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Evaluation of the Sound Insulation of Liquid-Filled Panels

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To my family, who have always encouraged me to do my best

To Joana

"…the manager of a large wind tunnel once told one of the

authors that in the evening he liked to hear, from the back porch of his home, the steady hum of his machine 2 km away, for to him the hum meant money. However, to his neighbours it meant only annoyance and he eventually had to do without his evening pleasure."

*Bies, D., Hansen, H. "*Engineering Noise Control – Theory and Practice*"*

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ABSTRACT

A liquid is generally thought of as a better sound transmission medium than air, since the sound wave propagation velocity is normally three or four times higher, as is the case with water, for example, where $c_L \approx 1480$ m/s compared to air, with $c_L \approx 343$ m/s at 20ºC. This holds true when sound waves travel within the same medium, since the bonds between molecules of the same material facilitate the propagation of energy within itself. However, if the sound source is in air and a liquid is used as a sound barrier, according to Snell's law and Barron [2], there is a significant reduction in sound transmission when there is a change in medium and the receiving medium has larger specific acoustic impedance than the first. This is due to the interface phenomenon, where there are no bonds in a molecular level between the two materials, and therefore a significant transmission loss is felt at the interface between the media. This is the starting point for the evaluation of sound insulation of liquid-filled panels, in order to understand the potential of these for noise control in buildings or other applications.

Two liquids have been tested (water and vegetable oil) using the Acustilab, a small reverberation chamber with a removable concrete lid, used for study purposes in the Laboratory of Acoustics of the Faculty of Engineering, University of Porto, Portugal. A modified concrete lid with an opening was designed and built so that a test sample – horizontal panel made of a wooden frame, a thin LDPE sheet and steel grill – could be fitted into the lid and different liquids tested. A sound source was placed inside the Acustilab with the lid. The sound pressure level difference between the inside and outside of the chamber was measured with both liquids and different layer thicknesses (20, 40 and 80 mm). The liquids have similar density and sound propagation velocity and therefore gave almost identical results, which have practically no correlation with existing sound insulation prediction models. The obtained R_W values are 34 dB, 43 dB and 48 dB for 20 mm, 40 mm and 80 mm of water layer thickness respectively, and for vegetable oil R_W values are 33 dB, 42 dB and 49 dB for the 20 mm, 40 mm and 80 mm thick layer respectively. A test method and a test sample design for liquids are both proposed, although the limitations of the equipment used reduced the frequency spectrum that produces reliable results to 315-3150 Hz.

A prediction formula for the sound insulation index (R_W) of liquids is proposed, based on Snell's law and a correction for liquid layer thickness. Further testing is required to evaluate the performance of liquids in lower frequencies, with a larger test sample and better laboratory facilities. Possible applications for liquids in construction are suggested.

KEYWORDS: liquids, sound insulation, acoustics, noise control, water

RESUMO

Um líquido é geralmente considerado como um melhor meio para a propagação de som relativamente ao ar, visto que a velocidade de propagação das ondas sonoras de num líquido será três a quatro vezes superior ao ar, como acontece com a água, por exemplo, em que $c_L \approx 1480$ m/s relativamente ao ar, $c_L \approx 341$ m/s a 20 ºC. Isto é válido quando a fonte sonora mantém-se no mesmo meio, visto que as ligações inter-moleculares são tais que facilitam a propagação da energia sonora dentro deste meio. No entanto, se a fonte sonora estiver no ar e o líquido for utilizado como uma barreira acústica, de acordo com a lei de Snell e com Barron [2], há uma redução significativa de transmissão das ondas sonoras quando há uma mudança no meio de propagação e o meio receptor tem uma impedância específica bastante maior do que o meio emissor. Esta ocorrência deve-se ao fenómeno de interface entre materiais diferentes, onde não existem ligações moleculares entre os materiais e por isso verifica-se nessa interface uma grande perda de transmissão de ondas sonoras. Este é o ponto de partida para a avaliação experimental do comportamento de lâminas de líquidos em isolamento sonoro, de forma a ser possível compreender o seu potencial no controlo de ruído em edifícios ou em outras aplicações.

Foram realizados ensaios com dois líquidos (água e óleo vegetal) utilizando o Acustilab, uma pequena câmara de ensaio com tampa amovível em betão, utilizada para efeitos de estudo no Laboratório de Acústica da Faculdade de Engenharia da Universidade do Porto. Foi projectada e construída uma tampa com abertura própria para suportar uma amostra de ensaio – uma moldura em madeira, uma folha fina de plástico LDPE e uma estrutura metálica – que possibilitasse a realização de ensaios em diferentes líquidos. Foi colocada uma fonte sonora dentro do Acustilab e a tampa colocada por cima, foi medida a diferença de pressão sonora entre o interior e exterior da câmara de ensaio induzida por líquidos diferentes a espessuras diferentes (20, 40 e 80 mm). Os dois líquidos testados têm densidade e velocidade de propagação de ondas sonoras equivalentes e por isso os resultados foram quase idênticos, não tendo, no entanto, qualquer correlação com modelos de previsão de isolamento sonoro existentes. Os valores de R_W obtidos para a água a espessuras de 20 mm, 40 mm e 80 mm são, respectivamente, 34 dB, 43 dB e 48 dB, enquanto que para o óleo vegetal nas mesmas espessuras são 33 dB, 42 dB e 49 dB. É proposto um método de ensaio e uma solução construtiva para a amostra de ensaio para uma futura validação experimental de líquidos. No entanto as limitações impostas pelo equipamento e instalações reduziram o espectro de frequências onde há resultados fiáveis para 315-3150 Hz, em bandas de terço de oitava.

Também é proposto um modelo de previsão do índice de redução sonora R_W com base na lei de Snell e uma correcção para a espessura da lâmina de líquido. Será necessário proceder a mais ensaios de forma a validar o comportamento de líquidos a baixas frequências, recorrendo a uma amostra de ensaio de maiores dimensões e instalações laboratoriais mais adequadas. São também sugeridas algumas possíveis aplicações de líquidos no âmbito da construção.

PALAVRAS-CHAVE: líquidos, redução sonora, acústica, controlo de ruído, água

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1
Introduction

1.1 FIRST THOUGHTS

The advances in construction technology and the development of new, more efficient, materials have created a demand for new solutions that give more flexibility to architects and project designers. These aspects, allied to the growing demand for better acoustic quality and performance of buildings promote experiments on new materials and a look into non-traditional ways of using 'traditional' materials.

This work is the written part of a Master Course in Civil Engineering (Mestrado Integrado em Engenharia Civil) at the Faculty of Civil Engineering, University of Porto, according to the Bologna Declaration of 1999. Its scope is Noise Control, a part of the Building and Architectural Acoustics sector, a branch of Acoustics and ultimately, Physics, as described in Fig. 1.1.

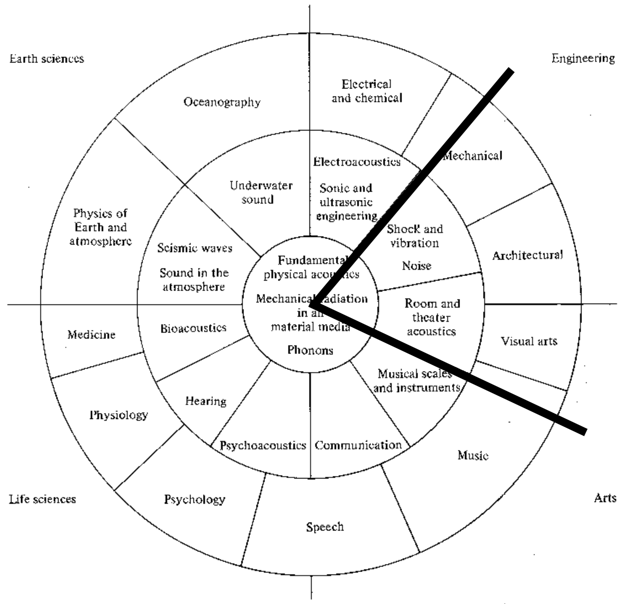

Fig. 1.1: Circular chart illustrating the scope and ramifications of Acoustics (adapted from R.B. Lindsay, J. Acoust. Soc. Am., 36:2242 (1964))

1.2 AIM/PRINCIPLE

A known architect designed an open space in which various different areas were to be simultaneously used as conference rooms, lecture halls and meeting spaces. These different areas needed to be reasonably isolated from their surroundings – from both traffic noise and noise produced from the other spaces – so that people could hear themselves inside the spaces and have some degree of privacy. At the same time this means of sound insulation would have to be lightweight and easily removable, to add flexibility to the open space. This put a problem to the acoustic consultant: in one hand the partitions needed to be lightweight and portable, and at the same time have adequate sound insulation properties.

This real and concrete problem made the author think about the need for a new material which could work in such a way that it could be both thin and have adequate sound insulation. It would, in turn, conflict with the idea that a partition with good sound insulation needs to be both thick and heavy.

A possible solution for this problem thought by the author would be to use water as a sound barrier (with an adequate container).This was based on the fact that when one dives into a pool, he or she doesn't hear sound emitted outside the water, due to the air/water interface. A thin layer of water would probably still produce that effect, since the air/water interface would be present in both sides of the partition – sound would have to pass from air into water, and then back into air again.

The main aim of this project is to evaluate the transmission loss or sound insulation index R_W of a liquid-filled panel in order to understand the usefulness of different liquids as sound insulators in various possible applications. The two liquids tested are water and a vegetable oil.

The main challenge of this project is the test sample design and construction, in which a few issues must be considered:

- The sample has to hold the fluid during laboratory testing and for a reasonable amount of time, taking into account hydrostatic pressure exerted by the liquid on the walls of its container;
- It should create a thin layer of water/liquid in which both surfaces are reasonably plane and homogeneous;
- The test sample or panel that contains the liquid must be made of materials that have very high sound transmission loss around the liquid and very low transmission loss between the liquid and the source/receiving rooms, so that meaningful results can be obtained.

1.3 STRUCTURE OF THIS WORK

This work begins with an introduction that gives some insight into its scope in Chapter 2, explaining the need for noise control and a short history of the development of acoustics as a science. Chapter 3 gives basic theory of acoustics, namely sound insulation and transmission, all of which are related to noise control and building acoustics. The relevant European Standards and Portuguese legislation are also referred to in that Chapter. Chapter 4 describes the test method and prediction model calculations, going into detail about the test sample and its design. The results are then presented on Chapter 5, where test results and calculations are compared, and the work is concluded with Chapter 6, where not only a formula for the prediction of the sound insulation index is described, as future projects and possible applications for liquids in construction and other uses are listed.

Annex A includes diagrams of the prediction models used, and detailed drawings of the Acustilab modified lid. A table of symbols is also included.

2
Acoustics – Historical, Social and Economic Context

2.1 INTRODUCTION

In order to properly discuss any problem it must be put into context. The following chapter attempts to do just that, giving an overview of the importance of acoustics and noise control in buildings, a brief history of acoustics and its evolution into a full-fledged science, and some insight on the current developments in Acoustics.

2.2 SOCIAL AND ECONOMIC CONTEXT (THE IMPORTANCE OF NOISE CONTROL)

According to Bies and Hansen [1], the recognition of noise as a source of annoyance began in antiquity, but there is a growing relationship in more recent times between noise and money, in a sense that noise may be a desired end or an inconsequential by-product for one group, i.e. disco club or factory, respectively, or the annoyance of another group (i.e. disgruntled neighbours who can't sleep!). According to the authors, each group can have what it wants only to the extent that control is possible. In the 21st Century, most of the world's population lives in an urban setting, and so is exposed to noise created from artificial sources on a daily basis. This affects quality of life in a quite significant way, which means noise levels are too high or too continuous, causing discomfort or in more extreme cases severe health problems.

Through the internet and globalization comes about the information age, which has helped to create a greater awareness of the world as a whole, and in turn has helped the world's citizens, especially in developed countries, to form the idea of 'quality of life'. This translates into a greater demand for quality of life, mainly in urban settings where it can be reduced due to various forms of pollution, namely sound pollution.

In most cities noise pollution due to road traffic and industry has become commonplace. This aspect, combined with the reduction of construction quality due to economic reasons and/or new, untested or badly executed construction techniques has resulted in the public putting pressure on legislators to act upon sound pollution, therefore limiting exposure to high noise levels that can disturb quality of life.

The chief idea behind this work is to help understand if liquids are efficient enough to be introduced in various different noise control applications.

2.3 ACOUSTICS, A BRIEF HISTORY IN TIME

According to Britannica [3] and Rayleigh [4], the basic theory of acoustics, i.e. the origin, propagation and reception of sound was produced by the ancient Greeks, namely Pythagoras (6th century BC), and Aristotle (4th century BC), the first having recorded experiments on vibrating strings, and the latter having correctly assumed that sound waves are transmitted by the air through some undefined motion, although having mistakenly assumed that higher frequency sounds travelled faster than lower frequencies.

The Roman architectural engineer Vitruvius (1st century BC) published "*Ten Books on Architecture*", which refers to the mechanism for transmission of sound, and provided a substantial advancement in the acoustic design of theatres. The Roman philosopher Boethius suggests a link between the human perception of pitch and the frequency of sound waves, in the 6th century AD.

A few centuries later Galileo Galilei (1564–1642) studied vibrations and "…discovered the general principles of sympathetic vibrations, or resonance, and the correspondence between the frequency of vibrations and the length of a pendulum."[5]

Various scientists quickly followed on Galileo's groundwork, such as the French mathematician Marin Mersenne, who's "…*description in his* Harmonic universelle *(1636) of the first absolute determination of the frequency of an audible tone (at 84 Hz) implies that he already demonstrated that the absolute-frequency ratio of two vibrating strings, radiating a musical tone and its octave, is as 1:2. The perceived harmony (consonance) of two such notes would be explained if the ratio of the air oscillation frequencies is also 1:2, which in turn is consistent with the source-air-motion-frequency-equivalence hypothesis*"[6]. Some authors attribute the first measurement of the speed of sound to Marin Mersenne, although further research reveals that in fact Pierre Gassendi, a contemporary of Mersenne, actually was the first to record the speed of sound: "*As early as 1635, Pierre Gassendi, while in Paris made measurements of the velocity of sound in air. His value was 1473 Paris feet per second. (The Paris foot is approximately equivalent to 324.8 mm). Later Marin Mersenne* [3]*, often referred to as the "father of acoustics," corrected this to 1380 Paris feet per second, or about 450 m/s. Gassendi also demonstrated conclusively that velocity is independent of frequency*"[7].

When Francis Bacon (1561–1626) first mentioned the "Acoustique Art" in his *Advancement of Learning* (1605), the word 'acoustics' was used for the first time to describe the study of sound, which is derived from the ancient Greek word ακουστός, (akoustos) meaning able to be heard.

G.L. Bianconi demonstrated in 1740 that the speed of sound in air increases with temperature, and in the Academy of Sciences in Paris was conducted the earliest precise experimental value for the speed of sound, in 1738 – 332 metres per second. The now accepted value of the speed of sound at 0º C is 331.29 metres per second, amended in 1986.

"*The speed of sound in water was first measured by Daniel Colladon, a Swiss physicist, in 1826. Strangely enough, his primary interest was not in measuring the speed of sound in water but in calculating water's compressibility—a theoretical relationship between the speed of sound in a material and the material's compressibility having been established previously. Colladon came up with a speed of 1,435 metres per second at 8° C; the presently accepted value interpolated at that temperature is about 1,439 metres per second* [3]."

