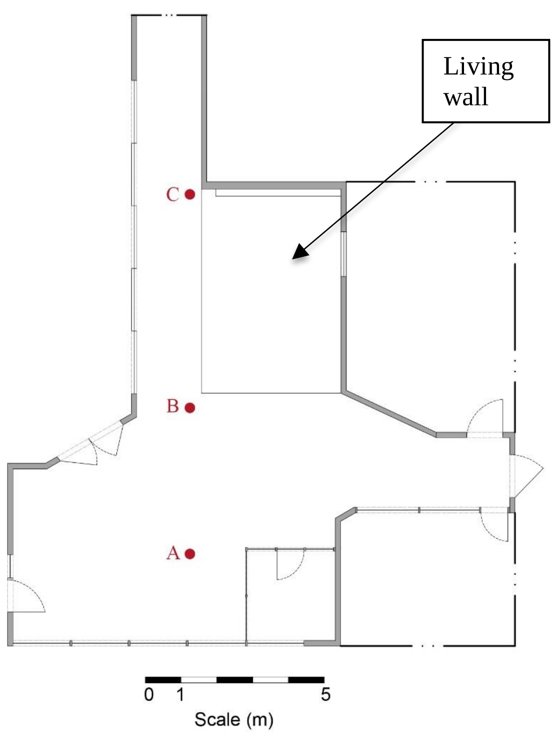


Figure 8: Location 3 floor plan

The background noise level was measured at 3 points on the main floor level and are summarized in Table 5. The A-weighted average for Point A is 56.1 dBA, Point B is 59.0 dBA, and Point C is 56.3 dBA.

Table 5: Ambient sound levels of the living wall, location 3

Band Frequency, Hz	63	125	250	500	1 K	2 K	4 K
Location 3 SPLs, dB	62	65	57	50	48	49	48

The impulse response measurements were conducted at Points B and C. Two sets of measurements were done for each point location. It is observed from the results of the acoustic measurements that the presence of the waterfall in this location led to high background noise level, affecting the measurements in the lower frequency range. The reverberation time results are shown in Figure 9. The clarity of sound within the space measured for the lower frequency ranges 125 – 500 Hz, demonstrate acceptable values, while the values in the higher frequency ranges of 2000 -8000 Hz exceed the ideal values significantly, causing the sound reaching the listener to be highly ‘muddy’ with minor intelligibility.

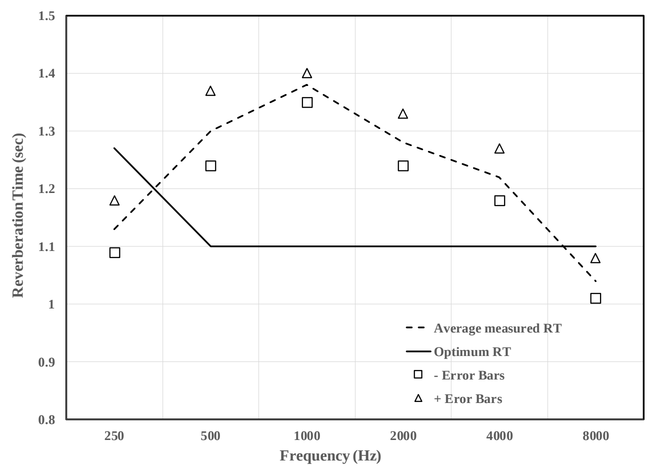


Figure 9: Reverberation time results for location 3

The Room Criteria is evaluated as RC-54 for this location, which is higher than the recommended RC-40. In this case, the effect of the waterfall had a significant impact on the results of the conducted measurements, raising the sound pressure level in the lower and higher frequency ranges much higher than the ideal values. The results of Table 5 show that the sound character is both ‘rumbly’ and ‘hissy’ as the low and high frequency levels much higher than the acceptable RC-40.

The acoustic performance of the lobby atrium is summarized in Table 6 below.

Table 6: Acoustic results of location 3

Acoustic Metric	Ambient sound	RT@500 Hz	C_{80}, dB
Condition	RC54 Rumbly & Hissy	1.3	-36 to 9
Acceptability	NO	➢ than 1.1 No	NO

Location 4

The layout of Location 4 and the five measurement locations are shown in Figure 10 below.

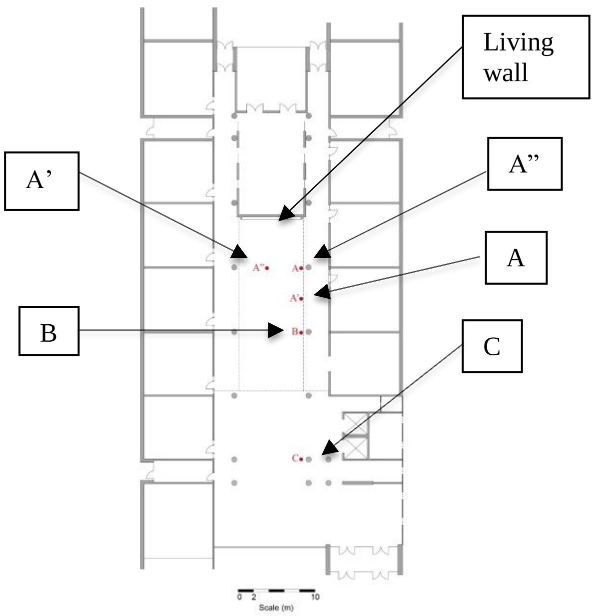


Figure 10: Location 4 floor plan

The background noise level was measured at 3 points within the ground floor area, Point A, B and C and are summarized in Table 7.

Table 7: Ambient sound levels of the living wall, location 4

Band Frequency, Hz	63	125	250	500	1 K	2 K	4 K
Location 4 SPLs, dB	54	48	46	45	41	38	34

Similar to the previous locations, the evaluated Room Criteria at RC-41 is higher than the recommended value of RC-30. The majority of the surface area of construction material are hard surfaces, or tile, concrete and glass, have a more significance impact on the overall acoustics of the space, when compared to the effect of the living wall. While the living wall in this location covers a total of 126 m^2 and its mechanical system causes an increase in the background noise, this leads to higher sound levels that need to be absorbed across the 740 m^2 floor area. The results of Table 7 show that the sound character is both 'rumbly' and 'hissy' as the low and high frequency levels are much higher than the acceptable RC-30.

The impulse response test measurements were carried out with the microphone at Points A, A' and A". Two sets of measurements were done for each point location. The reverberation time results are shown in Figure 11. The clarity values demonstrate higher values than the ideal design recommendation, although the average measured RT is lower than the optimum RT. As previously mentioned, having a lower reverberation time will lead to an increase in the clarity of the sound, however that is applicable to a certain degree. The ideal values of RT ensure that the sound reaching the listener has a well-defined clarity.

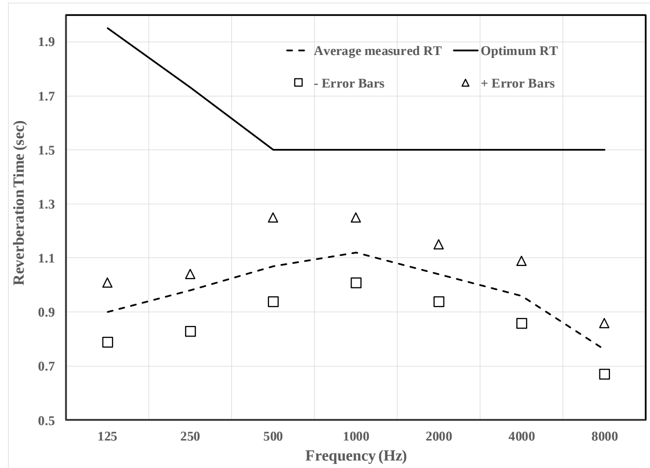


Figure 11: Reverberation time results for location 4.

The acoustic performance of the lobby atrium is summarized in Table 8 below.

Table 8: Acoustic results of location 3

Acoustic Metric	Ambient sound	RT@500 Hz	C_{80}, dB
Condition	RC41 Rumbly & Hissy	1.5	6 to 21
Acceptability	NO	➢ than 1.1 No	NO

The results presented so far for, the four locations, showed that the living walls, as applied, did not provide acceptable acoustic comfort to the four spaces. As a final exercise, the living wall was used as absorptive wall covering to a highly reverberant gallery space at Ryerson University to determine if the reverberation time can be adequately reduced to an acceptable level. The results are discussed below.

4 Application

The Paul Crocker Gallery, shown in Figures 12 and 13, is a multi-purpose space, located at Ryerson University's Department of Architectural Science. In addition to being used, primarily, as an exhibition space, the gallery is used as a 'Crit' space where the review of student's work takes place with about 20 people. The Gallery is used in the current case study, to evaluate the acoustic performance of the space to be used as a critique space, where speech intelligibility becomes a significant acoustic parameter to be achieved. The gallery is constructed with concrete ceilings, gypsum/plywood composite partition (with 4" airgap) walls with felt covers at the entrance, floor tiles, and three glass doors. The acoustics of the gallery varies with the density of occupants in the space, as the presence of people affect the performance of the room as well.

The objective of the acoustic simulation of the gallery is to increase the overall sound absorption of the space, ultimately reducing the reverberation time. The main acoustic metric applied for the gallery results is the reverberation time (RT).

Figure 12: Front view of Paul Cocker gallery

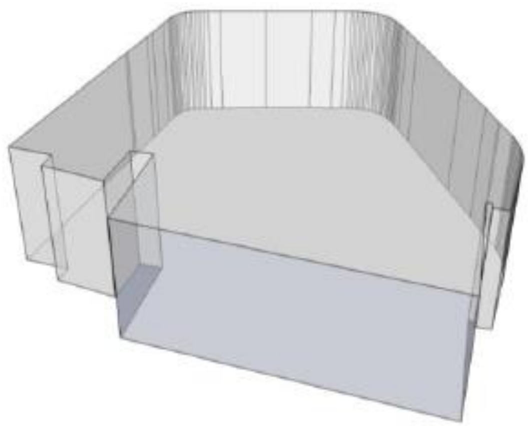


Figure 13: 3D sketchup model of Paul Cocker gallery

The acoustic absorption coefficient values for typical building envelope materials were obtained from Reynolds [14]. The absorption values for the living walls were obtained from Thomazelli et.al. [15]. The room acoustics software, 'ODEON' was used for simulations of the gallery [16]. The optimum reverberation time for the gallery space with a volume of 350 m^3, and requirements for high speech intelligibility would be around 0.5 seconds at 500 Hz.

The first simulation was undertaken to validate the bare gallery results to that of the measured reverberation evaluated with a dodecahedron source located near the central column seen in Figure 12. The absorption coefficient values used for the simulation, including that of the living wall (from Reference 15) are shown in Table 9.

The living wall was simulated within the model of the gallery to evaluate the installation area necessary to assess the prospect of using it as a passive sound absorption mechanism within an interior space, such as that of the gallery, to provide the required acoustic performance for occupant use. Three different trials were carried out, simulating various living wall areas installed within the space. Figure 15 below demonstrates the average measured reverberation time compared with the 3 trials (16.5 sq.m, 25.12 sq.m, and 45.94 sq.m of living wall) and the optimum reverberation time for the space. Figures 14 and 15 also include measured values of reverberation times using a sine-sweep signal and the Bruel & Kjaer dodecahedron speaker systems.

Table 9: Absorption coefficients of gallery surfaces

Band Frequency, Hz	63	125	250	500	1 K	2 K	4 K
Ceiling	.01	.01	.01	.01	.02	.02	.02
Floor	.02	.02	.03	.03	.03	.04	.07
Walls	.22	.4	.07	.06	.05	.3	.35
Glass	.35	.35	.25	.18	.12	.07	.04
Felt	.08	.08	.08	.3	.6	.75	.8
Living Wall	.01	.1	.1	.8	.9	.9	.9

The validation results are shown in Figure 14 below.

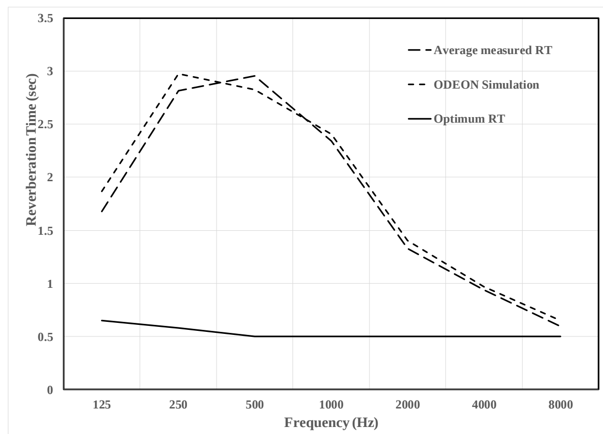


Figure 14: Validation of the bare gallery simulation

It can be seen from Figure 14 that the gallery simulation is satisfactory and additional simulations with the living wall can be undertaken with acceptable precision. The results also show that the gallery is highly reverberant across the frequency bands up to 4000 Hz.

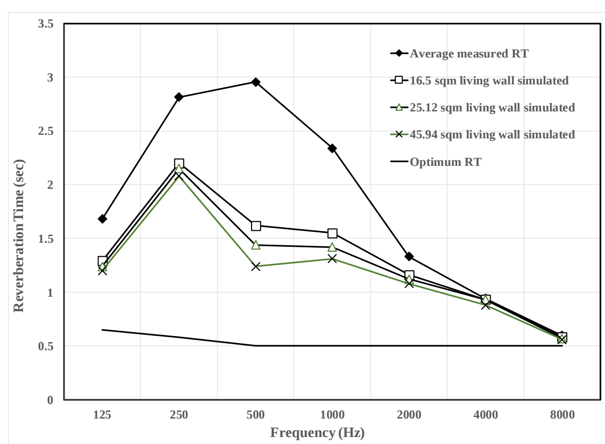


Figure 15: Reverberation time with three simulations.

The results of Figure 15 showed that the living wall area was not sufficient to provide the required reverberation time. Finally, the entire wall area of the gallery was covered with the living wall and the results are presented in Figure 16 below. Integrating the living wall within the Gallery space to provide acoustic comfort and high speech intelligibility is not achieved through the simulation, where the entire available wall area of 111.87 m^2 is covered with the living wall. The application in the gallery space requires an increased area of the living wall due to the already poor acoustic conditions of the space, where the existing wall surfaces were not sufficient. In addition, the average absorption coefficient of the living was not high across the frequency spectrum [15].

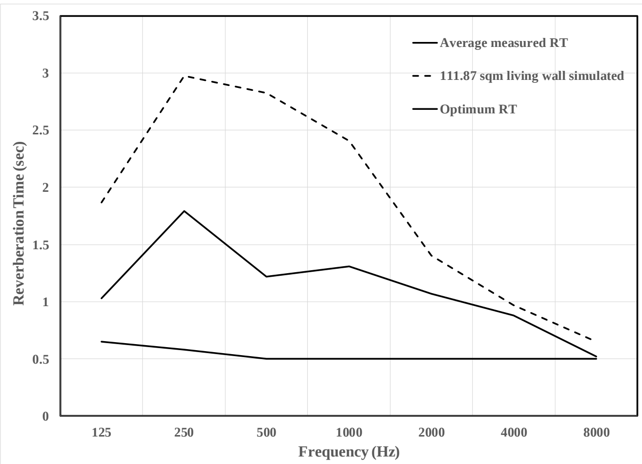


Figure 16: Reverberation time of the final Simulation

The final simulation, shown in Figure 16, used the entire available wall are of 118 sq.m. Even then the reverberation time of the galley did not meet the required optimum reverberation time.

5 Conclusion

The aim of this paper was to evaluate the potential of integrating living walls as passive interior absorption techniques to provide indoor acoustic comfort as the necessity of acoustic comfort reflects on the occupant productivity levels within the space [3]. The application of living walls was assessed through a series of acoustic measurements and experimentations carried out on in-situ living walls.

The absorption coefficient used for the simulations was obtained from the experimental results of Reference 15.

The RC rating evaluated for each of the four sites showed a higher value than the recommended design guideline for the spaces. The noise generated from the mechanical system of the living wall and the presence of the waterfall in some of the cases need to be attenuated, in order for the living wall to provide the acoustic comfort, and its sound absorption be at full potential.

The application of the living wall within the case study simulation did not achieve the desirable guideline acoustic parameters, which includes the reverberation time of 0.5 seconds to be achieved at 500 Hz within the space.

Acknowledgements

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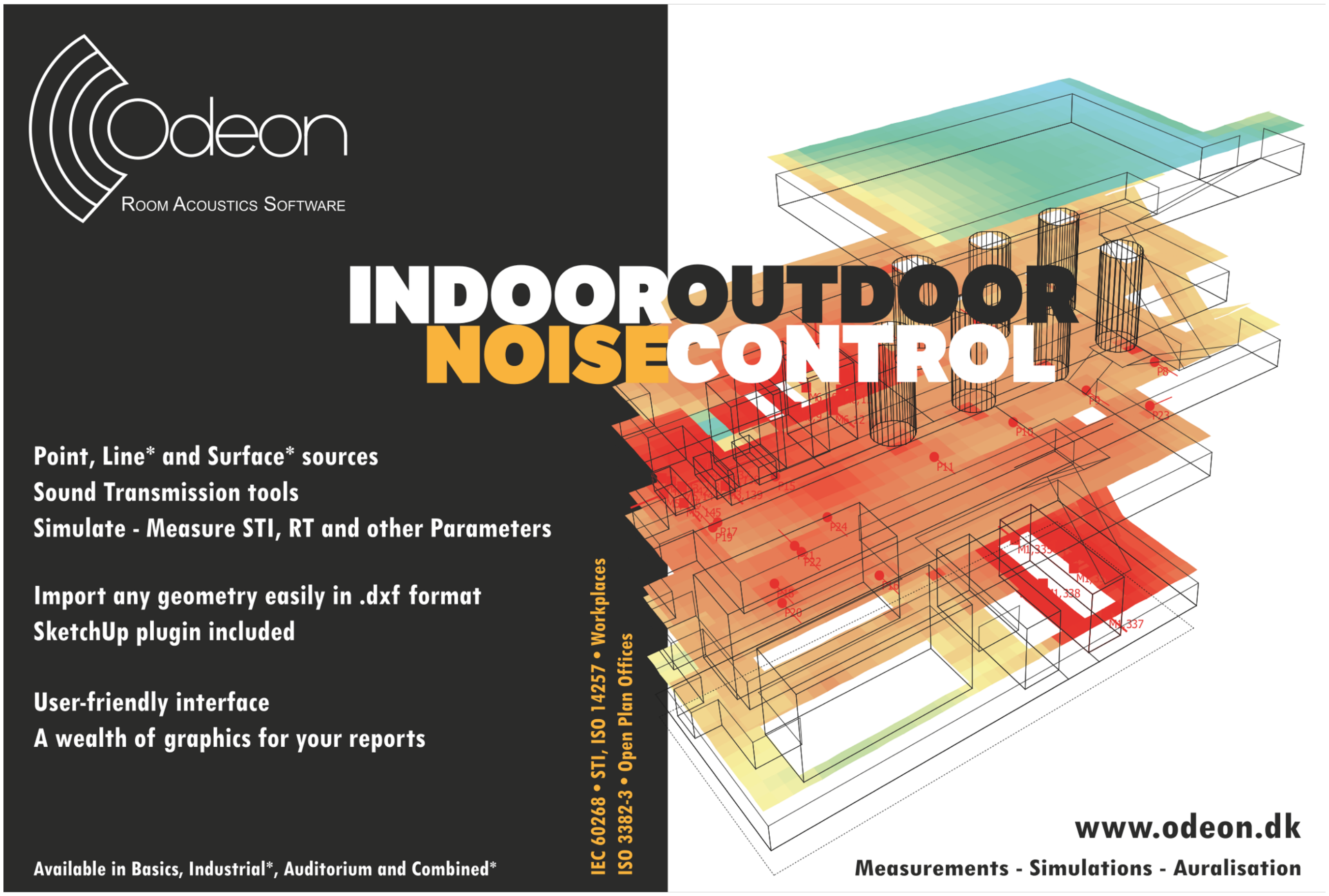