Parallel to these early experiments conducted in the 18th century, a more theoretical approach was used in order to develop the mathematical theory of waves, in acoustics and other branches of physics: "*The mathematical theory of sound propagation began with Isaac Newton (1642-1727), whose Principia (1686) included a mechanical interpretation of sound as being "pressure" pulses transmitted through neighbouring fluid particles*. […] *Substantial progress towards the development of a viable theory of sound propagation resting on firmer mathematical and physical concepts was made during the 18th century by Euler (1707-1783), Lagrange (1736-1813), and d'Alembert (1717-1783). During this era, continuum physics, or field theory, began to receive a definite mathematical structure. The wave equation emerged in a number of contexts, including the propagation of sound in air. The theory ultimately proposed for sound in the eighteenth century was incomplete from many standpoints, but modern theories of today can be regarded for the most part as refinements of that developed by Euler and his contemporaries.*" [6]

In the science of optics, closely related to acoustics, there seemed to be a conflict between ray and wave theories, but the theory of sound developed as a wave theory since its beginning. In this sense, ray-tracing concepts used by various scientists in the 19th century, such as John Scott Russell in the *Treatise of Sightlines* (1836), or Rayleigh (see below) to explain certain phenomena were viewed as mathematical approximations to the wave theory. These are still incorporated into a more comprehensive wave theory and used as approximate models that simplify complicated wave phenomena in applications such as architectural acoustics, underwater acoustics and noise control [6].

In the 19th century, one of the most significant developments was the theory of vibrating plates, developed by German physicist Ernst E. F. Chladni. He developed a technique in 1787 to show the various modes of vibration (standing-wave patterns) in a mechanical surface by vibrating a metal plate and sprinkling sand over it. In 1816 French mathematician Sophie Germain provided a sophisticated theoretical explanation of this phenomenon, whose errors in her approach were only identified by German physicist Gustav Robert Kirchhoff about 35 years later.

Jean Baptiste Joseph Fourier, a French mathematician and physicist established the theoretical analysis of a complex periodic wave into its spectral components in 1822, which is now known as the Fourier theorem. The German physicist Georg Simon Ohm proposed in 1843 that a musical sound is perceived by the ear as the sum of a number of pure harmonic tones, and it is sensitive only to the amplitudes but not the phases of the harmonics of a complex tone - this is known as Ohm's acoustic law.

Hermann von Helmholtz (1821-1894), wrote *On the Sensations of Tone as a Physiological Basis for the Theory of Music* (1863), in which significant advancements in the understanding of the mechanisms of hearing and in psychophysics of sound and music. He also constructed a set of resonators that were used in the spectral analysis of musical tones, known as Helmholtz resonators, which are have been incorporated in architectural acoustics for sound absorption.

One of the most important works that marked the beginning of modern acoustics was John William Strutt, 3rd Baron Rayleigh's two-volume treatise, *The Theory of Sound* (1877-8), which included a large variety of his research on acoustics. It was the first to examine questions of vibrations, the resonance of elastic solids and gases and acoustical propagation in material media.

The 20th century boasts innovators such as American physicist Wallace Sabine (1868-1919), considered to be the originator of modern architectural acoustics, who searched for ways to correct the acoustics of noisy rooms, and was the acoustical architect Boston's Symphony Hall, widely considered one of the two or three best concert halls in the world for its acoustics. He also developed the popular Sabine's formula that relates the sound decay time (reverberation time) of a room with its volume and sound absorption of its surfaces. Hungarian-born American physicist Georg von Békésy validated Helmholtz's theory of hearing with his *Experiments in Hearing* (1960), the classic of the modern theory of the ear [3] and [5].

2.4 The Principles of Noise Control

In the spirit of Chapter 2.2, in this search for better quality of life, the term 'noise control' starts to make sense, as the acoustic consultant tries to reduce sound exposure according to existing legislation. Different criteria allow him to act at the *noise source*, at the *receiver* (see Fig. 2.1), in the path between them, be it direct or indirect (sound barriers), or in a combination of these, keeping costs in mind, which in turn shall be materialized into a sustainable and cost efficient solution.

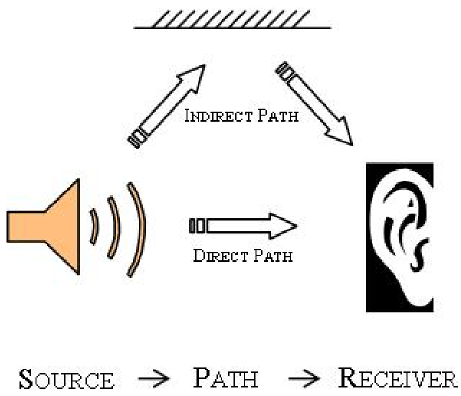

Fig. 2.1: Three components of a general noise system: source of noise, path of the noise, and the receiver.

These are the three main principles of noise control, and as M. Barron says: *"…in many situations, of course, there are several sources of sound, various paths for the sound, and more than one receiver, but the basic principles of noise control would be the same as for the more simple case. The objective of most noise control programs is to reduce the noise at the receiver. This may be accomplished by* *making modifications to the source, the path, or the receiver, or to any combination of these elements*.”[2]

Sound may be transmitted by various different paths, either directly, such as the air between the source and receiver (in an open-air rock concert, for example, where music is transmitted from the stage to the listener’s ears), or it may be transmitted indirectly, such as the sound being reflected by a wall to a person in the room. “*Solid surfaces, such as piping between a vibrating pump and another machine element, may also serve as the path for the noise propagation. It is important that the acoustic engineer is able to identify all possible acoustic paths when considering a solution for a noise problem.*”[2].

Although in most cases the receiver is usually the human ear, some types of sensitive equipment are affected when exposed to excessively intense noise. It is therefore important for the acoustician or acoustic engineer to define the “failure mode” for the receiver in any noise control project. Examples of a noise control project are: prevention hearing loss for personnel, to allow effective simultaneous translation in an international conference, to allow effective face-to-face communication or telephone conversation, or to reduce discomfort in dwellings caused by traffic noise, nearby industrial plants or commercial spaces, or even neighbours who appreciate loud music at night. The engineering approach is often different in each of these cases.

2.4.1 Noise Control in the Transmission Path

Changing the noise transmission path is usually used when it is either impossible or impractical to modify the noise source (or sources). One simple way of resolving a noise problem when the source is outdoors is to move it farther away from the receiver, in order to make the path longer, so that the sound energy that reaches the receiver is a lot weaker than when it was closer.

The transmission path may also be changed by the placing of a sound barrier between the source and the receiver. One solution that is used quite frequently to reduce traffic noise is to place acoustic barriers along the sides of roads and highways passing near residential zones, hospitals or other sensitive areas. This type of solution may not be effective when both source and receiver are in the same room, where there are various different paths besides the direct path, such as reflections of sound waves off the surrounding walls, and therefore the direct path may lose its significance. In this case the reflecting walls may be mounted with sound absorbing material or additional sound absorbing surfaces may be placed in the room (i.e. acoustic absorption panels suspended on ceilings of large spaces, such as public pools or large exhibition halls).

When source and receiver are indoors and between two contiguous rooms, another approach is needed, since sound is transmitted not only through the wall or partition directly between the two rooms, but also through the structure surrounding them, as shall be explained later on in this work. This means that the sound barrier in itself loses significance, and therefore additional measures must be taken, such as the construction of floating floors (the floor is mounted on a resilient layer that absorbs structural vibrations) to prevent the transmission through the underlying structure. Liquids could have a very important role in these situations, since integrated solutions could be designed by taking in account the various sound transmission paths.

According to Barron [2], “*A very effective, although sometimes expensive, noise control procedure is to enclose the sound source in an acoustic enclosure or enclose the receiver in a personnel booth.*”, which is a very interesting solution for musicians who need to practice at home and cannot annoy the neighbours, or during a conference where simultaneous translation is needed in different languages, or even for factory workers who need control booths to safely and comfortably operate noisy, large and potentially dangerous machines: *“…An air-conditioned control booth is also more comfortable for the operator of a paper machine than working in the hot, humid area surrounding the wet end of the paper machine, for example.*”[2].

2.5 CURRENT DEVELOPMENTS IN ACOUSTICS

2.5.1 FOREWORD

Building and architectural acousticians are mainly concerned with two aspects:

- *Acoustical correction*, which is simply the *improvement of sound transmission* between source and receiver, and normally involves the correction of the acoustics of a certain room for sounds produced inside that same room.
- *Sound insulation* or *noise control*, which is the *degradation of sound transmission* from source and receiver, and may involve the correction/treatment of a room according to sound produced from another room/volume connected to the first.

These two aspects, which apparently oppose each other, form the basis of the acoustic engineer's work. He (or she) must always know which one to apply and where.

The following section is an attempt to give an idea of the state of acoustics around the world and in Portugal, in order to further define the scope of this work.

2.5.2 ACOUSTICS DEVELOPMENTS AROUND THE WORLD

2.5.2.1 THE BUILDING AND ARCHITECTURAL ACOUSTICS SECTOR

Both the potential escalation of building costs due to high acoustic material prices, and the increasing need to be more cost-efficient put a lot of pressure on architectural acousticians around the world in terms of developing a design that can meet the acoustic and budget requirements. Therefore, there is high demand for increasingly accurate and elaborate prediction methods so that there are as few 'surprises' as possible. The industry is now directed mainly towards the production and testing of such methods through information exchange and development of new approaches to parameter calculations using numerical methods such as the Finite Element Method (FEM) or the Boundary Element Method (BEM) which use all the processing power of today's computers, which was previously unavailable or inadequate.

Other approaches include trying to correlate physical (measurable) parameters with psychological sensations (psychoacoustics, through inquiries), in order to understand what the actual 'public' feels when inside a room with specific acoustic characteristics [8], [9]; or creating error reduction methods [10], so that tests may give more accurate results.

2.5.2.2 PRINCIPAL ORGANIZATIONS, CONGRESSES AND WEBSITES

There are various different international organizations related to Acoustics in its various fields, which have the following common goals:

- Publication of scientific documents related to Acoustics;
- Support for the study and implementation of new and improved acoustics technology, the main aim being the continuous advancement of the field;
- Organisation of meetings, symposiums and/or congresses in order to promote information exchange between scientists and engineers that work in acoustics;
- Promotion of acoustics as a scientific field of study and creating an interest amongst students who are potential workers in the sector.

The main international organisations are:

European Acoustics Association (EAA) - Organised in the form of a non-profit making enterprise under the ownership of its member societies, it provides a firm base for many activities in acoustics which earlier European groupings could not achieve. The European Acoustics Association EAA is a service provider to currently 29 Acoustical Societies throughout Europe, with more than 8500 individual members – www.eaa-fenestra.org.
Acoustical Society of America (ASA) - The Society is primarily a voluntary organization and attracts the interest, commitment, and service of a large number of professionals. Their contributions in the formation, guidance, administration, and development of the ASA are largely responsible for its

world-wide pre-eminence in the field of acoustics. It was founded in 1929 and has around 7000 members who work in acoustics throughout the U.S. and abroad - http://asa.aip.org/.

Other organisations include:

Australian Acoustical Society (AAS) - www.acoustics.asn.au/index.php
Acoustical Society of Japan (ASJ) - www.asj.gr.jp/eng
NAFEMS, UK - www.nafems.org
New Zealand Acoustical Society (NZAS) - www.acoustics.ac.nz
Vibration Institute, USA - www.vibinst.org

Meetings include:

Forum Acusticum, International Congress on Acoustics (ICA), International Conference on Noise and Vibration Engineering, International Congress on Sound and Vibration (ICSV), Acústica (Portugal).

International publications:

Acta Acustica united with Acustica (EAA), Applied Acoustics (Elsevier), Journal of the Acoustical Society of America (ASA), Acoustics Today (ASA) and Building Acoustics (Multi-Science Publishing).

2.5.3 Acoustics in Portugal

2.5.3.1 The Building and Architectural Acoustics Sector

With constructions of considerable scale such as the Casa da Música in Porto, the Holy Trinity Church in Fátima (Igreja da Santíssima Trindade), and others that have specific acoustic requirements, one can say that there are engineers in Portugal who's expertise and know-how are more than adequate to find effective and cost efficient solutions to meet those requirements. In fact, from 1995 to 2005 about 30 new halls of various types and with different acoustic requirements have been built throughout the country. There is also a growing demand for cost efficient solutions to be implemented in dwellings, office spaces and commercial areas due to recent Portuguese legislation that imposes demanding high sound insulation index between spaces of different uses in order to guarantee acoustic comfort for the general public.

2.5.3.2 Principal Organizations, Congresses and websites

The main acoustics organisation in Portugal is the *Portuguese Acoustical Society* (*Sociedade Portuguesa de Acústica*), a member society of the European Acoustics Association, presided by Jorge Viçoso Patrício of the National Laboratory of Civil Engineering (LNEC). It is an independent, non-profit and non-governmental organisation and its main purpose is to "…congregate the Portuguese acousticians and to promote and develop acoustics within the country, being simultaneously a forum to exchange information amongst its members. Similarly, another purpose of the SPA is to serve as link between several areas of different scientific fields that have acoustics as a common subject. It also aims at promoting technical and scientific advances in acoustics throughout the country and to facilitate the recognition – both in Portugal and abroad – of activities developed by those basically working in acoustics and by those who are simply interested in it.

The SPA is currently affiliated to EAA, ICA, IIAV and I-INCE. It is also a member of FIA (Ibero-American Federation of Acoustics). SPA has presently a representative number of effective members, including individuals and associates. There are also other individuals who have strong liaisons with SPA without being real members."[11]

The "Ordem dos Engenheiros" (Engineers Association), is the Portuguese public association that represents all of those who have a valid degree in engineering and regulates the engineering practice. It has around 31.000 members and has delegations all over the Portuguese territory. The association organizes congresses, and short courses for engineers in the various fields available.

It has established a Specialization in Acoustic Engineering in the year 2000, composed of around 20 Specialists. This Specialization aims to improve the overall quality of acoustic engineering practised by the association's members and also to create awareness of the problems faced by engineers who work in this area of expertise.

Portuguese universities in general also include at least one course on Acoustics and/or Building Acoustics in the Civil Engineering courses available.

3
Theoretical Background

3.1 Introduction

"Acoustics *is the science of sound, including its production, transmission, and effects. In present usage, the term sound implies not only phenomena in air responsible for the sensation of hearing but also whatever else is governed by analogous physical principles. Thus, disturbances with frequencies too low (infrasound) or too high (ultrasound) to be heard by a normal person are also regarded as sound. One may speak of underwater sound, sound in solids, or structure-borne sound. Acoustics is distinguished from optics in that sound is a mechanical, rather than an electromagnetic, wave motion.*"[6]

The following chapter describes the theory related to the scope of this work, giving general definitions of acoustics and sound waves, and then going into more detail in the aspects related to sound insulation of both solids and fluids. Existing theory on the sound insulation of solid (better yet, rigid) panels is included in order to provide a knowledge base and reference values that can then be compared to results of liquid-filled (or non-rigid) panels. This was deemed necessary due to the lack of information available on the sound insulation of liquids, which further justifies the driving force of this work (besides scientific curiosity).

3.2 General Definitions

3.2.1 Sound and Noise

Sound may be defined by an alteration in pressure or a wave motion caused by a disturbance propagated in an elastic medium (See Fig. 3.1) and is perceived by some kind of hearing mechanism, i.e. the human ear or a microphone. Concerning human beings, one could say that the perception of that sound will cause a *sensation*. If this sensation is pleasant or has meaning to the receiver it can be called s*ound*, whereas if it is unpleasant or devoid of meaning it is referred to as *noise*.