Experimental Evaluation Of Acoustic Characteristics Of Bale Straw Walls

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Résumé

Les fibres naturelles deviennent une alternative valable, à coût réduit, pour les traitements d'absorption acoustique. En effet, ces fibres présentent généralement de bonnes propriétés d'isolation et ne présentent aucun risque pour la santé. Pour isoler les bruits industriels, routiers et ferroviaires, des bottes de paille peuvent être envisagées comme matériaux d'isolation. Ainsi, cet article présente une recherche sur les caractérisations de murs acoustiques développés à partir des bottes de paille. Afin de protéger les fibres de paille et d'améliorer le comportement mécanique, une seconde variante de paroi en paille a été réalisée avec de la paille immergée dans du mortier de ciment. Les essais expérimentaux ont été menés dans un espace ouvert pour prédire les performances acoustiques et mesurer le coefficient d'absorption et le facteur de transmission. Les propriétés acoustiques sont déterminées expérimentalement pour des fréquences comprises entre 250 Hz et 4000 Hz. Les résultats montrent de très bonnes performances acoustiques pour la paroi de botte de paille, avec une absorption de plus de 70% du bruit incident pour les moyennes et hautes fréquences et un indice d'isolation atteignant 58 dB pour les hautes fréquences.

Mots clefs : réduction du bruit, fibre de paille, mur de botte de paille, mur anti-bruit, coefficient d'absorption, indice d'isolation

Abstract

Natural fibres are becoming a valid alternative to sound absorption treatments at a reduced cost. Indeed, these fibres generally present good insulation properties and present no health hazards. To insulate industrial, road traffic and railroad noises, straw bale can be envisaged as insulation materials. Thus, this paper investigates the characterizations of acoustic walls developed from straw bale. In order to protect straw fibres and improve the mechanical behaviour, a second variant of straw wall is made with bale straw immersed in the cement slurry. The experimental tests have been conducted in an open space to predict acoustic performances and measure absorption coefficient and insulation index. The acoustic properties are determined by experiments at frequency bands ranging between 250 Hz and 4000 Hz. Simulation shows excellent acoustic performances for the straw bale wall, with the absorption of more than 70 % of the incident noise for medium and high frequencies and an insulation index that reaches 58 dB for high frequencies.

Keywords: sound reduction, straw fibre, straw bale wall, noise barrier, absorption coefficient, insulation index

1 Introduction

Protection from the negative effects of the road traffic noise, and with the consideration of the ever-evolving of roads in cities, has become more and more needed. Indeed, one of the most common sources of noise pollution affecting the quality of life of residents near a road is traffic [1]. For this purpose, two solutions can be considered: (i) the improvement of the acoustic insulation of buildings and (ii) the creation of acoustic barriers. The latter is an effective solution to reduce traffic noise. A large amount of research is interested in predicting their performance and developing more effective barriers. Many researchers [1-4] determine the acoustic performance of noise barriers, basically their insulation and absorption, by in-situ measurements.

The improvement of an acoustic barrier is essentially based on two principles: improving the acoustic performance of the materials and using new shapes.

To improve the acoustic performance of the barrier, many materials with high acoustic performance can be envisaged. Although concrete, wood, plastic or glass panels and earth berms are generally used to create noise barriers, a large number of absorbent materials made from synthetic fibres are acoustically efficient, but their use is expansive and can create environmental problems.

If the cost of synthetic products and the impact of their use on the environment and health is high, natural materials represent a very interesting alternative [5]. Among all natural materials that can be used in construction, the production of the natural fibres has a low environmental impact, a low level of emitted pollution and a low embodied energy [6]. In addition, natural fibres are characterised by a good sound insulation performances [7], and consequently,

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they may be considered in solution for noise control elements in a wide range of applications.

In addition to their cheap price [8], natural fibres are abundant, renewable and biodegradable [9]. Many studies have shown that natural fibres, retained from agricultural production, are acoustically similar to traditional porous materials and generally have good acoustic absorption properties at medium and high frequencies, while the acoustic absorption coefficient at low frequencies is weak [8], [10-11].

By consequence, materials such as coir, kenaf, bamboo, tea-leaf and straw fibres have been qualified as suitable as raw materials for acoustic solution [12-13].

Indeed, some research has found that tea-leaf fibre present better absorption properties than polypropylene and polyester fibre. And that compressed coconut coir fibre sheet is a high sound absorbent [8]. Oil palm fruit fibre has shown a good sound absorption coefficient for frequency 2500Hz and above [9].

Other studies conducted on straw bales building have shown that the weighted apparent sound reduction index in the range 42-53 dB.

However, many works mention that long term performance can be reduced due to the problems that may arise, such as moisture damages, fungi, parasites and fire hazards [22], [9]. Therefore, a straw treatment is recommended in order to improve its fire resistance and avoid other types of degradation.

One of the recommended treatments is the Alkali one (usually with KOH or NaOH) [23]. But also cement, with his alkaline characteristics, and its ability to reduce the amount of oxygen in the straw bale, it can improve the mechanical characteristics on the one hand, and improve the resistance to fire, fungi, parasites on the second hand. Thus, it can be used as treatment.

In this work, we are interested in the characteristics of acoustic walls developed from both treated and untreated straw bales. Thus, a treatment of straw bales is manually made with cement slurry. Then, in-situ tests are carried out to determine the characteristics of the sound reflection and the transmission of the built walls. An old and rough concrete wall, a hollow brick wall and a stone wall are considered as reference walls. For each, we associate two walls of straw of the same thickness, one treated and the other untreated.

The in-situ tests conducted in an open space allows to predict acoustic performances and to measure the absorption coefficient and the insulation index.

2 Materials

In this paper, oats straw is used. The straw bale has a dimension of 35 cm × 45 cm × 115 cm. The apparent density of the bale is about 120kg/m^3. The individual straw fibre diameter is lower than 50 mm, a cut on a straw fibre is shown in Figure 1.

The three reference walls are made from concrete, hollow brick and stone, and have thickness of 17, 19 and 45 cm respectively. In addition, six walls are constructed, three with untreated straw and three with treated one (Figures 2, 3, 4).

To build the treated straw walls, moulds with specified dimensions made out of wooden strips are constructed and cement slurry is prepared with Portland cement and water. The straw is, then, cut into the desired dimension, immersed in the slurry and then raised and fitted manually in the mould until the frame space is completely filled. The wooden frames are essential not only for constructing the wall with the right dimension but also to fix the wall during drying period in his place. After drying, the walls are ready to be used.

The used slurry is composed by one third of his weight of cement and two thirds of water. The untreated straw wall is constructed following the same procedure, but without immersing the straw in the slurry.

Figure 1: A cut on a straw fibre

Figure 2: References walls

Figure 3: Treated straw wall

Figure 4: Untreated straw wall

Table 1: The superficial density of straw walls (kg/m^2)

Thickness of wall	Untreated straw	Treated straw
17 cm	10.71	13.71
19 cm	12.35	16.85
45 cm	51.75	66.75

The superficial density of each straw wall is shown in Table 1. It should be noted that the construct process of the 17 and 19 cm thick walls, in which one is forced to open the straw bale, caused a decrease in its apparent density.

3 Method and setup

3.1 Introduction

Different methods exist to measure the acoustic characteristics of a wall, in the laboratory as well as in-situ. The results of the in-situ and laboratory measurements are generally comparable but not identical. However, many studies suggest a correlation between the two measurement methods results.

In the laboratory, sound absorption measurements are generally performed using the impedance tube method with the use of small samples.