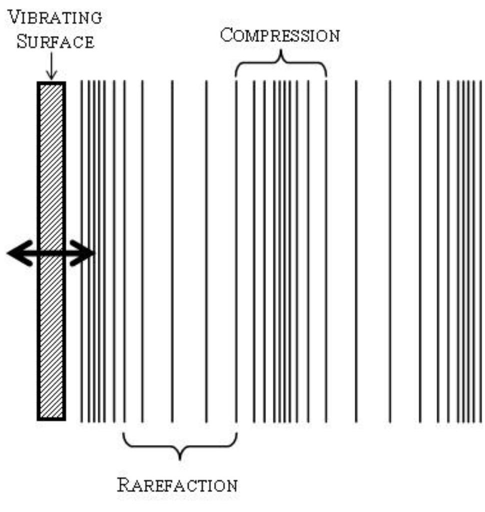

Fig. 3.1: Sound propagation in air caused by a vibrating surface.

3.2.2 FREQUENCY, WAVELENGTH AND WAVE NUMBER

A single frequency f is associated with a simple harmonic wave or sinusoidal wave. This frequency depends on the frequency of vibration of the source of sound and is independent of the material through which the sound is transmitted for non-dissipative sound transmission. According to Barron [2], "…*the unit for the frequency is hertz (Hz), named in honour of the German physicist Heinrich Rudolph Hertz, who conducted pioneering studies in electromagnetism and in elasticity* […].

The unit hertz is the same as the unit cycle/sec. To get a physical understanding of the magnitude of the frequency of sound waves usually considered in noise control, we may note that the range of audibility for the undamaged human ear is from about 16 Hz to about 16 kHz. Frequencies below about 16 Hz are considered infrasound, and frequencies above 16 kHz are ultrasound.[…] the female voice has a frequency on the order of 500 Hz. The baritone voice usually ranges from about 90 Hz to 370 Hz, so the male voice has a frequency on the order of 200 Hz."

The period T for a wave is defined as the time elapsed during one complete cycle for the wave, or the time elapsed between the passage of the successive peaks for a simple harmonic wave, as shown in Fig. 3.2(A). The frequency is the reciprocal of the period, $f = 1/T$. The wavelength of the sound wave is an important parameter in determining the behaviour of sound waves. If we take a ''picture'' of the wave at a particular instant in time, as shown in Fig. 3.2(B), the wavelength is the distance between successive peaks of the wave. The wavelength and speed of sound for a simple harmonic wave are related through:

$$\lambda = c/f \tag{3.2.1}$$

Where:

λ – wavelength of a sound wave, in m;
c - sound propagation speed in a medium, in m/s;
f - frequency of a sound wave, in Hz.

Another parameter that is encountered in analysis of sound waves is the wave number k, an associated quantity of a wave that represents phase change per unit distance, and also represents the magnitude of a vector quantity that indicates the direction of propagation as well as the spatial variation:

$$k = \frac{2\pi}{\lambda} = \frac{2\pi f}{c} \tag{3.2.2}$$

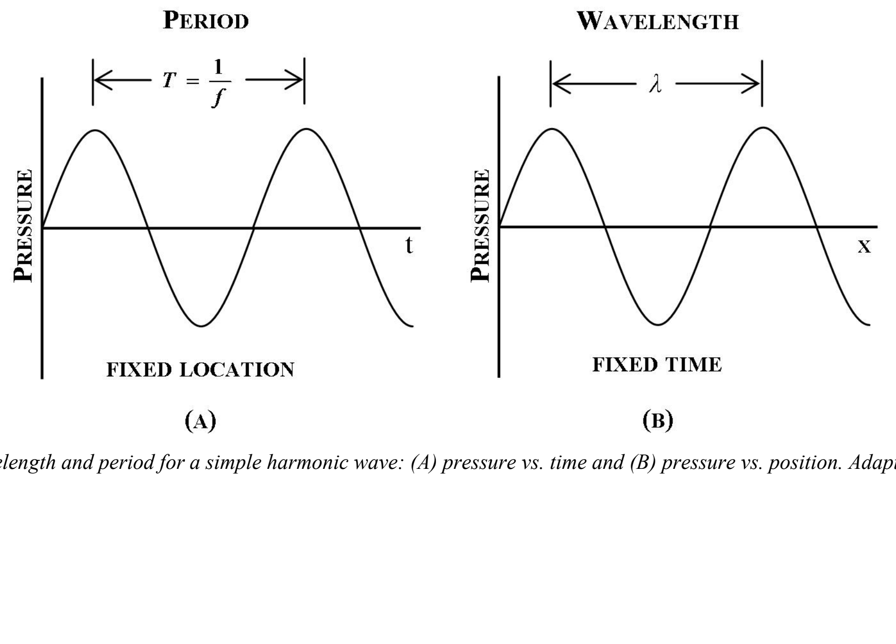

Fig. 3.2: Wavelength and period for a simple harmonic wave: (A) pressure vs. time and (B) pressure vs. position. Adapted from: [2]

3.2.3 FREQUENCY ANALYSIS

The human ear is sensitive to sounds in the frequency range from about 16 Hz to 16 kHz. Single frequency sound is useful for analysing acoustic phenomena, but due to the fact that is not practical to measure the sound level at each of the 15,984 frequencies in this range, acoustic measuring instruments generally measure the acoustic energy included in a range of frequencies, or frequency bands. As said by Barron [2], "*The human ear also responds more to frequency ratios than to frequency differences, so the frequency ranges generally have terminal frequencies (upper and lower frequencies of the range) that are related by the same ratio.*"

The interval between terminal frequencies, over which measurements are made, is called bandwidth. The bandwidth is described by the lower frequency of the interval and the upper frequency of the interval, f_1 and f_2 respectively. In acoustic measurements bandwidths are often specified in terms of octaves and third-octaves. An octave is a frequency interval in which the upper frequency is double the value of the lower frequency:

$$f_2 = 2f_1 \qquad \text{or} \qquad \frac{f_2}{f_1} = 2 \tag{3.2.3}$$

Where:

f_1 – lower frequency of a frequency band interval, in Hz;
f_2 – lower frequency of a frequency band interval, in Hz.

In this work, a more refined division of bandwidths is needed, and so third-octave bands are used, where the intervals between frequencies are determined by:

$$\frac{f_2}{f_1} = 2^{1/3} \tag{3.2.4}$$

The frequency bands in general are defined by the geometric mean of the respective terminal frequencies (upper and lower), known as the centre frequency f_0:

$$f_0 = (f_1 f_2)^{1/2} \tag{3.2.5}$$

Where:

f_0 – centre frequency of a frequency band, in Hz;
f_1 – lower frequency of a frequency band interval, in Hz;
f_2 – lower frequency of a frequency band interval, in Hz.

Which means that for octave bands:

$$f_1 = \frac{f_0}{2^{1/2}} \qquad \text{and} \qquad f_2 = 2^{1/2} f_0 \tag{3.2.6a,b}$$

For 1/3-octave bands:

$$f_1 = \frac{f_0}{2^{1/6}} \qquad \text{and} \qquad f_2 = 2^{1/6} f_0 \tag{3.2.7a,b}$$

Table 3.1 shows the standardised one-third-octave and octave band centre frequencies used in acoustic measurements:

Table 3.1: Standardised one-third octave and octave (bold characters) band centre frequencies (in hertz).

20	25	**31,5**	40	50	**63**	80	100	**125**	160	200	**250**	315	400	**500**
630	800	**1000**	1250	1600	**2000**	3150	**4000**	5000	6300	**8000**	10000	12500	**16000**	20000

3.2.4 RANDOM NOISE

As referred to by [12]: "*Many generators of sound produce noise rather than pure tones. Whereas pure tones and other periodic signals are deterministic, noise is a stochastic or random phenomenon. Stationary noise is a stochastic signal whose statistical properties do not change with time.*

White noise *is stationary noise with a flat spectral density, that is, constant mean square value per hertz. The term white noise is an analogy to white light. When white noise is passed through a band pass filter, the mean square of the output is directly proportional to the bandwidth of the filter. It follows that when white noise is analysed with constant percentage filters, the mean square of the output is proportional to the centre frequency of the filter. For example, if white noise is analysed with a bank of octave band filters, the mean square values of the output signals of two adjacent filters differ by a factor of two.*

Pink noise *is stationary noise that has the same mean square value in bands with constant relative width, e.g. octave bands. Thus compared with white noise low frequencies are emphasised by pink noise; hence the name, which is an analogy to an optical phenomenon. It follows that the mean square value of pink noise in octave bands is three times larger than the mean square value of the noise in one-third octave bands*."

For laboratory tests, white noise is more commonly used, since the constant mean square value per hertz enables easy comparison between frequencies.

3.2.5 SOUND PRESSURE LEVEL (SPL) AND DECIBELS

The human ear can withstand sound pressure variations in an exponential range (more than a million times from the lowest to the highest value). Due to this wide range, acoustic quantities and sound pressure are typically measured on a logarithmic scale. "This is also used because the subjective impression of how loud noise sounds correlates much better with a logarithmic measure of the sound pressure than with the sound pressure itself.[12]" The unit in the scale is called decibel (dB), and is a relative measure that always requires a reference quantity. The result is called a level.

The sound pressure level (also known as SPL) is the difference between the value of the total pressure and static pressure, which is what we hear.

$$L_p = 10\log_{10}\left(\frac{p_{rms}^{\;2}}{p_0^{\;2}}\right) = 20\log_{10}\left(\frac{p_{rms}}{p_0}\right) \qquad (3.2.8)$$

Where:

L_p – sound pressure level, in dB;
p_{rms} – root-mean-square pressure of a sound wave, in Pa;
p_0 – static pressure, in Pa.

p_{rms} is the root-mean-square pressure, or rms pressure, of which the SI unit is N/m^2, more commonly known as Pascal (Pa) which is defined as:

$$p_{rms} = \sqrt{\overline{p^2(t)}} = \left(\lim_{T \to \infty} \frac{1}{T} \int_0^T p^2(t) dt \right) \tag{3.2.9}$$

It may also be defined by the relation with the pressure amplitude of a sound wave:

$$p_{rms} = \frac{p_{\max}}{\sqrt{2}} \tag{3.2.10}$$

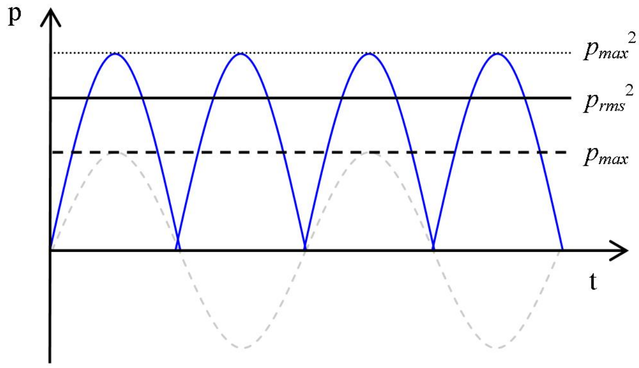

Fig. 3.3: Relationship between sound pressure, maximum pressure and rms pressure of a sound wave

The value p_0 is the reference sound pressure is 20 μPa for sound waves in air. For the sake of simplification, the base 10 logarithm of values shall be expressed as log instead of $\log_{10}$, and p_{rms} shall be simply defined as p.

Some typical sound pressure levels are given in Fig. 3.4.

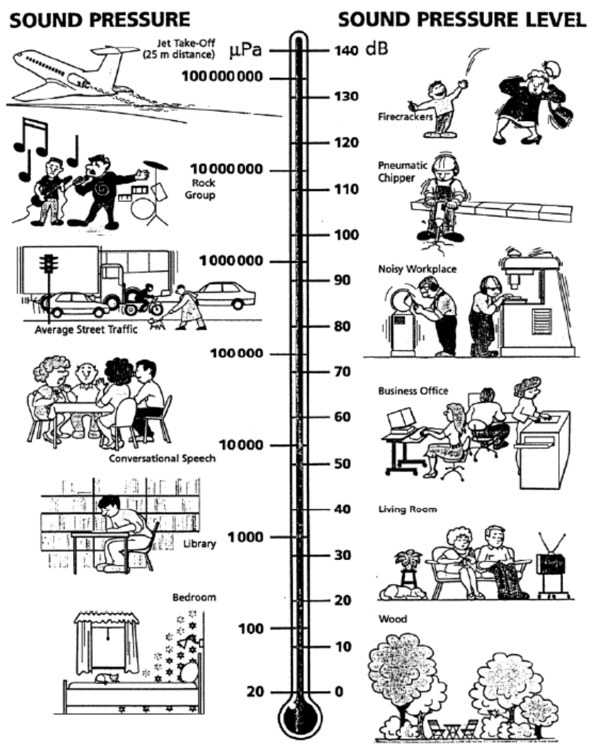

Fig. 3.4: Typical sound pressure levels. Source: Brüel & Kjaer

3.3 SOUND WAVES IN FLUIDS

As referred to by Jacobsen et al. [12], fluids, that is, gases and liquids, are characterised by the lack of constraints that oppose deformation. Fluids are unable to transmit shearing forces and react against a change of shape due solely to inertia, and a change in volume, on the other hand, is translated into a change in pressure. "*Sound waves are compressional oscillatory disturbances that propagate in a fluid. The waves involve molecules of the fluid moving back and forth in the direction of propagation (with no net flow), accompanied by changes in the pressure, density and temperature* […]." Sound waves are longitudinal waves, and so particles move back and forth in the direction of propagation, as opposed to transverse waves (i.e. bending waves in beams) in which particles vibrate perpendicular to the direction of propagation.

"*Sound waves exhibit a number of phenomena that are characteristics of waves* […]. *Waves propagating in different directions* interfere; *waves will be* reflected *by a rigid surface and more or less* absorbed *by a soft one; they will be* scattered *by small obstacles; because of* diffraction *there will only partly be shadow behind a screen; and if the medium is inhomogeneous for instance because of temperature gradients the waves will be* refracted*, which means that they change direction as they propagate. The speed with which sound waves propagate in fluids is independent of the frequency, but other waves of interest in acoustics, bending waves on plates and beams, for example, are* dispersive, *which means that the speed of such waves depends on the frequency content of the waveform*"[12].

3.3.1 THE WAVE EQUATION

The wave equation that describes the propagation of small acoustic disturbances through a homogeneous, inviscid, isotropic, compressible fluid may be written in Cartesian coordinates (x, y, z) in terms of the variation of pressure p, about the equilibrium pressure, is governed by the following principles:

- the conservation of mass;
- the local longitudinal force caused by local pressure difference is balanced by the inertia of the medium;
- sound acts almost as an adiabatic phenomenon, and so there is no heat flow.

This takes the form:

$$\nabla p = \frac{\partial^2 p}{\partial x^2} + \frac{\partial^2 p}{\partial y^2} + \frac{\partial^2 p}{\partial z^2} = \frac{1}{c^2}\frac{\partial^2 p}{\partial t^2} \tag{3.3.1}$$

Where:

c – frequency-independent speed of sound, in m/s;
p – sound wave pressure, in Pa;
x,y,z – Cartesian co-ordinates, in m;
t – time, in s.

According to Fahy [13], the same equation is applied to describe variations in fluid density and temperature, and also describes particle displacement, velocity and acceleration:

$$c^2 = \left(\gamma P_0 / \rho_0\right) \tag{3.3.2}$$

Where:

c – frequency-independent speed of sound, in m/s;
P_0 – mean fluid pressure (101.3 kPa for air under normal ambient conditions);
ρ_0 – mean density, in kg/m^3;
γ – adiabatic bulk modulus of the fluid (≈ 1.401 for air).

The adiabatic bulk modulus can also be expressed in terms of the gas constant R, the absolute temperature T_K, and the equilibrium density of the medium, which in turn translates to:

$$c^2 = \gamma R T_K \tag{3.3.3}$$

Where:

c – frequency-independent speed of sound, in m/s;
R - gas constant ($\approx 287\ \mathrm{J{\cdot}kg^{-1}K^{-1}}$ for air);
T_K – absolute temperature, in Kelvin;
γ – adiabatic bulk modulus of the fluid (≈ 1.401 for air).

At 293.15 K = 20°C the speed of sound in air is 343 m/s. Under normal ambient conditions (20°C, 101.3 kPa) the density of air is 1,204 kg.m^{-3}. Note that the speed of sound of a gas depends only on the temperature, not on the static pressure.

The general solution of the wave equation may be expressed in terms of various coordinate systems: it is separable in all common forms such as rectangular Cartesian, cylindrical, spherical and elliptical. When studying the interaction between plane structures and fluids, the most useful form of the equation is a two-dimensional form involving only variations in two orthogonal directions, in association with simple harmonic time dependence:

$$\frac{\partial^2 p}{\partial x^2} + \frac{\partial^2 p}{\partial y^2} = -\left(\frac{\omega}{c}\right)^2 p = -k^2 p \tag{3.3.4}$$

This is known as the two-dimensional Helmholtz equation, in which p is measured in Pa k is the wave number (See Chapter 3.2.2):

$$k = 2\pi/\lambda \tag{3.3.5}$$

Equation (3.3.4) is linear due to the fact that higher order terms are neglected and also because $\partial^2/\partial x^2$ and $\partial^2/\partial t^2$ are linear operators. This important property implies that sinusoidal source will generate a sound field in which the pressure at all positions varies sinusoidally. It also implies linear superposition: sound waves do not interact; they simply pass through each other (see Fig. 3.5).

Due to the various possible sound fields that may occur, one must add some additional information about the sources that generate the sound field, surfaces that reflect or absorb sound, objects that scatter sound, etc. This is called the boundary conditions, which are mostly expressed in terms of the particle velocity. In one example, given in [12], the normal component of the particle velocity u is zero on a rigid surface. Therefore an additional equation is needed to relate the particle velocity to the sound pressure, a relation known as Euler's equation of motion:

$$\rho \frac{\partial u}{\partial t} + \nabla p = 0 \tag{3.3.6}$$

which is simply Newton's second law of motion for a fluid. The operator ∇ is the gradient (the spatial derivative $(\partial/\partial x, \partial/\partial y, \partial/\partial z)$). Note that the particle velocity is a vector, unlike the sound pressure, which is a scalar.

3.3.2 Plane Sound Waves

According to [12], plane waves are waves in which any acoustic variable at a given time is a constant on any plane perpendicular to the direction of propagation. In a limited area at a far enough distance from a sound source in free space so that the curvature of the spherical wavefront is negligible, the wave can be assumed as locally plane. The plane wave is a solution to the one-dimensional wave equation,

$$\frac{\partial^2 p}{\partial x^2} = \frac{1}{c^2}\frac{\partial^2 p}{\partial t^2} \tag{3.3.7}$$

That can be translated into the solution:

$$p = f_1(ct - x) + f_2(ct + x) \tag{3.3.8}$$

Where f_1 and f_2 are arbitrary functions in which the argument of f_1 constant and if t increases with x, the first term of this expression therefore describes an undisturbed propagation of one wave moving in the positive direction, being the second term another wave that propagates in the opposite direction, as described in Fig. 3.5.

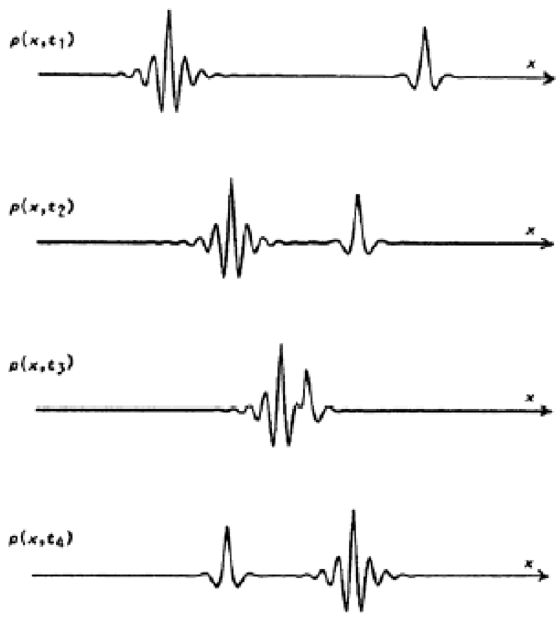

Fig. 3.5: Two plane waves travelling in opposite directions and passing each other. Source:[12]

3.3.2.1 Harmonic Plane Waves

The special case of a harmonic plane progressive wave is of great importance. As written in [12]: "Harmonic waves are generated by sinusoidal sources, for example a loudspeaker driven with a pure tone. A harmonic plane wave propagating in the x-direction can be written as:

$$p = p_1 \sin\left(\frac{\omega}{c}(ct - x) + \varphi\right) = p_1 \sin(\omega t - kx + \varphi) \tag{3.3.9}$$

Where:

ω – angular (or radian) frequency, in rad/s ($\omega = 2\pi f$);
k – angular wavenumber, rad.m/s^2 (k = ωc);
p_1 – amplitude of the wave, in Pa;
φ – phase angle, in degrees (º).

At any position in this sound field the sound pressure varies sinusoidally with the angular frequency ω, and at any fixed time the sound pressure varies sinusoidally with x with the spatial period:

$$\lambda = \frac{c}{f} = \frac{2\pi c}{\omega} = \frac{2\pi}{k} \qquad (3.3.10)$$

The quantity λ is the wavelength, which is defined as the distance travelled by the wave in one cycle. Note that the wavelength is inversely proportional to the frequency. […] In rough numbers the audible frequency range goes from 16 Hz to 16 kHz, which leads to the conclusion that acousticians are faced with wavelengths (in air) in the range from 17 m at the lowest audible frequency to 17 mm at the highest audible frequency. Since the efficiency of a radiator of sound or the effect of an obstacle on the sound field depends very much on its size expressed in terms of the acoustic wavelength, it can be realised that the wide frequency range is one of the challenges in acoustics. It simplifies the analysis enormously if the wavelength is very long or very short compared with typical dimensions.

Fig. 3.6: The sound pressure in a plane harmonic wave at two different instants of time. Source:[12]

3.3.2.2 COMPLEX NOTATION

Sound fields are often studied frequency by frequency. As already mentioned, linearity implies that a sinusoidal source with the frequency ω will generate a sound field that varies harmonically with this frequency at all positions. Since the frequency is given, all that remains to be determined is the amplitude and phase at all positions. This leads to the introduction of the complex exponential representation, where the sound pressure is written as a complex function of the position multiplied with a complex exponential. The former function takes account of the amplitude and phase, and the latter describes the time dependence. Thus at any given position the sound pressure can be written as a complex function of the form:

$$\hat{p} = Ae^{j\omega t} = |A|e^{j\varphi}e^{j\omega t} = |A|e^{j(\omega t+\varphi)} \qquad (3.3.11)$$

Where:

φ – phase angle;
A – complex amplitude.

The real, physical sound pressure is the real part of the complex pressure:

$$p = \mathrm{Re}\{\hat{p}\} = \mathrm{Re}\{|A| e^{j(\omega t + \varphi)}\} = |A| \cos(\omega t + \varphi) \qquad (3.3.12)$$

Since the entire sound field varies as $e^{j\omega t}$, the operator $\partial/\partial t$ can be replaced by $j\omega$ (because the derivative of $e^{j\omega t}$ with respect to time is $j\omega e^{j\omega t}$), and the operator $\partial^2/\partial t^2$ can be replaced by $-\omega^2$. It follows that Euler's equation of motion can now be written:

$$j\omega\rho\hat{u} + \hat{p} = 0 \qquad (3.3.13)$$

And the wave equation is simplified to:

$$\frac{\partial^2 \hat{p}}{\partial x^2} + \frac{\partial^2 \hat{p}}{\partial y^2} + \frac{\partial^2 \hat{p}}{\partial z^2} + k^2 \hat{p} = 0 \qquad (3.3.14)$$

which is known as the Helmholtz equation."

3.3.3 IMPEDANCE

Impedance is, by definition, the ratio of the complex amplitudes of two signals representing cause and effect, for example, the ratio of a mechanical force to the resulting particle vibration velocity, or in the scope of this work, the ratio of sound pressure to particle velocity, which is known as specific acoustic impedance [14], or characteristic impedance [12]. It is important in describing the reflection and transmission of sound on a partition of a room (i.e.: wall or ceiling).

As described in [12] "*…the term has been coined from the verb 'impede' (obstruct, hinder), indicating that it is a measure of the opposition to the flow of current etc. The reciprocal of the impedance is the* admittance*, coined from the verb 'admit' and indicating lack of such opposition.*"

It is therefore defined by:

$$Z = \frac{p}{v} \qquad (3.3.15)$$

Where:

Z – specific acoustic impedance, in rayl;
p – sinusoidally time-varying sound pressure on the boundary, in Pa;
v – sinusoidally time-varying particle velocity, in m/s.

According to [14], "*…in free space and at large distances from a source, any wave approaches plane wave propagation and the characteristic impedance of the wave always tends to ρc, being ρ the mean density of the fluid and c the sound propagation velocity in the medium*." This is also true for diffuse sound fields (in enclosed spaces). The SI units for specific acoustic impedance are Pa.s/m. This combination of units has been given the name rayl, in honour of Lord Rayleigh, who wrote the famous book on acoustics[4]: i.e., 1 rayl ≡ 1Pa.s/m.

3.4 SOUND TRANSMISSION BETWEEN FLUIDS WITH NORMAL INCIDENCE

The speed of sound is much higher in liquids than in gases. For example, the speed of sound in water is about 1480 m/s. The density of liquids is also much higher; the density of water is about 1000 kg/m^3. Both the density and the speed of sound depend on the static pressure and the temperature, and there are no simple general relations corresponding to Equation (3.3.14).

When a sound wave in one fluid is incident on the boundary of another fluid, for example, when a sound wave in air is incident on the surface of water, it will be partly reflected and partly transmitted. Using the example suggested in [12], a plane wave in fluid 1 strikes the surface of fluid 2 at normal incidence as shown in Fig. 3.6. Anticipating a reflected wave we can write:

$$\hat{p}_1 = p_i e^{j(\omega t - kx)} + p_r e^{j(\omega t + kx)} \tag{3.4.1}$$

for fluid 1 and,

$$\hat{p}_2 = p_t e^{j(\omega t - kx)} \tag{3.4.2}$$

for fluid 2. There are two boundary conditions at the interface: the sound pressure must be the same in fluid 1 and in fluid 2 (otherwise there would be a net force), and the particle velocity must be the same in fluid 1 and in fluid 2 (otherwise the fluids would not remain in contact):

$$p_i + p_r = p_t \qquad \text{and} \qquad \frac{p_i - p_r}{\rho_1 c_1} = \frac{p_t}{\rho_2 c_2}\,. \tag{3.4.3a, b}$$

Where:

p_i – incident sound wave pressure, in Pa;
p_r – reflected sound wave pressure, in Pa;
p_t – transmitted sound wave pressure, in Pa;
ρ_1 – density of fluid 1, in kg/m^3;
c_1 – sound propagation speed in fluid 1, in m/s;
ρ_1 – density of fluid 2, in kg/m^3;
c_1 – sound propagation speed in fluid 2, in m/s.

Combining these equations gives:

$$\frac{p_r}{p_i} = Q = \frac{\rho_2 c_2 - \rho_1 c_1}{\rho_2 c_2 + \rho_1 c_1} \tag{3.4.4}$$

which shows that the wave is almost fully reflected in phase ($Q \approx 1$) if $\rho_2 c_2 >> \rho_1 c_1$, almost fully reflected in antiphase ($Q \approx -1$) if $\rho_2 c_2 << \rho_1 c_1$, and not reflected at all if $\rho_2 c_2 = \rho_1 c_1$, irrespective of the individual properties of c_1, c_2, ρ_1 and ρ_2.

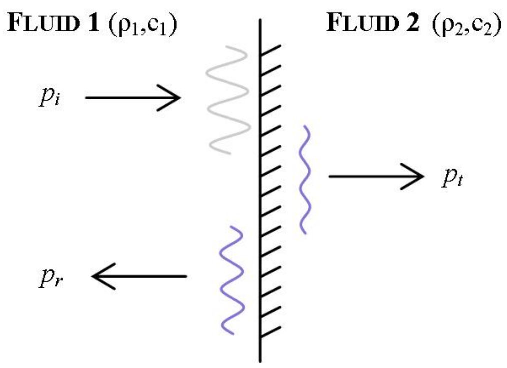

Fig. 3.7: Reflection and transmission of a plane wave incident on the interface between two fluids.

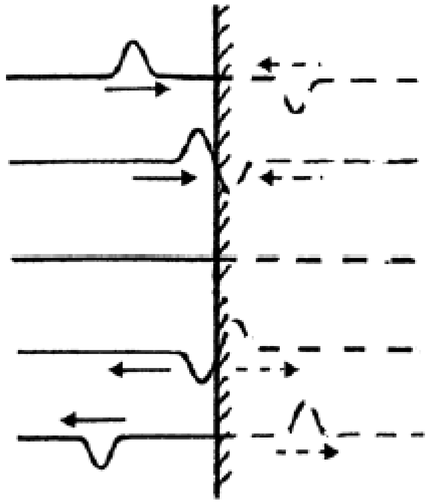

Fig. 3.8: Reflection of a pressure wave at the interface between a medium of high characteristic impedance and a medium of low characteristic impedance. Source: [12]

Because of the significant difference between the characteristic impedances of air and water (the ratio is about 1:3600) a sound wave in air that strikes a surface of water at normal incidence is almost completely reflected, and so is a sound wave that strikes the air-water interface from the water, but in the latter case the phase of the reflected wave is reversed, as shown in Fig. 3.8.