In the case where a small sample does not represent the wall (wall of great thickness as an example), which does not allow to obtain the real characteristics, the in-situ methods remain the most adapted to obtain reliable characteristics.

For in-situ characteristics, many methods exist, and the must used method in literature is the impulse measurement methods. These methods require to measure, previously, the direct sound under free-field conditions, and to use the Fourier transformation to find the values of an equivalent real noise.

Theoretically, every method, capable of separating incident from reflected waves, can be used to measure the walls characteristics [24].

This is why, in what follows, we will determine the amplitude of the incident, reflected and transmitted waves without using the Fourier transformation. This will determine the desired acoustic characteristics.

3.2 Acoustic measurement setup

When a sound wave is incident on a wall, a part of the energy will be absorbed, another will be reflected and a part will be transmitted by the material (Figure 5).

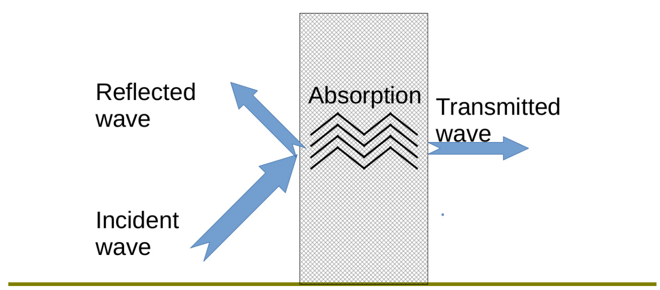

Figure 5: Sound wave paths in the presence of a wall

The acoustic performance of this wall is mainly determined by its acoustic insulation and its acoustic absorption. In fact, a good acoustic insulation permit to minimize transmission of sound through the wall, and a good acoustic absorption permit to minimize reflection of sound at the wall.

Absorption and insulation can be characterized by the absorption coefficient and the insulation index, which characterizes the material of each wall.

To determine the characteristics of the straw-based walls, in-situ measurements of sound reflection and insulation were performed.

The sound tests are carried out using an acoustic system comprises of a "BEHRINGER C-1U" omnidirectional microphone, a commercial loud speaker and a decibel meter application in the computer by the measurement of the sound intensity in a series of points as shown in Figure 6 for absorption test, were three positions for the microphone are chosen between the source and the wall.

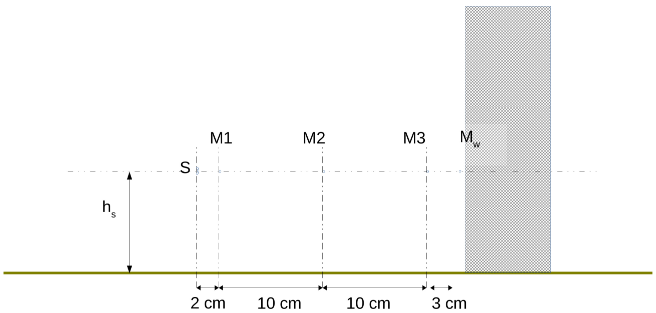

Figure 6: A schematic diagram of the measurement setup for absorption test.

For insulation test, two points are considered, one just before the wall and the other just behind (at 1 cm of the wall) as represented in Figure 7.

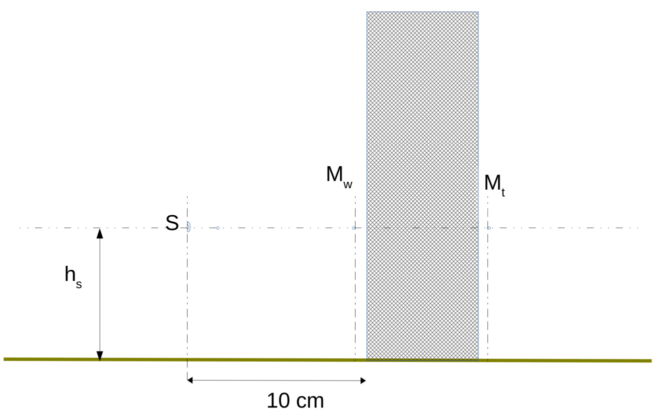

Figure 7: A schematic diagram of the measurement setup for transmission test.

All measurement points are in the same horizontal axis, located at a height h_s.

Low frequencies (250 Hz), lower middle (500 Hz), upper middle (1000 Hz), and high (2000 and 4000 Hz), were selected to be generated via the sound source. Using the microphone, we measure the incident noise, the background noises, transmitted noise and reflected noise by the wall.

For this in-situ octave band analysis, we assume that different sounds arriving at a point of measurement combine incoherently. The very small uncertainty, obtained in the results, between different measurement points, confirms this hypothesis.

Considering by I_i the sound intensity at the receiver location, the sound intensity level, L_I, is defined as in equation 1 ([25]).

$$L_I = 10\, Log_{10}\left(\frac{I_i}{I_0}\right) \tag{1}$$

Where I_0 represents the reference intensity, $I_0 = 10^{-12}$ Wm^{-2}.

Thus, in a point M_i before the wall, we have a combination of different sound intensities: the background intensity I_{b_i}, the direct intensity I_{d_i} the reflected intensity I_{r_i} and the ground-reflected intensity I_{g_i}.

The background intensity

The value of the incident intensity level is obtained by considering the measurement positions in a location free of reflections coming from nearby walls, and the background noises will be noted in all the measurement positions. For the rest of the paper, the value of the incident intensity will designates the measured value of the incident intensity with subtraction of the value of the background intensity.

The direct intensity

Incident intensity I_{in} can be written as the sum of direct and reflected from the ground intensities. Since at each point M_i, the ground-reflected intensity is closely related to the direct intensity, we can write the following relation:

$$I_{in_i} = Q_i I_{d_i} \tag{2}$$

Where Q_i is a correction factor which depends on the frequency of the sound, and the height of the point M_i from the ground.

In a location free of reflections coming from nearby walls, a first series of measurement, where the height of the measurement axis is varied, has been carried out. The figures 8 and 9 show that from a height of 65 cm, the effect of the ground becomes negligible.

Thus, using the equation 3, we can calculate the Q factor at any measuring point before the wall (for 45 cm of height). The results are shown in the Table 2.

$$Q_i = \frac{10^{\left(\frac{L_{I_i}}{10}\right)}}{10^{\left(\frac{L_{I_{i0}}}{10}\right)}} \tag{3}$$

Where $L_{I_{i0}}$ represents the sound level without ground effect (i.e. to a height of 85 cm) and L_{I_i} the sound level at the height of 45 cm.

Two remarks are noted: the first is that the values for the point M_1 are very close to 1, what is expected as the point is near the source. The second remark is that for points far away from the source, the values are greater than 3, this is due to the type of sound source that is not omni-directional.

The correction factor Q_i calculated, the equation 3 will be used to calculate the direct intensity and eliminate the soil effect.

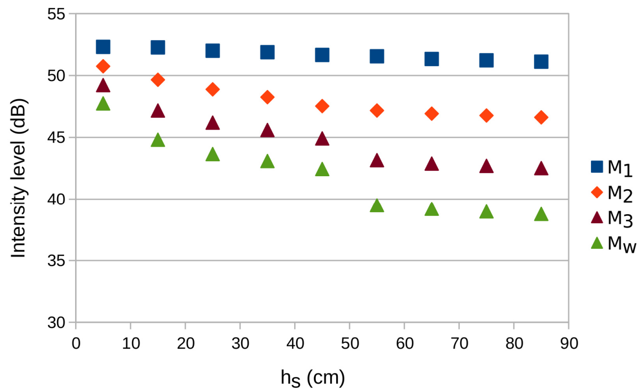

Figure 8: Effect of the ground-reflection in the intensity levels for 250 Hz.

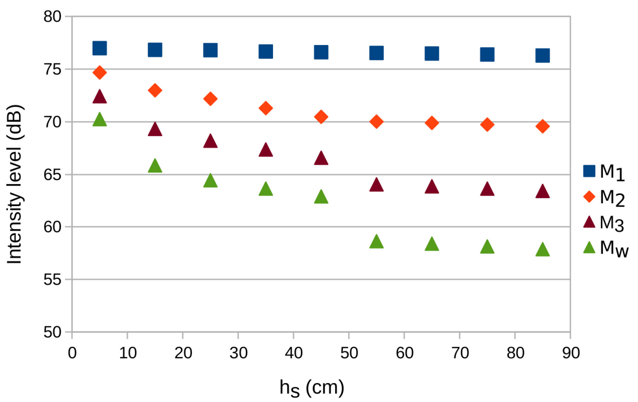

Figure 9: Effect of the ground-reflection in the intensity levels for 4000 Hz.

Table 1: Correction factor for different frequencies.

Frequency	M_1	M_2	M_3	M_w
250	1.133	1.233	1.737	2.328
500	1.153	1.264	1.840	2.534
1000	1.134	1.264	1.948	2.819
2000	1.115	1.259	2.026	3.044
4000	1.074	1.229	2.063	3.210

The reflected intensity

Between the source and the wall (whose characteristics are to be measured), we measure the sound level L_{I_i}. The effect of the ground reflections is assumed similar to the incident recordings (without wall). Thus, the subtraction of the two intensities should give the reflected intensity. The reflected intensity is deduced as shown in equation 4.

$$I_{r_i} = I_0(10^{(\frac{L_{I_i}}{10})} - 10^{(\frac{L_{b_i}}{10})} - 10^{(\frac{L_{in_i}}{10})}) \tag{4}$$

Where L_{b_i} is the background intensity level and L_{in_i} is the incident intensity level.