3.5 SOUND TRANSMISSION BETWEEN FLUIDS WITH OBLIQUE INCIDENCE

According to Barron [2], in most cases in noise control, sound waves may strike a surface at various angles of incidence. Considering a plane sound wave which strikes the interface between two fluids at an angle of incidence θ_i with the normal to the interface, as shown in Fig. 3.9:

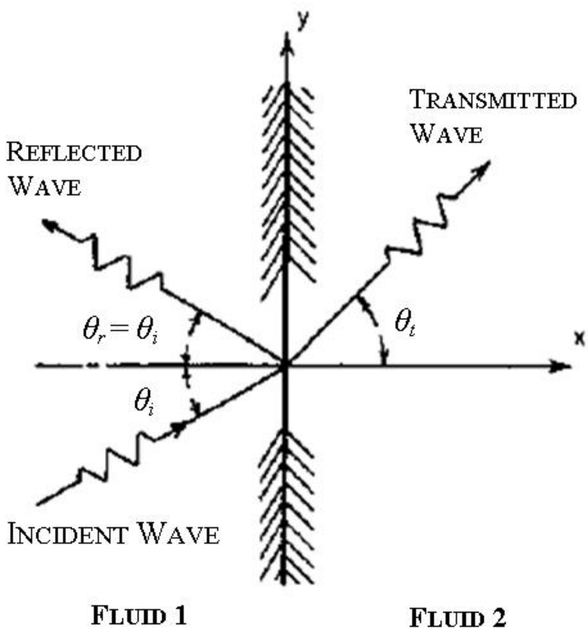

Fig. 3.9: Transmission of sound from one fluid into another for oblique incidence of the sound wave. Adapted from: [2]

The expression for a wave moving at an angle θ with the normal to the interface may be written in the following form:

$$p(x,y,t) = Ae^{j(\omega t - kL)} \tag{3.5.1}$$

The quantity L is related to the coordinates by the following expression, as illustrated in Fig. 3.9:

$$L = x\cos\theta + y\sin\theta \tag{3.5.2}$$

The corresponding expression for the acoustic wave in medium 1 may be written as follows:

$$p_1(x,y,t) = A_1 e^{j[\omega t - k_1(x\cos\theta_i - y\sin\theta_i)]} + B_1 e^{j[\omega t + k_1(x\cos\theta_r - y\sin\theta_r)]} \tag{3.5.3}$$

It has been assumed that the angle of reflection is equal to the angle of incidence, or $\theta_r = \theta_i$. The expression for the wave moving in medium 2 is:

$$p_2(x,y,t) = A_2 e^{j[\omega t - k_2(x\cos\theta_t - y\sin\theta_t)]} \tag{3.5.4}$$

The angle of transmission θ_t is related to the angle of incidence θ_i through *Snell's law* (see Fig. 3.9). For the sound wave to remain a plane sound wave, the wave must travel the distance L_1 in the same time as it travels the distance L_2, as illustrated in Fig. 3.11.

$$L_1 = d\sin\theta_i = c_1\Delta t \tag{3.5.5a}$$

$$L_2 = d\sin\theta_t = c_2\Delta t \tag{3.5.5b}$$

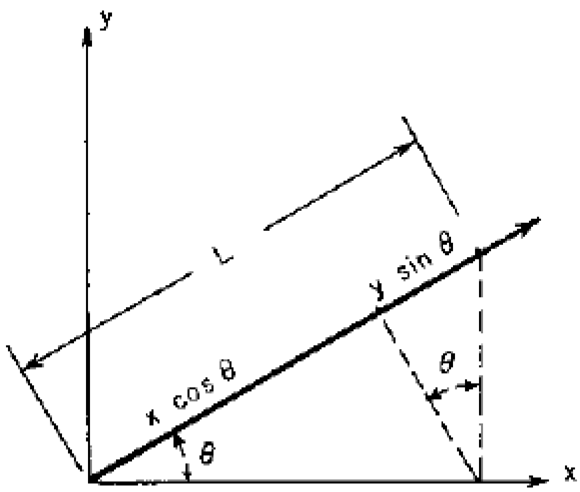

Fig. 3.10: Relationship between the x- and y-coordinates and the coordinate L in the direction of propagation of an oblique sound wave. Source:[2]

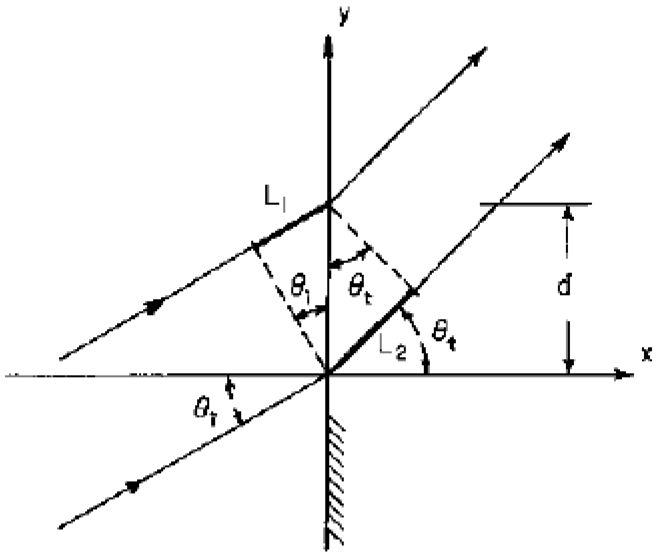

Fig. 3.11: Wave front striking an interface at oblique incidence. Source:[2]

By dividing the two expressions we obtain Snell's law:

$$\frac{\sin\theta_i}{\sin\theta_t} = \frac{c_1}{c_2} = \frac{k_2}{k_1} \tag{3.5.6}$$

From the Snell law expression, Equation (3.5.6), we note that there can be a critical angle of incidence θ_{cr} for which the transmitted wave will make an angle of 90º with the normal to the interface. For this condition, sin θ_t = sin 90º = 1. The expression for the critical angle of incidence may be found by making this substitution into the Snell law expression:

$$\sin\theta_{cr} = \frac{c_1}{c_2} \tag{3.5.7}$$

A critical angle of incidence exists only if $c_1 < c_2$. If the actual angle of incidence is equal to or greater than the critical angle of incidence, then no acoustic energy will be transmitted into the second material. Fig. 3.11 and 3.12 show how sound is transmitted into water from an airborne source:

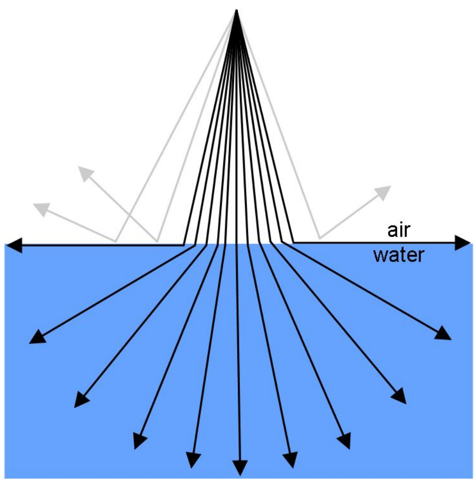

Fig. 3.12: The apparent radiation pattern of an airborne source as observed by a distant waterborne receiver. Adapted from: [15]

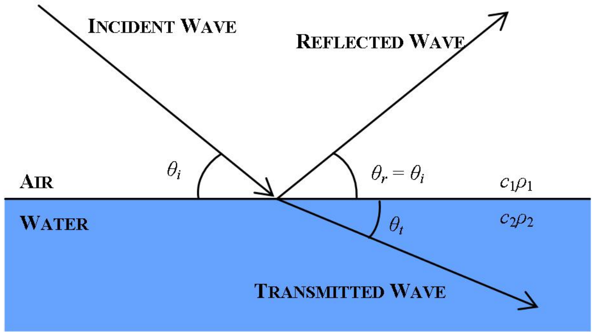

Fig. 3.13: Reflection and transmission of a plane wave from an airborne source to water.

From the Snell law expression Barron [2] has defined an expression for the sound power transmission coefficient due to the change in media with oblique incidence:

$$a_t = \frac{4Z_1Z_2\cos\theta_i\cos\theta_t}{(Z_1\cos\theta_t + Z_2\cos\theta_i)^2} \qquad (3.5.8)$$

Where:

a_t – sound power transmission coefficient (Note: if the angle of incidence is greater than the critical angle, $a_t = 0$)
Z_1 – specific acoustic impedance of the source medium (in rayl or Pa.s/m);
Z_2 – specific acoustic impedance of the receiving medium (in rayl or Pa.s/m);
θ_i – angle of incidence of the sound wave on the interface;
θ_t – angle of transmission of the sound wave from the interface.

From this formula the sound power transmission loss due to the change in media with oblique incidence R_{Snell} (dB) may be calculated for a single angle of incidence:

$$R_{Snell} = 10\log\left(\frac{1}{a_t}\right) \qquad (3.5.9)$$

3.6 SOUND INSULATION OF SINGLE PANEL PARTITIONS

When a sound wave hits a wall or partition separating two adjacent rooms its energy is divided into various parts – some is reflected back into the source room, some is converted into heat inside the material of the separating partition, another part is transmitted through the surrounding structure and some energy is transmitted into the adjacent room.

The transmission of sound energy can be defined by the sound transmission coefficient τ, which is the relation between the incident sound pressure and the transmitted sound pressure. Since the usual values of τ are very small, sound insulation R with the unit decibel is used. This is defined by:

$$R = 10\log\frac{P_{receiving\ room}}{P_{source\ room}} = 10\log\frac{1}{\tau} \qquad \text{(dB)} \qquad (3.6.1)$$

Where:

R –	sound insulation, in dB;
$P_{receiving\ room}$ –	sound pressure in the receiving room, in Pa;
$P_{source\ room}$ –	sound pressure in the source room, in Pa;
τ –	sound transmission coefficient.

According to Fahy [13], in order to limit the transmission of sound energy from one partition to another there are two main approaches: the first involves the absorption of this energy by materials that receive the sound waves and convert these into heat. Examples of this method include room wall absorbers and ventilation duct lining. The second approach is to force waves to be reflected from the transmission path by introducing a large change of acoustic impedance in that path. Examples include partitions such as solid walls and noise control enclosures, or sound booths for musicians. Due to the nature of this work, only the sound insulation of single panel partitions shall be addressed, since the lining of the liquid-filled panel shall be considered acoustically ‘transparent’, and therefore shall not influence calculations significantly.

3.6.1 Sound Insulation between Two Rooms

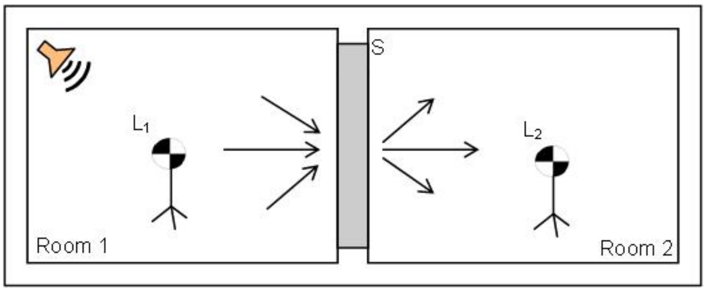

Fig. 3.14: Measuring airborne sound transmission between two rooms

Assuming that the sound fields of both rooms are diffuse i.e. the sound energy density is the same in all points of the room and sound propagates in all directions with the same probability, it is possible to relate the sound insulation of the separating wall with the sound pressure levels measured in the source room (room 1) and the receiving room (room 2), which are L_1 and L_2 respectively, in dB. The following expression can be obtained:

$$R' = L_1 - L_2 + 10\log\frac{S}{A_2} \qquad \text{(dB)} \qquad (3.6.2)$$

In which *R’* is the apparent sound transmission loss, where the various sound transmission paths that exist between the two rooms are taken into account, S is the partition surface area in m^2 and A_2 the equivalent absorption area of the receiving room, calculated by Sabine’s formula:

$$A_2 = \frac{0{,}16V}{T} \qquad (\text{m}^2) \qquad (3.6.3)$$

The reverberation time is obtained by measuring the sound decay in the receiving room (the amount of time it takes for the sound pressure to drop 60 dB after it has been turned off), T, in seconds. V is the receiving room total volume in m^3.

The additional transmission paths referred to before are called flanking transmission paths, where sound propagates not only directly through the partition but also through the connecting structure, as seen in the following illustration.

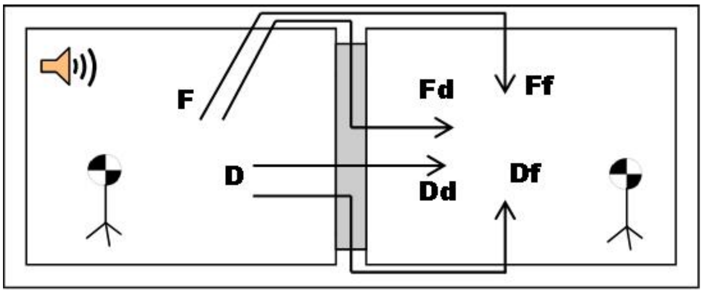

Fig. 3.15: Transmission of airborne sound between two adjacent rooms

The picture above shows the typical sound transmission paths for each structure connection. These either go directly through the partition (**D** going in or **d** going out) or travel around the partition through the surrounding structure (**F** going in and **f** going out). The apparent sound transmission takes all of these paths into account, since measurements are taken inside both the source and receiving room.

The sound reduction index or sound insulation is measured in third octave bands in a frequency range of 50 Hz up to 5000 Hz. The main reason for measuring such low frequencies is to account for the effect of these frequencies in the sound insulation of lightweight constructions. These solutions have been used more frequently in new buildings, in detriment of heavy constructions that would normally be used for sound insulation.

Sound pressure levels are measured as an average of several microphone positions in the same room, measured over a certain time span. The reverberation time of the receiving room is also calculated in order to calculate the equivalent absorption area, according to Sabine's formula.

3.6.2 SOUND INSULATION PREDICTION MODELS

There are various prediction models that calculate the sound insulation of partitions between rooms, either for single panels, double-leaf panels or multiple panels, each taking into account different parameters and neglecting others. When using such prediction methods in order to find a solution for a design, the acoustician must always take into account how the model works and what parameters are used, so that there are no undesired effects with the constructed partition that may increase costs and waste precious time during construction. Different prediction models are used in this work, and they shall be described in the following section. Please note that these prediction models are only used to compare test results and for obtaining reference values. A liquid layer will not function in the same way as a rigid panel because it is a non-rigid panel – bending waves, for example, will have no effect on the non-rigid panel, and neither will the critical frequency, as shall be seen in Chapter 5.

For single panels the sound reduction index or transmission loss is influenced by four factors: size, bending stiffness, mass and damping [16]. In the audible frequency spectrum a simple partition acting as an infinite plate and submitted to a diffuse sound field will not behave according to the same one law. Therefore it is important to define certain values of frequency that limit the different frequency regions of panel behaviour, as shown in Fig. 3.16:

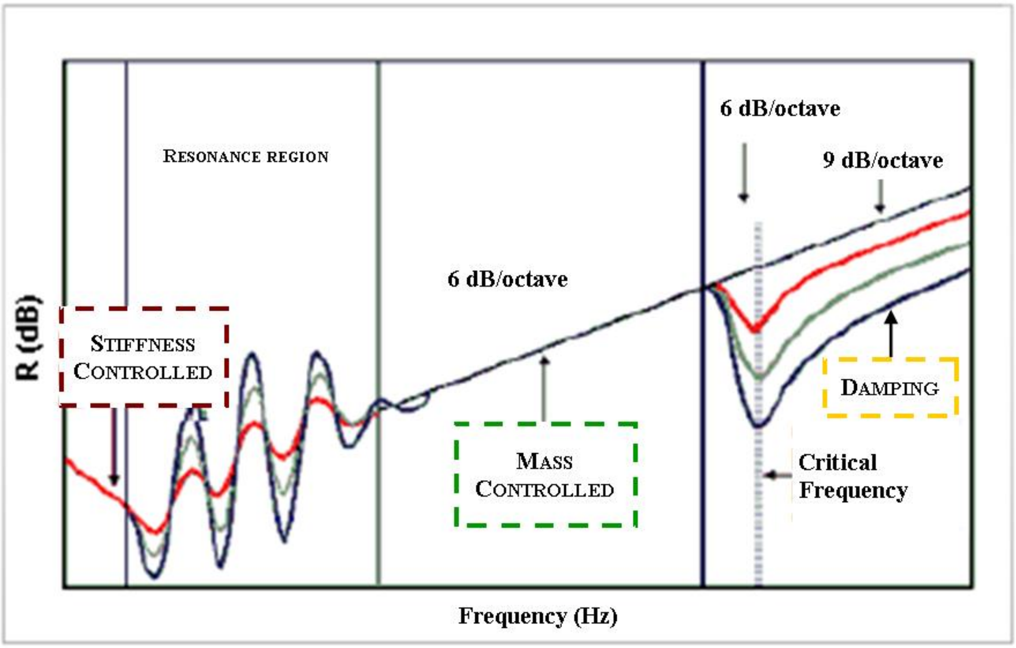

Fig. 3.16: Sound reduction index of a single panel according to frequency.