The sound absorption is calculated using the equation 5

$$\alpha = 1 - \frac{I_r}{I_d} \tag{5}$$

With I_r and I_d are the reflected and direct intensities at the surface of the wall (measurement point M_w). If I_d can be directly measured at the surface of the wall, the reflected intensities at the surface of the wall must be calculated using the recorded value at one point from M_1, M_2 and M_3 and using the equation 6 :

$$R_{r_i}^2 \cdot I_{r_i} = R_r^2 \cdot I_r \tag{6}$$

Where R_{r_i} represents the distance covered by the sound from source to the point M_i after reflection, and R_r is the distance between the source and the wall.

The transmitted intensity

The sound transmission factor τ of a wall is defined as the ratio of the two sound energies: transmitted by the wall to the incident upon the wall.

So, the transmission factor can be approximated using the equation 7,

$$\tau = \frac{I_t}{I_{d_w}} \tag{7}$$

With I_t is the transmitted intensity deduced from the intensity level measured in the point M_t behind the wall, and I_{d_w} is the direct intensity at the point M_w before the wall.

The insulation index R can be defined as:

$$R = 10\, Log_{10}\left(\frac{1}{\tau}\right) \tag{8}$$

Or more simply written directly as a function of the sound intensity level as:

$$R = L_{d_w} - L_t \tag{9}$$

4 Sound absorption measurement

4.1 The absorption factor

The intensity levels of sound at the various points in a free field without walls, and for different walls are measured. For the same frequency, the same sound intensity was used for all the walls. Raw data is available from the authors upon request.

The average of the measured level of background noise is 19.88 dB with a variation of no more than 0.9% throughout the tests.

The variation of the absorption coefficients as a function of the frequencies are plotted in the figures 10, 11 and 12. For all the walls, and for all the considered frequencies, the variations of the absorption coefficient between the three measurement points are lower than 0.8% in all cases.

The obtained results indicate that, like all natural fibres, treated and untreated straw have excellent acoustic absorption coefficients, especially at medium and high frequencies. We also note that the absorption coefficient of treated and untreated straw walls has very similar values. It is concluded that the effect of straw treatment on the absorption coefficient is minimal.

For low frequencies, the absorption coefficients of all walls, including reference walls, are very low. Indeed, the absorption coefficients are around 0.1 for the reference walls and 0.15 for the straw.

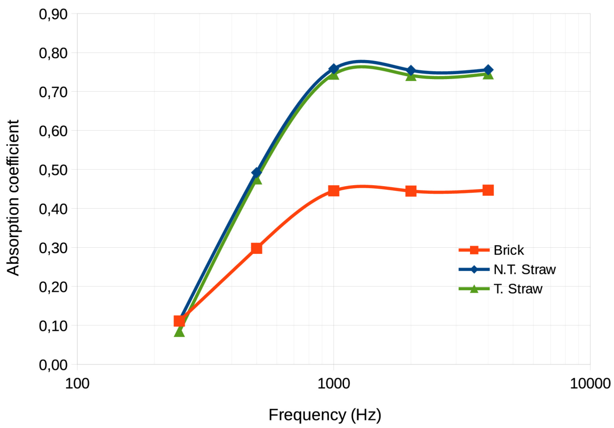

Figure 10: Absorption factor for 17 cm walls.

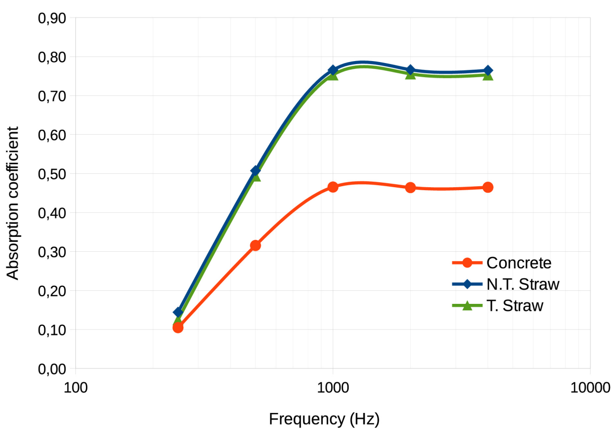

Figure 11: Absorption factor for 19 cm walls.

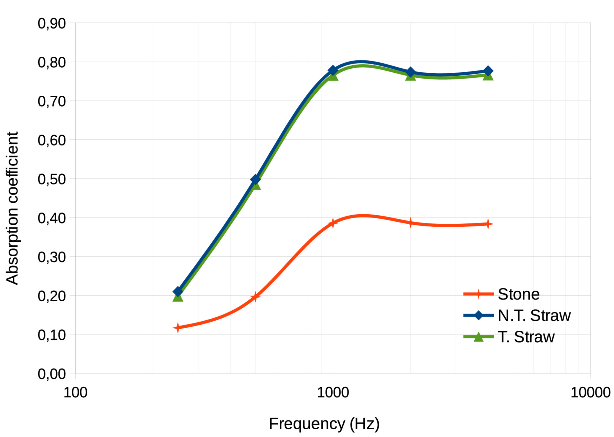

Figure 12: Absorption factor for 45 cm walls.

For the medium and high frequencies, the obtained results show that the absorption coefficient of the straw walls is much higher than that obtained for the reference walls. These last are characterized by a sound absorption coefficient which varies between 0.2 and 0.47 depending on the frequency and type of walls, while all values of the absorption coefficient of straw walls are greater than 0.5 for all frequency greater than 500 Hz. So, the obtained results show that, as all natural fibres, treated and untreated straw has excellent acoustic absorption coefficients at medium and high frequencies. Note that the obtained values for the concrete wall are very high compared to what is usually found in the literature; this is due to the degraded and rough state of the surface of the concrete wall.

The results of sound absorption measurements carried out on the straw by [10] using the impedance tube method are presented in the Figure 13. This last shows an identical shape curve to that obtained in this study. Contrariwise, values obtained in this study are lower than those obtained in [10]. In fact, if the curve of the Figure 13 shows a sound absorption value equal to 0.25 at 250 Hz, a peak value greater than 0.95 at 1200 Hz and reachs a value of 0.9 for 2000 Hz. The values obtained in this study varied from 0.1 to 0.3 for 250 Hz, show a peak value slightly lower than 0.8 for 1200 Hz and reach a value of 0.75 between 2000 Hz and 4000 Hz.

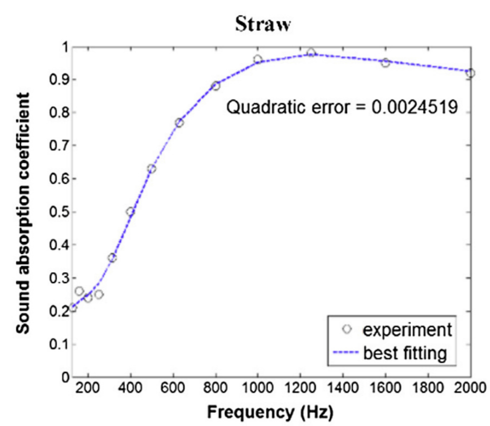

Figure 13: Sound absorption coefficient for straw [10].

4.2 The insulation index

Three measurements are considered for each frequency and in each point to calculate the insulation index *R*. The Figures 14, 15 and 16 show the insulation index for the different walls.

The experimental results indicated that at low frequency (250 Hz), the insulation index value was about 25 dB for the 19-cm-thick straw wall.

The insulation index values were higher for middle (f=500-1000 Hz) to high frequencies (> 1000 Hz). Thickest straw walls had a higher *R* value than other walls. The *R* value was greater than 25 dB, and it achieved a maximum of approximately 56 dB at a frequency of 4000 Hz for 19-cm-thick wall and 58 dB for of 45-cm-thick wall.

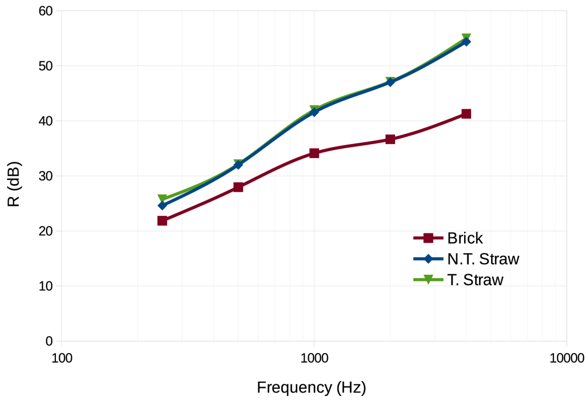

Figure 14: Transmission factor for 17 cm walls.

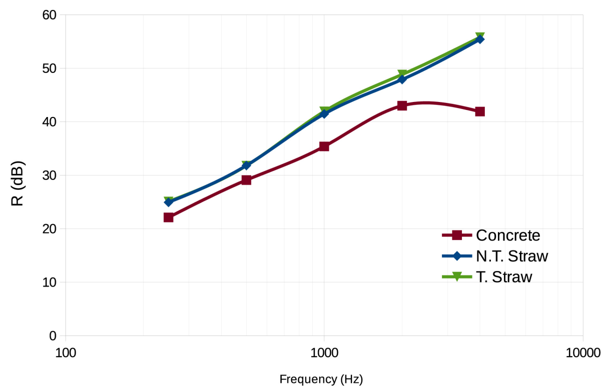

Figure 15: Transmission factor for 19 cm walls.

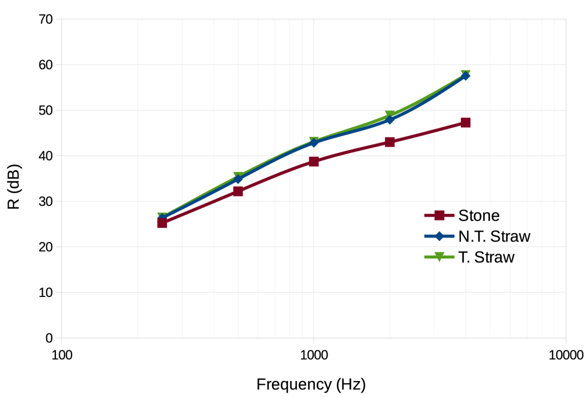

Figure 16: Transmission factor for 45 cm walls.

For low frequencies, the value obtained, although low, remains higher than that of the reference walls. For the frequency range between 1000 and 4000 Hz, differences with reference walls values in the range 5-13 dB are noted. We note that the effect of the thickness of the straw walls is small.

Similarly, we note also that the effect of the treatment of straw on the insulation index is negligible.

Compared to the reference walls, the straw walls are characterized by a higher insulation index. A difference that varies, depending on the frequency, between 4 and 14 dB with brick, 3 and 14 dB with concrete and 1 and 10 dB with stone.

5 Conclusion

In this paper, in-situ experimental tests have been conducted to study the acoustic properties of straw bale wall. Untreated and treated straw fibres are considered. The straw treatment is made with immersing straw in the cement slurry.

Through experiments at frequency ranges between 250 Hz and 4000 Hz, absorption coefficient and insulation index are determined. Experiences show that the acoustic behaviour of straw walls is similar to the known behaviour of vegetable fibres described in the literature.

We also find good acoustic proprieties with an absorption coefficient which reaches 0.8 and a value of insulation index which reaches 58 dB. So, the observed acoustic characteristics for the straw walls are much better than those of the considered reference walls.

Finally, we can see that the treatment of straw has a minimal effect on sound properties. Thus, treated or not, the bale straw walls absorb most of the sound and transmit only a small part of the absorbed sound.

All these properties can make straw walls, a good choice for as an acoustic barrier, essentially for its low cost, low environmental impact and acoustic performance.

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CANADIAN ACOUSTICAL ASSOCIATION

Minutes of the Board of Directors Meeting
Thursday, May 3, 2018 2:00 PM – 4:00 PM (Eastern Time)
Conference to be held online at: http://appear.in/caa-aca

1. Call to Order

Meeting called to order at 14h10 EDT.
Present online: Jérémie, Dalila, Bill, Andy, Hugues, Michael, Bryan, Roberto, Frank, Umberto, Joanna.
Excused: Mehrzad, Alberto.
Approval of agenda: Moved by Jérémie, seconded by Michael.
Approval of past minutes: Moved by Dalila, seconded by Roberto.

2. President's Report (Jérémie)

Follow up – committed to improve website and journal website; automated members webpage; automated inclusion of logo for sustaining subscribers; improved rendering on mobile devices; corrected problems on Online Conferences System re archiving old versions.

New business:

ASA conference – Stan Dosso will be able to attend the Minneapolis meeting and will promote the CAA interests regarding the proceedings, etc.

Discussion about proceedings: CAA members who have papers approved to ASA conference will be invited to provide 2-page articles for the conference issue (which will be December this year).

Deadline of 29 May is the same for ASA and CAA, because ASA approval is prerequisite for submitting a paper for the CAA journal.

There will be no awards presentation at the ASA/CAA conference – no opportunity in the program.

All CAA board members are invited to a dinner the day before the conference; the BoD meeting will be scheduled just before that event.

ICSV conference – Question from Jérémie about providing free advertising in JCAA for sponsors to the ICSV. General sentiment is that this would be unfair to the regular advertisers, so not endorsed by the Board.

AWC 2019 – Benjamin Tucker willing to organize the conference in Edmonton; waiting until after their Alberta conference in October to start working on the program. Last time we had a conference in Edmonton was 2003; Jérémie will send to Benjamin the list of organizers from that 2003 event together with a list of AB-based CAA members, as well as past reports from Halifax, Vancouver and Guelph conferences. Frank suggested reaching out to the Alberta organization of noise control engineers to engage them in the process (Bill volunteered to do that).

Sherbrooke museum – CAA logo and information to be displayed at the exhibition on sound and acoustics.

Claire Wakefield retiring – Need a replacement for him on the awards evaluation committee for Eckel award. Hugues will take up that role.

Murray Hodgson – Widow received the card and was very appreciative.

3. Social Media Outreach (Frank)

Activity continues under Frank's coordination; Frank is posting on Twitter (no longer on LinkedIn) with the assistance of Huywen who selects material. Need to find a replacement / complement for Huywen who will be finishing her postdoc. Suggestion to set up an organizational LinkedIn account since Frank was using his personal one.

4. Awards Report (Joana/Hugues)

Joana is taking over from Hugues, who has sent her all the information. Nothing has moved yet on the awards front for this year (no nominations or applications yet). Bryan asked what the approach is if an applicant is from the same lab as the coordinator; Frank suggested that Bryan should seek out an alternative adjudicator for the instance.

Updated online system for listing all past awards.

Dalila informed that all cheques have been sent for last year's awards.

5. Past and Upcoming Meetings

<u>AWC-2017 (Peter or Jérémie)</u>

Dalila presented information on behalf of Peter; extremely successful conference in numbers and revenue; Dalila lauded the personal involvement on the part of Peter and his family to organize and run the event. Budget of $71,000 for the conference but the revenue amply made up for the high cost. The report is due out shortly. General discussion on the value of good organizing and outreach for a successful conference.

<u>AWC/ASA-2018: Victoria, BC (Roberto and Jérémie)</u>

Roberto pointed out the rigid organizing of the ASA. Bill mentioned the disadvantage of not having exhibit/sponsor revenue, but Dalila pointed out that this year is a no revenue / no cost participation for the CAA.

<u>AWC-2019: Alberta (Frank and Jérémie)</u>

Benjamin Tucker now organizing event for Edmonton; Jérémie is providing material from previous conferences to assist.

Dalila raised question about AWC 2020; should be in the Maritimes for a good rotation; suggestion was made to consider St John's but Charlottetown would also be a good location. St John's has a better base for organizing committee. Dalila asked whether we should consider having the AWC only every two years, interleaved with other conferences; opinion is to stay with current formula but have a plan B in case the organizing thins out. Andy mentioned that UNB has an active underwater acoustics group.

<u>ICSV26-2019: Montréal, QC (Jérémie)</u>

Jérémie is organizing and all is well; CAA members will receive a discount.

<u>INTERNOISE 2021B (Roberto)</u>

Nothing to report; not aware of anyone really having taken the lead in Vancouver (last contact in July indicated that there was some general interest but no momentum). When asked earlier on, we had indicated that the CAA-

ACA would support through endorsement any initiative to have the event in Vancouver, but not make a monetary commitment to the event.

<u>ICA2025B (Jérémie)</u>

Jérémie is the point of contact. Group from NRC with Michael R. Stinson has asked for potential interest from the CAA to hold the event in Montreal. Suggestion from the Board is to hold back a decision until closer to the year. Jérémie will give a qualified expression of interest.

<u>Conference Manual – update (Frank)</u>

Frank is integrating an older version of the manual with new information; will share draft with Benjamin Tucker. ETA for review is next month.

6. Treasurer's Report (Dalila)

Association is in excellent shape after revenue from AWC 2017; investments to be considered soon; HST refund from 2016 should be forthcoming after closing fiscal year. Dalila asked approval to move $50,000 from operating account to investment account (to be rolled in with approval of report)

Dalila asked whether records older than 7 years should be kept, otherwise she will have all older records destroyed. Nobody objected to shredding older records.

Motion to approve report: moved by Roberto, seconded by Umberto; carried unanimously.

7. Secretary's Report (Roberto)

The current tally of Association members and JCAA subscribers is summarized in the table below, which shows by comparison the numbers for October 2017 and June 2017 reported at the corresponding meetings.

Category	**Paid-up 2018 (as of 3 May 2018)**	**Paid-up 2017 (as of 8 Oct 2017)**	**Paid-up 2017 (as of 1 June 2017)**
Regular member	152 (+42)	146 (+42)	145
Emeritus	1	1	1
Student	20 (+7)	14 (+7)	34
Sustaining subscriber	20	25	28
Indirect subscribers			
- Canada	3	6	6
- USA	1	4	4
- International	2	4	4
Direct subscribers	3	1	1
Total	**202 (+49)**	**201 (+49)**	**223**

8. Editor's report (Umberto)

All is fine on the Journal side; there is a healthy queue of accepted papers in the pipeline. June issue will be a mixed edition of a regional issue for Vancouver/BC and a regular complement of papers. September issue will be a regular one. December issue will be the conference edition. Next year there will be no regional issues but other forms of dedicated content are planned. Roberto mentioned the problem we faced with some potential contributors to the BC issue being concerned that a short publication on current research might jeopardize full publications later; Umberto noted that if the call for a regional issue results in interest in the submission of full papers instead, a positive goal has still been reached.