The *resonance frequency* originates a standing bending wave which fits the element. If the considered element has simply supported edges:

$$f_0 = 0{,}45 \cdot d \cdot \sqrt{\frac{E}{\rho}} \cdot \left[\left(\frac{h}{l}\right)^2 + \left(\frac{g}{b}\right)^2\right] \tag{3.6.4}$$

If the considered element has clamped edges:

$$f_0 = 0{,}45 \cdot d \cdot \sqrt{\frac{E}{\rho} \cdot \left[\left(\frac{d}{l}\right)^2 + \left(\frac{g}{b}\right)^2\right]} \tag{3.6.5}$$

Where:

f_0 - first resonance frequency (Hz);
h - thickness of the element (m);
E - Young modulus (Pa);
ρ - density (kg/m^3);
d, g - natural numbers ($h = g = 1$ for the first resonance frequency);
l , b - dimensions of the element (m).

At the frequency of first panel resonance, there is a large decrease in sound insulation, reaching a minimum determined in part by the damping in the system.

The *critical frequency* or coincidence frequency is related to the coincidence effect. This frequency is the one whose wavelength projected on the elements' plane is the same as the wavelength of the free bending waves of the element (material characteristics). This means that the freely travelling plane wave that hits the element will transmit its energy easily to the panel and because of this there will be a drop in sound insulation. It can be obtained from the following formula:

$$f_c = \frac{c^2}{1{,}8 \cdot c_L \cdot h} \tag{3.6.6}$$

At critical frequency the sound insulation is controlled by damping.

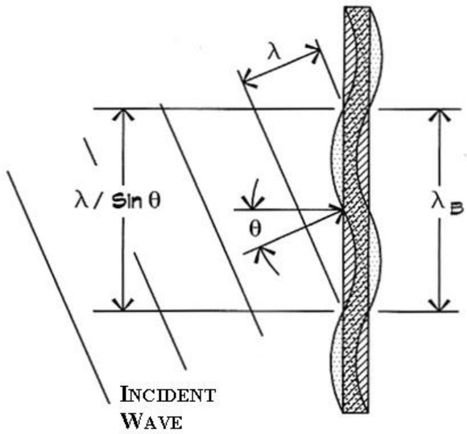

Fig. 3.17: The coincidence effect. Adapted from [16].

3.6.2.1 MODIFIED "MASS LAW"

Along the audible frequency range a single panel will have different sound reduction index values (*R* in dB), or sound insulation index, which can be divided in the following manner (see Fig. 3.16):

$f < f_0$:

Below this frequency the sound transmission of the element is determined by the rigidity or stiffness of the wall and the *R* value is a function of (K/ω) with:

K – Rigidity of the element (N/m);
ω – Angular frequency, ω = 2πf (rad/s)

We can see that the *R* value decreases with the increase in frequency – the frequency appears in the denominator of the equation.

$f_0 > f > f_c$:

Above the resonance frequency the vibration of the element is controlled by the mass. For a diffuse sound field and by integration of the equation over all relevant angles of incidence (up to 78º) [16], we can obtain the sound insulation of a given element by means of the following simplified expression known as the mass law, where R increases 6 dB/octave:

$$R = 20 \cdot \log(m \cdot f) - 47 \quad (3.6.7)$$

Where:

R - Sound insulation of a building element (dB)
m - Surface mass of a building element (kg/m^2)
f - Frequency (Hz)

$f > f_c$:

Above this frequency the increase of sound insulation is greater. It can be expressed by the following equation:

$$R = 20 \cdot \log\left(\frac{\pi \cdot m \cdot f}{\rho_0 \cdot c_0}\right) + 10 \cdot \log\left(\frac{f}{f_c}\right) + 10\log(\eta) - 2 \qquad (3.6.8)$$

Where:

R - Sound insulation of a building element (dB);
m - Surface mass of a building element (kg/m^2);
f – Frequency of incident wave (Hz);
ρ_0 – density of air (kg/m^3);
c_0 – sound wave propagation speed in air (m/s);
f_c – critical frequency (Hz);
η – total loss factor of the building element.

In this region of the equation the R value increases about 9 dB/octave.

For very thick walls another expression by Sharp must be used, since the predominant sound transmission ceases to be through flexural waves, but through shear waves. The cross-over frequency where this phenomenon exists is when:

$$f_s = \frac{c_0{}^2 \cdot (1 - \sigma)}{59 \cdot h^2 \cdot f_c} \qquad (3.6.9)$$

Where:

f_s - Shear wave cross-over frequency (Hz)
c_0 - sound wave velocity in air (m/s)
σ - building element radiation factor
h - thickness of the building element (m)
f_c - critical frequency (Hz)

With very thick walls, when $f_s < f_c$ there ceases to exist a sound insulation dip in the vicinity of the critical frequency. In frequencies above f_s the wall performs the same way as the mass law, but a decrease in sound insulation values of around 6 dB is expected. When $f_s > f_c$, the sound insulation curve R increases 6 dB per octave.

As referred to by Bies et al. [14], in practice panels are not of infinite extent and therefore results obtained shall correlate better if a limiting angle of incident sound waves is defined, as was above to 78º from the direction normal to the panel, therefore eliminating the transmission of waves that hit the panel between the limiting angle and 90º (direction parallel to the panel). Other models use different limiting frequencies, as is described below.

3.6.2.2 SHARP PREDICTION MODEL (1973)

Sharp showed that a good correlation between prediction and measurement in the mass controlled region is obtained by using a constant limiting angle of about 85º. Therefore the field incidence sound reduction, R, is related to the normal incidence reduction index R_N, for predictions in third-octave bands, in which $\Delta f / f = 0{,}236$, by the following expression:

$$R = R_N - 10\log\left(1{,}5 + \log_e \frac{2f}{\Delta f}\right) = R_N - 5{,}5 \quad \text{(dB)} \qquad (3.6.10)$$

Note that the mass law predictions assume that the panel is limp, and as they become thicker and stiffer, their mass law performance drops below the ideal prediction, so that in practice very few constructions will perform according to the mass law prediction.

The field incidence sound reduction index in the mass law frequency range below $0{,}5f_c$:

$$R = 20\log\left(\pi fm/(\rho c)\right) - 5{,}5 \quad \text{(dB)} \quad \text{, when } f < 0{,}5 f_c \qquad (3.6.11)$$

This equation is not valid for frequencies below 1,5 times the first panel resonance frequency, but below $0{,}5f_c$ it agrees reasonably well with measurements taken in third-octave bands.

For frequencies equal to or higher than the critical frequency, the following equation applies:

$$R = 20\log\left(\frac{\pi fm}{\rho c}\right) + 10\log\left(\frac{2\eta f}{\pi f_c}\right) \quad \text{(dB) , when } f > f_c \qquad (3.6.12)$$

Note that Equation 3.6.12 only applies until the frequency is reached at which the calculated R is equal to that calculated by using Equation 3.6.11. From this point onward the latter expression applies.

The sound reduction index between $0{,}5f_c$ and f_c is approximated by connecting with a straight line the points corresponding to the calculated R values related to both these frequencies. It is important to refer that the graph must be on the logarithmic scale on both axes.

The prediction model is summarised in the following illustration:

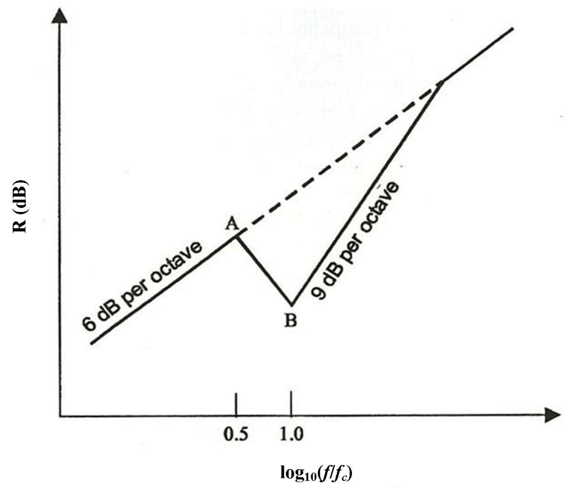

Fig. 3.18::A design chart for estimating the sound reduction index of a single isotropic panel. Adapted from:[14]

An algorithm that translates this model is described in Annex A. One should note that the first panel resonance has been omitted from calculations since it occurs most of the time below the frequency range of interest, and because the panel sound reduction around this resonance depends on the panel damping and characteristics of the incident sound field. Both of these parameters are very hard to estimate. According to [14], the best approach shall be to make the damping as large as practical and a sound reduction somewhat less than the predicted by the mass law portion of the curve should be expected at the lowest panel resonance.

3.6.2.3 Davy Prediction Model (1990)

A more recent prediction model which is claimed to be more accurate and allows the variation of the limiting angle θ_L (º) as a function of frequency and size of the panel, to be taken in account according to the following expression:

$$\theta_L = \cos^{-1}\sqrt{\frac{\lambda}{2\pi\sqrt{A_r}}} \tag{3.6.13}$$

Where:

θ_L – limiting angle, in degrees;
A_r – area of the panel, in m^2;
λ – wavelength of sound at the frequency of interest, in m.

A detailed prediction scheme is provided in Annex A, for third octave bands. According to Bies et al. [14], the Davy method seems to be more accurate than the Sharp model in lower frequencies, while the latter gives better results around the critical frequency of the panel.

3.6.2.4 INSUL

"INSUL is a program for predicting the sound insulation of walls, floors, ceilings and windows. The programme can make good estimates of the Transmission Loss (TL) in 1/3 octave bands and Weighted Sound Reduction Index (STC or R_w) for use in noise transfer calculations."[17]

The program is based on Sharp's model for single panels (see Chapter 3.6.2.2) and has been adjusted according to different types of construction and materials.

3.7 EUROPEAN STANDARDS AND PORTUGUESE LEGISLATION

3.7.1 INTERNATIONAL STANDARDS – SOUND INSULATION MEASUREMENT (ISO 140-3:1995)

Airborne sound insulation may be measured in situ or in the laboratory. In order to obtain valid results it is necessary to follow a strict and standardised measurement procedure, which in this case, a laboratory method is specified in ISO 140-3:1995, *Acoustics – Measurement of sound insulation in buildings and of building elements – Part 3: Laboratory measurement of airborne sound insulation of building elements*. This document was prepared by the Technical Committee ISO/TC 43 "Acoustics" in collaboration with CEN/TC 126 "Acoustic properties of buildings and building elements", submitted to vote and approved by CEN in 1995. In Portugal the national standard is the NP EN 20140-3:1998, which is based on ISO 140-3:1995.

The referred document refers the specific method of laboratory sound insulation measurements. Other parameters such as measurement chamber requirements or equipment specifications are specified in other international standards (see Fig. 3.18).

The results obtained can be used, according to ISO 140-3, "…to design building elements with appropriate acoustic properties, to compare the sound insulation properties of building elements and to classify such building elements according to their sound insulation capabilities."

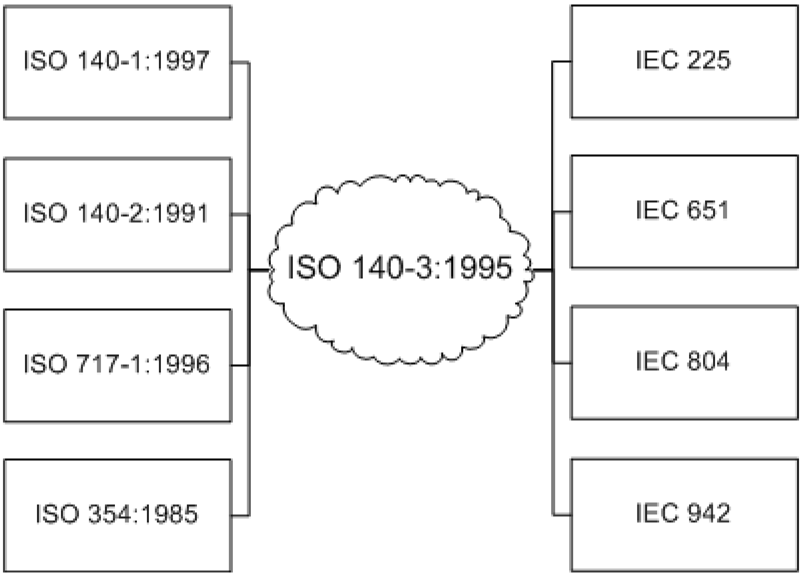

Fig. 3.19: Relevant International Standards

3.7.2 WEIGHTED SOUND REDUCTION INDEX (ISO 717-1)

This is the standard procedure that converts the various sound reduction index values per frequency band into a normalized single number rating, R_W, in dB. The measured or calculated values of R in third octave or octave bands are compared to reference curves of ISO 717-1. For third octave bands the centre frequencies range from 100 to 3150 Hz. The reference curve is shifted towards the measured values in steps of 1 dB. Each time the difference $R - R_{ref}$ is determined for each frequency band and only the negative values are considered. This process is repeated until the absolute value of the sum of all negative values is as large as possible, but not larger than 32,0 dB for third-octave bands or 10,0 dB for octave bands.

Then R_W is the value of the shifted reference curve at 500 Hz, given in full dB. Table 3.2 shows the reference values for the calculation of R_W in third-octave bands.

Table 3.2: Reference values for the calculation of R_W in third octave bands according to ISO 717-1

f (Hz)	100	125	160	200	250	315	400	500	630	800	1000	1250	1600	2000	2500	3150
Reference values in third octave bands (dB)	33	36	39	42	45	48	51	52	53	54	55	56	56	56	56	56

Fig. 3.19 and 3.20 show an example of the procedure in which the test sample has a weighted sound reduction index R_W of 30 dB:

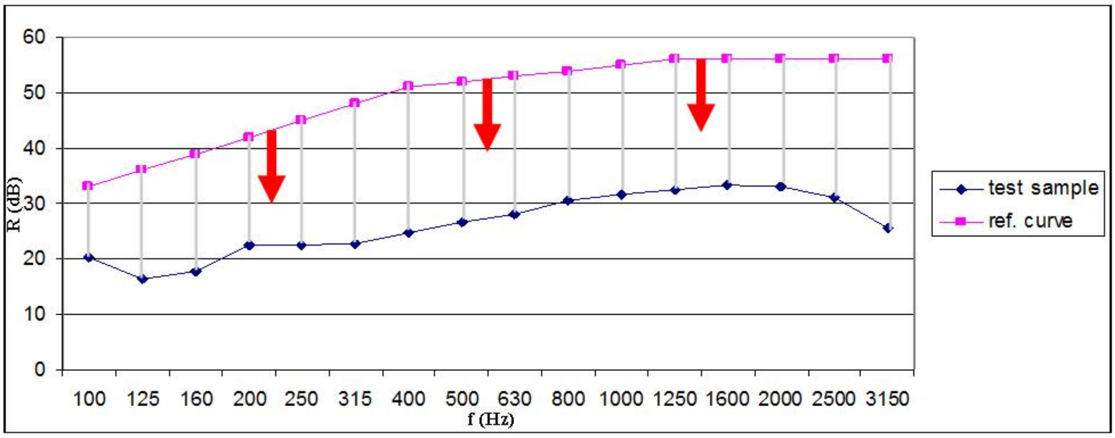

Fig. 3.20: Example of weighting the sound reduction index according to ISO 717-1(before weighting)

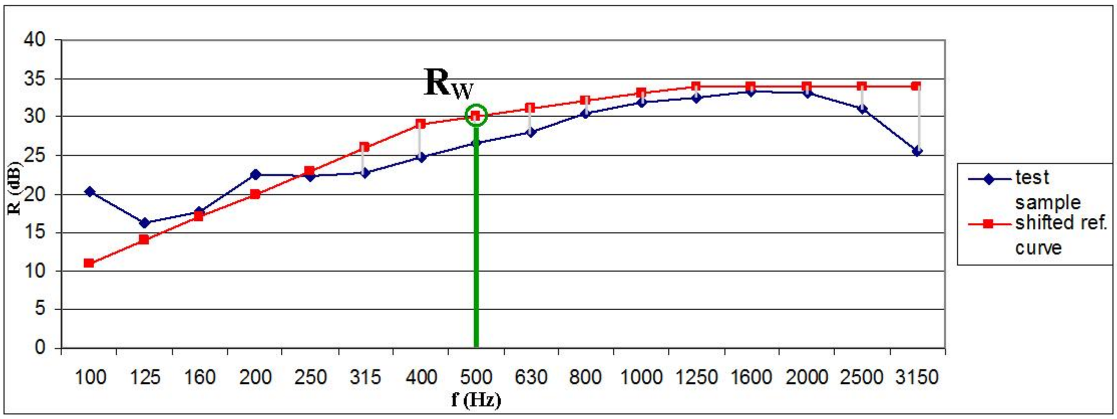

Fig. 3.21: Example of weighting the sound reduction index according to ISO 717-1(after weighting)

The single number sound reduction index R_W is usually used in noise control projects as a quick reference for a construction solution for general purposes, since current legislation uses this rating to evaluate a certain wall or partition's acoustic performance. However, this rating doesn't provide specific information about the behaviour of that solution in terms of frequency, which may be useful in certain noise control applications.

3.7.3 Portuguese Legislation

As referred in this beginning of this work (see Chapter 2.2), exposure to continuous or extremely loud noise sources can cause public health problems, and therefore, according to Portuguese law, in Article 66 of the Constitution, all people have the right to a humane and healthy environment.

In this spirit, more specific legislation has been passed to provide stricter laws on noise control in buildings and the environment.

Translating from the Decree nº 9/2007 of January 17th: "Noise and sound pollution control aimed at safeguarding human health and the well-being of the general population is the State's fundamental task, in terms of the Portuguese Republic Constitution and the Base Laws of the Environment."

According to the European Parliament and Council directive n. º 2002/49/CE, of June 25th, it was necessary to modify existing laws in order to simplify these and make them compatible with existing standards.

The existing laws on noise pollution and control are now compiled into a group of regulations called the General Regulations on Noise (*Regulamento Geral sobre o Ruído*). The main document of this group is the Decree nº 292/2000, of November 14th, also called General Noise Pollution Regulations

(*Regulamento Geral Sobre a Poluição Sonora*), which was recently updated and therefore (partially) replaced by the D.L. nº 9/2007 of January 17th. Additional legislation includes:

- D.L. nº 72/92, of April 28th;
- D. Regulamentar nº 9/92, of April 28th;
- D.L. nº 76/2002, of March 26th;
- D.L. nº 129/2002, of May 11th – Regulamento dos Requisitos Acústicos dos Edifícios – REAE;
- D.L. nº 293/2003 of November 19th.

The referred regulations determine if a building license is approved from a noise control point of view. Of course other design specialties must be coordinated into a project in order to have a sustainable building solution that may be approved by the relevant authorities.

4
Method

4.1 INTRODUCTION

The following Chapter describes the methods used to obtain significant values of airborne sound insulation of liquids. Both single panel prediction models (described in the previous Chapter) and laboratory test measurements are used and compared between each other, so that meaningful conclusions can be obtained. One should note that the sound insulation prediction methods (except Snell's law) are solely used as means of obtaining reference values of sound insulation for solid panels of the same characteristics as the liquids tested. These reference values will be used for comparisons between the various laboratory tests that have been carried out. Since the liquid layers will not have any rigidity, there will be no coincidence effect, for example.

4.2 PREDICTION MODEL CALCULATIONS

4.2.1 THE AIR-FLUID BOUNDARY (APPLYING SNELL'S LAW)

By applying the known characteristics of air and the studied fluids to Snell's Law (See Chapter 3.5), it is possible to predict in some measure the expected sound insulation (or transmission loss) in both the air-liquid and the liquid-air interfaces, when sound in air hits the liquid surface. By inserting a number of discrete sound wave incidence angles θ_i, the respective sound propagation velocities in each medium (c_1 and c_2) and the respective mean densities (ρ_1 and ρ_2), the formulas shall produce both transmission angle θ_t, and sound reduction index R (dB). Note that, as explained before, when transmission angles are equal to or above 90º the wave is reflected back into the source medium.

In the specific case of air-water interface, the data is:

Table 4.1: Sound insulation or transmission loss due to the air-water boundary, for various sound wave angles of incidence

Sound Transmission Loss from Air to Water in Various Angles of Incidence (dB)								
θ_i (º)	0	2	4	6	8	10	12	14
θ_t (rad)	0,00	0,15	0,31	0,47	0,65	0,85	1,13	>90º
θ_t (º)	0,00	8,71	17,62	26,98	37,16	48,91	64,47	>90º
a_t	0,001083	0,001071	0,001034	0,00097	0,000871	0,000723	0,000477	0
R (dB)	29,7	29,7	29,9	30,1	30,6	31,4	33,2	No Transmission!

Note that for angles of incidence above 13,3º, which corresponds to the critical angle of water, the sound wave is completely reflected. This, however, has little expression in room and building acoustics, since we can assume that a source room has a diffuse sound field and therefore sound waves come from all directions. The average of R in the air-to-water boundary is 30,9 dB.

In the water-to-air boundary, there is no critical angle of incidence, since the Snell law expression (see Chapter 3.5) gives real values for all angles of incidence. The average of the used values is also 30,9 dB, as values above 70º are disregarded. The R value increases exponentially and therefore has little expression in terms of sound transmission from water to air (when θ_i is 80º, R is 37,1 dB; when θ_i is 90º, R is 191,6 dB).

Calculations using the vegetable oil and air give the following results:

Table 4.2: Sound insulation or transmission loss due to the air-vegetable oil boundary, for various sound wave angles of incidence

Sound Transmission Loss from Air to Vegetable Oil in Various Angles of Incidence (dB)								
θi (º)	0	2	4	6	8	10	12	14
θt (rad)	0,00	0,15	0,30	0,45	0,62	0,82	1,06	>90º
θt (º)	0,00	8,42	17,01	26,00	35,71	46,74	60,68	>90º
at	0,001134	0,001123	0,001087	0,001025	0,00093	0,000789	0,000568	0
R (dB)	29,5	29,5	29,6	29,9	30,3	31,0	32,5	No Transmission!

The critical angle of incidence for the vegetable oil is 13,8 º and the average R value in the air-to-oil interface is 30,5 dB, just as the average sound insulation for the oil-to-air interface.

4.2.2 SOUND INSULATION PREDICTIONS

Using the available prediction models programmed in a Microsoft Excel™ spreadsheet (see Fig.4.1), the relevant data was introduced in order to get some grasp of how the liquid filled panel would behave if it were a rigid panel with the same physical properties. Six predictions were made in total, three for the water single panel (20 mm, 40 mm and 80 mm thick) and another three for the vegetable oil. The prediction models used are referred in more detail in Chapter 3.6.2.

Table 4.3: Input data for a water layer

Input Data – Water layer				
Model	**Parameter**	**Thickness**		
		20 mm	**40 mm**	**80 mm**
Insul	density (Kg/m^3)	1000		
	fc.m	43649		
	Loss Factor (η)	0,01		
	d (m)	0,02	0,04	0,08
Davy	m (Kg/m^2)	20	40	80
	d (m)	0,02	0,04	0,08
	c_L (m/s)	1480		
	Loss Factor (η)	0,01		
	Length (m)	5		
	Width (m)	2		
Sharp	m (Kg/m^2)	20	40	80
	e (m)	0,02	0,04	0,08
	c_L (m/s)	1480		
	Loss Factor (η)	0,01		

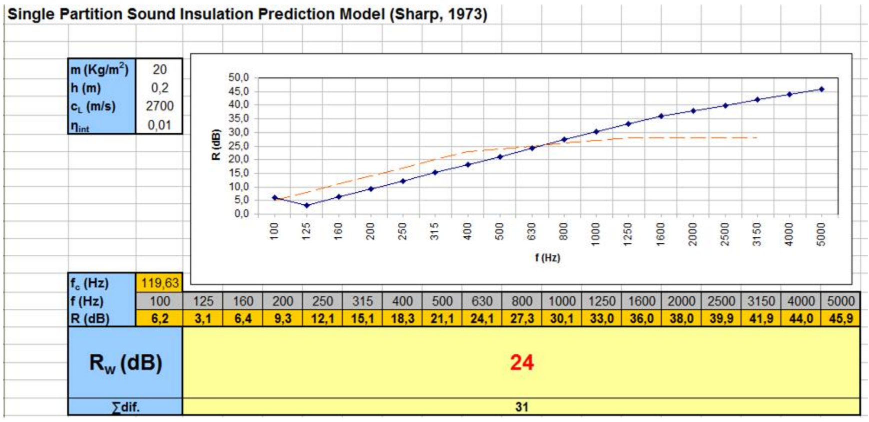

Fig. 4.1:Excel worksheet programmed with the Sharp prediction model algorithm

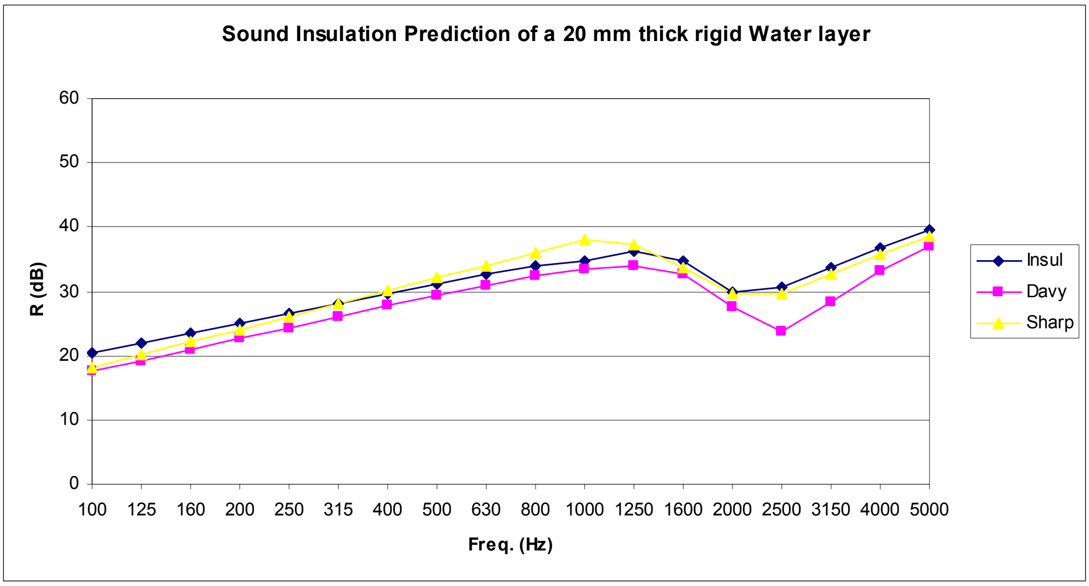

Fig. 4.2: Sound insulation prediction of a 20 mm thick rigid water layer

The calculated R_W values for the 20 mm thick rigid water layer are 33, 30 and 33 dB for Insul, Davy and Sharp models respectively. Note that the coincidence effect shall probably not occur with the non-rigid water layer.

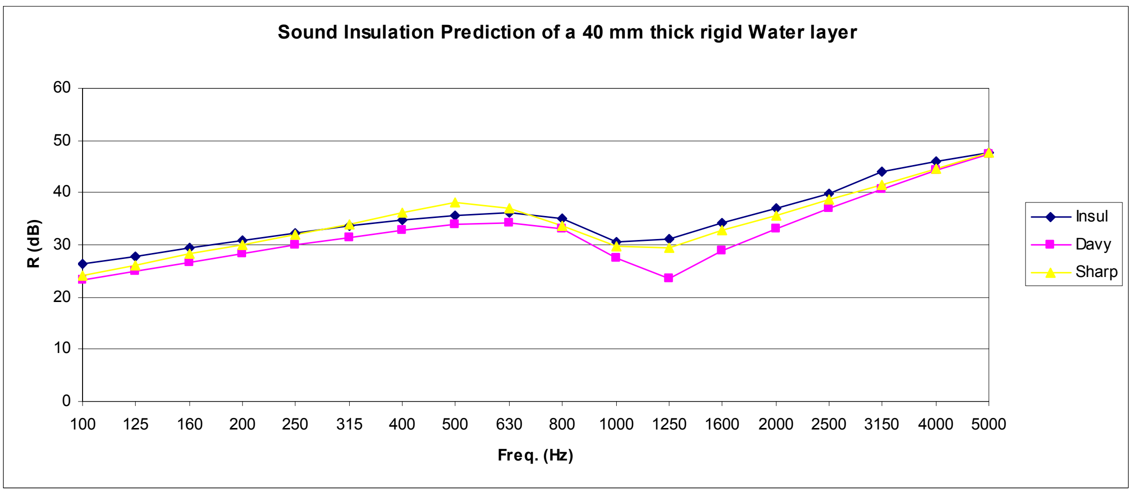

Fig. 4.3: Sound insulation prediction of a 40 mm thick rigid water layer

Calculated R_W values for the 40 mm thick rigid water layer are 36, 32 and 35 dB for Insul, Davy and Sharp models respectively. An increase of 2 to 3 dB is therefore expected when both surface mass and layer thickness are doubled.

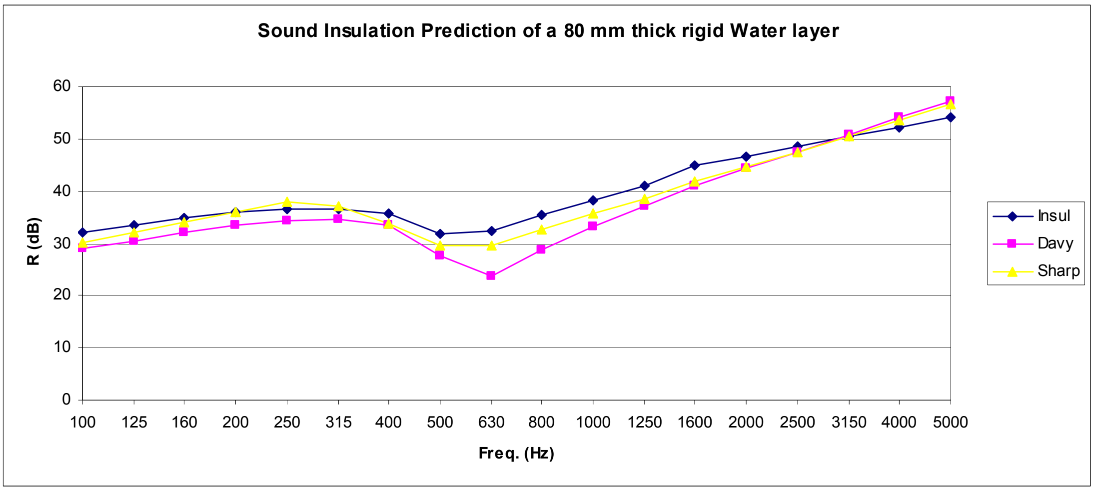

Fig. 4.4: Sound insulation prediction of an 80 mm thick rigid water layer

Calculated R_W values are now 39, 34 and 37 dB for Insul, Davy and Sharp models respectively for the 80 mm thick rigid layer. Again, the prediction models show a 2 to 3 dB increase when both surface mass and thickness are doubled. Note that the critical frequency halves when the thickness doubles (represented by the sound insulation dip in the graphs), being around 2500 Hz, then 1250 Hz and finally 630 Hz, for 20, 40 and 80 mm respectively. This phenomenon, however, is not expected to occur with the non-rigid water layers.