In Guelph there was a discussion of a "practitioners' corner" for less scientific content, proposed by Alberto; one such article was published, but Umberto is concerned that too many of these articles would lower the standing of the journal and is taking a cautious approach to this matter. Another problem is that practitioners are not used to the slow and rigorous process of editorial review and revision cycle, resulting in added burden on the editorial team compared to traditional research articles.

9. Varia

Roberto brought up the question from Ramani about student travel support for one of his students who will no longer be a student at the time of the conference. This issue opened the question of whether there will be CAA-ACA travel subsidies for the Victoria conference since it is primarily handled by the ASA and does not have a direct CAA budget. Frank suggested that for "shared" conference years we adopt the simple model to allocate a standard amount that is divided among all student applicants. Discussion to be continued in a separate session.

10. Next Meeting: Nov. 4th, 2018 @4pm(PST) in Victoria (BC)

Jérémie suggested to start at 3pm because it will give time to wrap up before the ASA dinner at 6:30pm.

11. Motion to Adjourn

Moved by Jérémie to adjourn at 16:03 EST.

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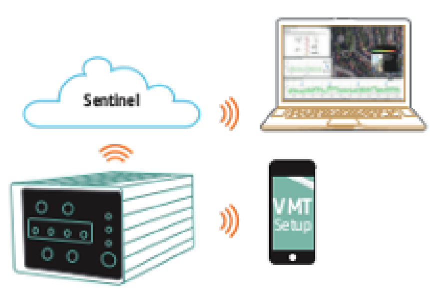

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AWC2018

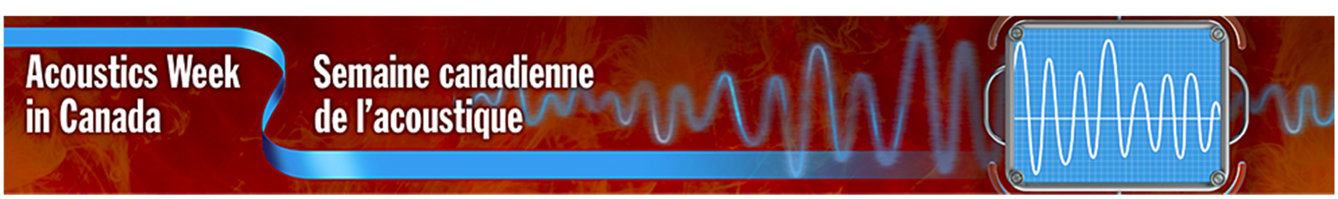
Joint AWC2018 and ASA 176th Meeting,
Victoria, BC, 5-9 November 2018

The 176th Meeting of the Acoustical Society of America (ASA) will be held jointly with the Acoustics Week in Canada 2018 of the Canadian Acoustical Association (CAA) in Victoria, BC, Canada, on 5-9 November 2018.

The conference will be organized by the Acoustical Society of America using their guidelines and procedures, while the Canadian Acoustical Association will organize some special sessions and handle its regular business and core activities, such as standards committee, student awards, etc.

For more information, visit http://acousticalsociety.org/meetings

To contact Dr. Roberto Racca, AWC2018 conference coordinator, please send an email to: conference@caa-aca.ca



AWC2018

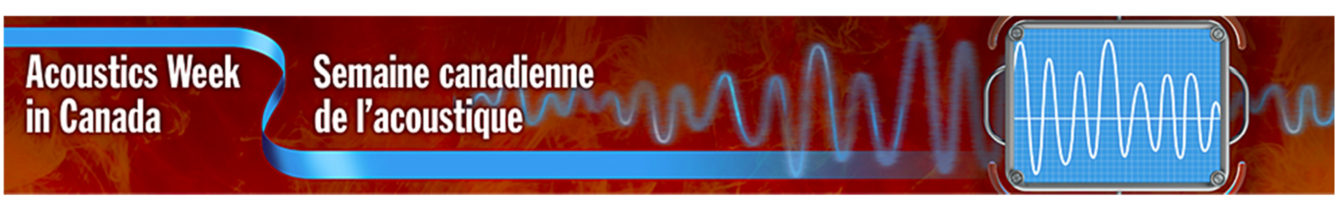
Congrès commun entre AWC2018 et la 176e rencontre de l'ASA, Victoria, C.-B., 5-9 novembre 2018

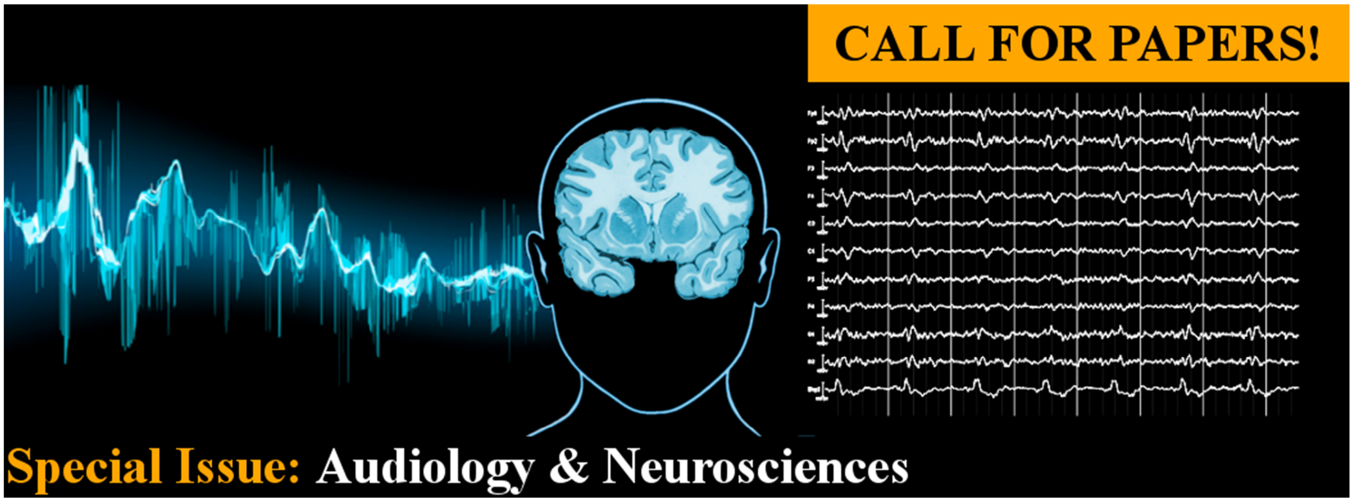
La 176e rencontre de l'Acoustical Society of America (ASA) se tiendra conjointement avec la Semaine canadienne d'acoustique 2018 de l'Association canadienne d'acoustique à Victoria, C.-B., du 5 au 9 novembre 2018.

La conférence sera organisée par l'Acoustical Society of America selon leurs méthodes et procédures, tandis que l'Association canadienne d'acoustique y organisera des sessions spéciales et tiendra ses rencontres régulières ainsi que ses activités propres, telles la rencontre des comités de normalisation, le programme de prix pour les étudiants, etc.

Pour plus d'information, visiter http://acousticalsociety.org/meetings

Pour contacter Dr Roberto Racca, le coordinateur du congrès AWC2018, merci d'écrire un courriel à : conference@caa-aca.ca





Acoustics is a broad subject matter that currently employs hundreds of us across Canada in fields as different as teaching, research, consulting and others. To reflect such diversity the Canadian Acoustics has been regularly publishing over the last 40 years a series of special journal issues to highlight thematic topics related to acoustics.

Therefore, the Canadian Acoustics journal is currently inviting submissions for the next special issue programmed for March 2019. **The focus of manuscripts submitted to this Special Issue may include (but are not restricted to) topics related to audiology and neurosciences such as:**

- Acoustics applications in electrophysiology (e.g. EEG, EOG, ECOG…)
- Hearing assessment (e.g. audiometry, DPOAE…)
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How to be part of it?

To contribute to these special "audiology and neuroscience" journal issues, authors are invited to submit their manuscript under the "Special Issue" section through the online system at http://jcaa.caa-aca.ca **before November 15th 2018.**

Each manuscript will be reviewed by the Canadian Acoustics Editorial Board that will enforce the journal publication policies (original content, non-commercialism, etc., refer to the Journal Policies section online for further details).

A unique special issue you want to appear in!

This special issue of the journal can be considered as a true directory for audiology and neuroscience in Canada. They will be published in hardcopies and sent to all CAA national and international members, while electronic copies will be made available in open-access on the journal website. The content of these issues will be entirely searchable and comprehensively indexed by scholar engines as well as by major internet search engines (Google, Bing, etc.). Authors are invited to carefully select their keywords to maximize the visibility of their articles.

If you have any questions, please contact Mr. Olivier Valentin (olivier.valentin@etsmtl.ca). To secure an advertisement for this special issue, please contact Mr. Bernard Feder (advertisement@caa-aca.ca).

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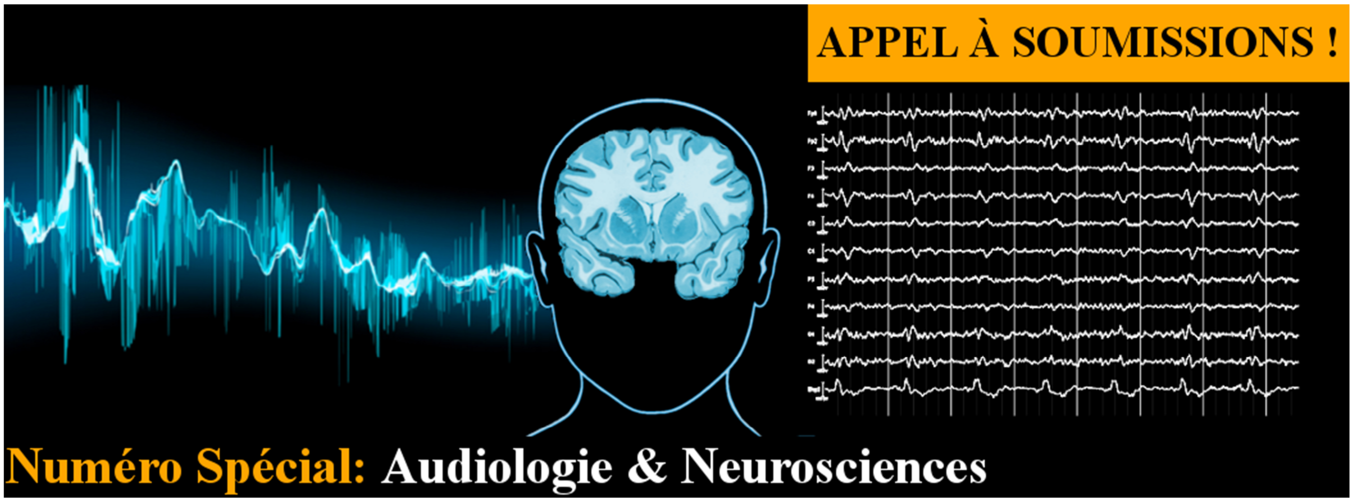

L'acoustique est un vaste domaine qui offre des centaines d'emplois à travers le Canada, et ce, dans différents secteurs tels que l'éducation, la recherche, la consultation professionnelle, etc... Afin de bien refléter cette diversité, l'Acoustique Canadienne a publié régulièrement au cours des 40 dernières années une série de numéros spéciaux pour souligner les divers champs d'applications de l'acoustique

L'Acoustique Canadienne fait donc un appel à soumettre une série d'articles pour le prochain numéro spécial planifié pour mars 2019. **Ce numéro spécial inclura principalement (mais ne se limitera pas à) des contributions dont le sujet est en lien avec l'audiologie et les neurosciences, tel que :**

- Applications de l'acoustique en électrophysiologie (EEG, EOG, ECOG, etc…)
- Évaluation de l'audition et de la surdité (audiométrie, DPOAE, etc…)
- Tout autre sujet en audiologie/neurosciences en lien avec l'acoustique.

COMMENT EN FAIRE PARTIE?

Pour contribuer à ce numéro spécial « audiologie et neurosciences », les auteurs sont invités à soumettre un article, sous la rubrique « Numéro spécial » dans notre système en ligne au http://jcaa.caa-aca.ca **avant le 15 novembre 2018**. Il est possible de soumettre un même article dans les deux langues officielles.

Chaque article sera révisé par le comité éditorial de l'Acoustique canadienne qui veillera à ce que les politiques de publications de la revue soient respectées (contenu original, contenu non commercial, etc. – voir les politiques de la revue pour de plus amples détails).

UN NUMÉRO UNIQUE DANS LEQUEL VOUS VOULEZ PARAÎTRE!

Ce numéro spécial « audiologie et neurosciences » peut être considéré comme un véritable répertoire à propos de l'audiologie et des neurosciences au Canada. Ils sont publiés en format papier et envoyés à tous les membres nationaux et internationaux de l'ACA. Une version électronique est aussi disponible en ligne sur le site internet de la revue. Le contenu de ces numéros est indexé, donc facilement trouvable au moyen de moteurs de recherche classiques, tels que Google, Bing, etc… Les auteurs sont invités à bien choisir les mots clefs pour maximiser la visibilité de leur article.

Pour toutes questions, vous pouvez communiquer avec Mr. Olivier Valentin (olivier.valentin@etsmtl.ca). Pour réserver un espace de publicité dans un de ces numéros spéciaux, veuillez communiquer avec Bernard Feder (advertisement@caa-aca.ca).

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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

ICSV26 to be held in Montreal, July 2019

The 26th International Congress on Sound and Vibration (ICSV26) will held in Montreal, Canada, from 07 - 11 July 2019 at Hotel Bonaventure.

The local organizing committee and scientific committees are currently being formed. Please contact us if you are interested to be part of the adventure! :-) - - You can also check out our website at www.icsv26.org - - Jeremie Voix (conference-chair@icsv26.org) - Franck Sgard (technical-chair@icsv26.org) -

October 12th 2017

AWC18 in Victoria, BC

The 176th Meeting of the Acoustical Society of America (ASA) will be held jointly with the Acoustics Week in Canada 2018 of the Canadian Acoustical Association (CAA) in Victoria, BC, Canada, on 5-9 November 2018.

The conference will be organized by the Acoustical Society of America using their guidelines and procedures, while the Canadian Acoustical Association will organize some special sessions and handle its regular business and core activities, such as standards committee, student awards, etc. - - For more information, visit http://acousticalsociety.org/meetings - - To contact Dr. Roberto Racca, AWC2018 conference coordinator, please send an email to: conference@caa-aca.ca

November 24th 2017

ASA Fall 2018 meeting/2018 Acoustics Week in Canada

The joint 176th Meeting of the Acoustical Society of America (ASA) and 2018 Acoustics Week Canada of the Canadian Acoustical Association (CAA) will be held Monday through Friday, 5–9 November 2018 at the Victoria Conference Centre, Victoria, British Columbia, Canada. The headquarters hotel is the Fairmont Empress. - -

Blocks of rooms have been reserved at the Fairmont Empress and the Victoria Marriott Inner Harbour Hotel at discounted rates. Please refer to the Victoria meeting website for the call for papers and instructions for submitting abstracts and making hotel reservations. - The deadline for receipt of abstracts is Tuesday, 29 May 2018. - - The call for papers describes the special sessions that will be organized by the ASA Technical and Administrative Committees and the CAA and other events such as a hot topics session, short course, and tutorial lecture. Social events will include the Social Hours, Women in Acoustics Luncheon, the Society Luncheon and Lecture, and the Jam session. - - Special events include the Undergraduate Research Exposition and Early-Career Speed-Networking. - - Student events, organized by the Student Council, will include a Student Orientation, a Meet and Greet, and the Student Reception. See the Student Information section of the meeting announcement for details including information about the Students Meet Members for Lunch program. - - Funding opportunities are offered including Student Transportation Subsidies, Dependent Care Subsidies, Young Investigator Travel Grants, Early Career Travel Subsidies, and Best Paper Awards for Students and Early Career Acousticians. - - Accompanying persons are welcome at the meeting and a program of activities will be organized. - - We hope that you will consider presenting a paper or attending the meeting to participate in the exchange of ideas and the latest research developments in acoustics and to meet with your colleagues in the ASA. -

April 19th 2018

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

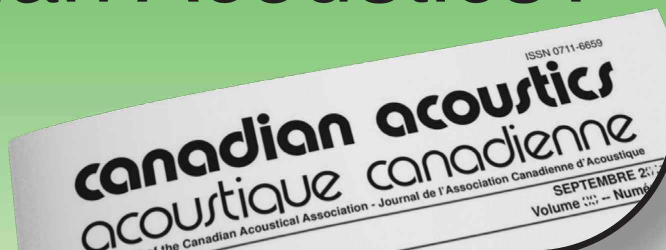
AWC18 à Victoria, B.-C.

La 176e rencontre de l'Acoustical Society of America (ASA) se tiendra conjointement avec la Semaine canadienne d'acoustique 2018 de l'Association canadienne d'acoustique à Victoria, C.-B., du 5 au 9 novembre 2018.

La conférence sera organisée par l'Acoustical Society of America selon leurs méthodes et procédures, tandis que l'Association canadienne d'acoustique y organisera des sessions spéciales et tiendra ses rencontres régulières ainsi que ses activités propres, telles la rencontre des comités de normalisation, le programme de prix pour les étudiants, etc. - - Pour plus d'information, visiter http://acousticalsociety.org/meetings - - Pour contacter Dr Roberto Racca, le coordinateur du congrès AWC2018, merci d'écrire un courriel à : conference@caa-aca.ca

November 24th 2017

Why publish in Canadian Acoustics?



Because, it is...

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

Pourquoi publier dans Acoustique canadienne ?

Parce que, c'est...

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

Application for Membership

CAA membership is open to all individuals who have an interest in acoustics. Annual dues total $110.00 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 $110.00 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.

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Formulaire d'adhésion

L'adhésion à l'ACA est ouverte à tous ceux qui s'intéressent à l'acoustique. La cotisation annuelle est de 105.00$ pour les membres individuels, et de 50.00$ pour les étudiants. Tous les membres reçoivent ***l'Acoustique Canadienne***, la revue de l'association.

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