Table 4.4 shows the input data for the vegetable oil rigid layer sound insulation prediction models, related to each thickness. The density of the liquid was determined in a laboratory, using the test vegetable oil.

Table 4.4: Input data for an oil layer

Input Data – Oil Layer				
		Thickness		
Model	**Parameter**	**20 mm**	**40 mm**	**80 mm**
Insul	density (Kg/m^3)	986		
	fc.m	44543		
	Loss Factor (η)	0,01		
	d (m)	0,02	0,04	0,08
Davy	m (Kg/m^2)	19,72	39,44	78,88
	d (m)	0,02	0,04	0,08
	c_L (m/s)	1430		
	Loss Factor (η)	0,01		
	Length (m)	5		
	Width (m)	2		
Sharp	m (Kg/m^2)	19,72	39,44	78,88
	e (m)	0,02	0,04	0,08
	c_L (m/s)	1430		
	Loss Factor (η)	0,01		

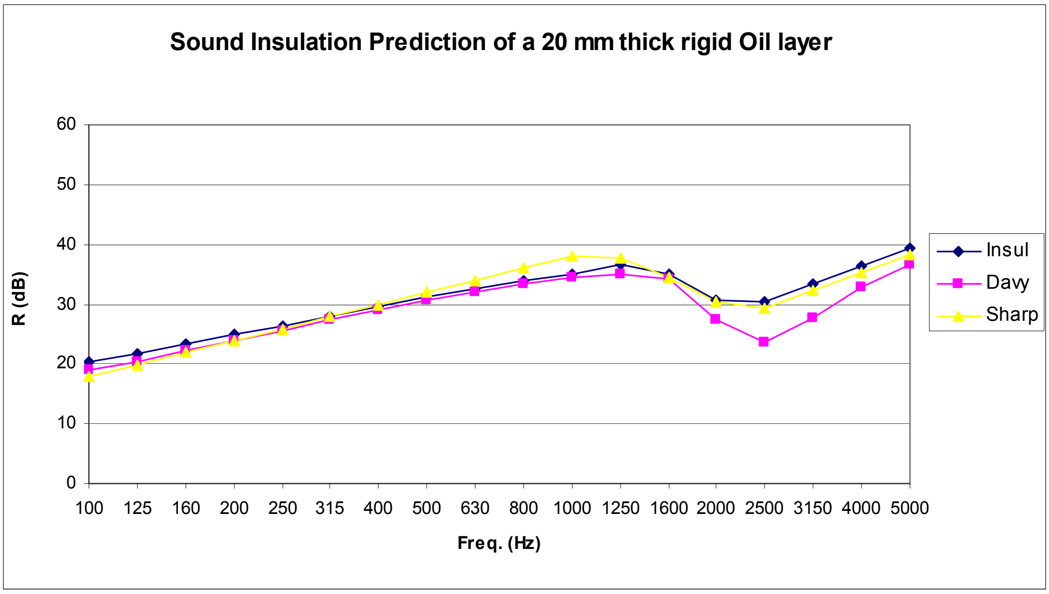

Fig. 4.5: Sound insulation prediction of a 20 mm thick oil layer

The calculated values shown in Fig. 4.4 are very similar to the 20 mm thick rigid water layer calculations in Fig. 4.2, since the input data is also similar. The critical frequency is also in the 2500 Hz third octave band, and R_W values calculated through Insul, Davy and Sharp are, respectively: 33, 31 and 33 dB, the only difference from the 20 mm water layer being the Davy model result, which is 1 dB higher with vegetable oil than with water.

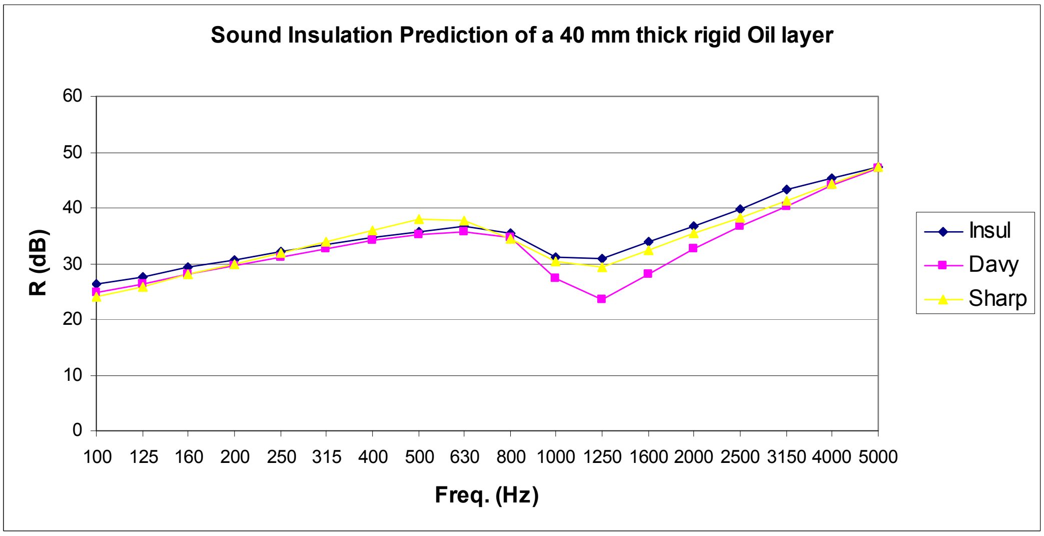

Fig. 4.6: Sound insulation prediction of a 40 mm thick oil layer

The graph above shows that when the thickness is doubled the critical frequency halves, since now the sound insulation dip is in the 1250 Hz third octave band, going down from 2500 Hz. Again, the sound insulation values are very similar to the equivalent water layer (Fig. 4.3). Calculated R_W values are 36, 32 and 35 dB for the Insul, Davy and Sharp models respectively – an increase of 1-3 dB is therefore expected in relation to the 20 mm thick rigid oil layer.

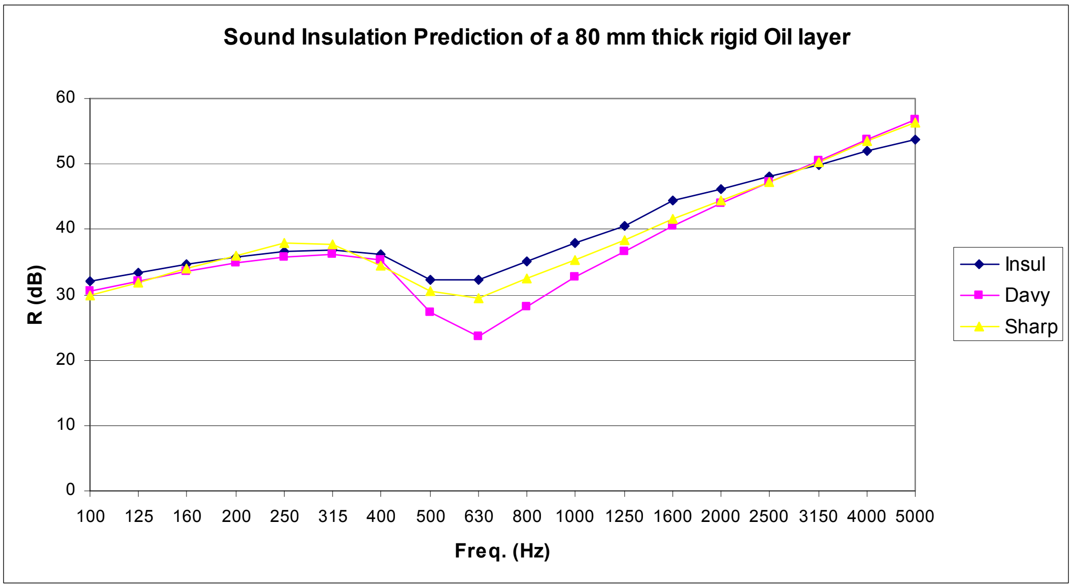

Fig. 4.7: Sound insulation prediction of an 80 mm thick oil layer

The last sound insulation calculations show a critical frequency in the 630 Hz third octave band, and calculated R_W of 39, 34 and 37 dB for the Insul, Davy and Sharp models. A 2-3 dB increase is expected from the previous solution, and values are identical to those of 80 mm thick water layer (Fig. 4.3).

4.3 TEST SAMPLE DESIGN

As said in the beginning of this work, one of the main challenges of this project is the test sample design and construction, in which a few issues must be considered:

- The sample has to hold the fluid during laboratory testing and for a reasonable amount of time, taking into account hydrostatic pressure exerted by the liquid on the walls of its container;
- It should create a thin layer of water/liquid in which both surfaces are reasonably plane and homogeneous;
- The test sample or panel that contains the liquid must be made of materials that have very high sound transmission loss around the liquid and very low transmission loss between the liquid and the source/receiving rooms, so that meaningful results can be obtained.

A first approach to this problem was to build a vertical wall with an MDF structure and two ETFE sheets melted together at the edges. This panel was built in the Eindhoven University of Technology (TU/e) acoustic laboratory in the Netherlands, but it proved very difficult to carry out the sound insulation measurements because various leaks formed on the bottom of the plastic sheet. The melting of the plastic sheets weakened these in such a way that it was impossible for them to withstand the hydrostatic pressure that formed inside the panel whilst it was being filled.

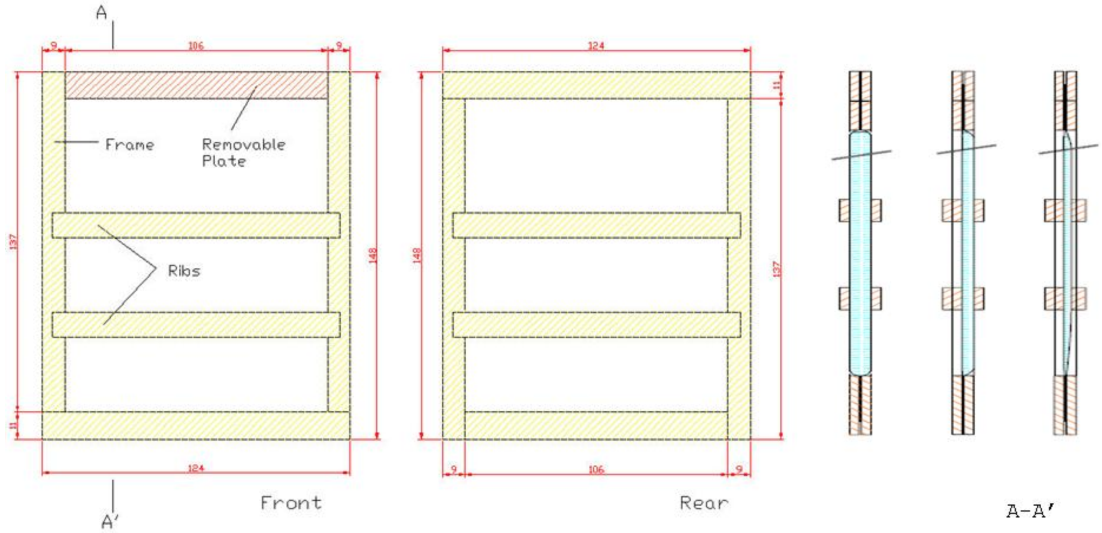

Fig. 4.8: Initial test sample design – a wall composed of two ETFE sheets melted together at the edges with an MDF frame to support them. Cross-section A-A' shows three panel configurations to test three different thicknesses: 10, 20 and 40 mm.

Fig. 4.9: Initial test sample: building the sample (left) and then filling it with water (right)

Later testing showed that in fact the hydrostatic pressure would be too high for any thin plastic sheet when using a wall (a 1,0 m water column, would produce on the bottom of the wall around 9,8 kN, which is quite a large force), since various known methods of joining plastic sheets weaken them to a point of not withstanding such forces.

Therefore it was decided that a horizontal panel would be used for testing, since the hydrostatic pressure exerted on the plastic walls would be a fraction of the one in the previous solution, controlling the thickness of the liquid layer would be much easier, as would be filling and emptying the test sample.

The tests were done inside the Acustilab, a small educational reverberant chamber of dimensions: 1,50 x 3,50 x 1,40 m^3 (see Fig.4.10). For further reading please refer to [18]. The chamber has a removable concrete lid, so the main idea was to create a new lid with an opening that would hold the test sample and then measure the sound insulation with a sound source inside the Acustilab, measuring sound transmission from the inside to the outside.

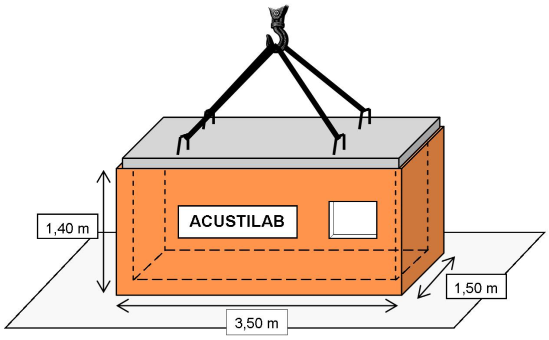

Fig. 4.10: Acustilab drawing

The region where results were predicted to lie in is depicted in the following illustration, as is the usable frequency spectrum in which the Acustilab gives meaningful results, either due to standing wave phenomena in the lower frequency range (below 315 Hz), or due to leaks in the higher frequency range (above 3150 Hz):

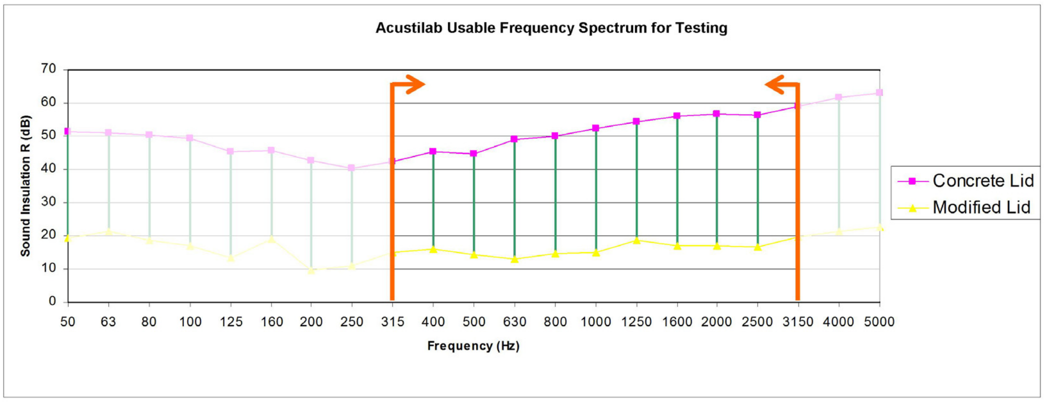

Fig. 4.11: Acustilab usable frequency spectrum for laboratory testing, in which the Acustilab gives adequate results

4.3.1 Acustilab Modified Lid

The new concrete lid is composed of a steel formwork with a 5 mm thick base plate (3,50 x 1,50 m^2) and 2 mm thick steel plates soldered to the edges to act as walls (to contain the concrete) 11 cm high. An opening was cut into the base plate, large enough to allow the test sample to cover the opening and small enough to support the frame of the test sample. Steel bars were soldered between the closer walls for added support and T-shaped metallic pieces were also fitted on the base of the frame to better hold the concrete to the steel. Four bent metallic bars were welded to the base plate so that the lid could be lifted onto the Acustilab with the laboratory winch. These were placed in such a way that the winch would lift the lid above its centre of gravity, and therefore not tilt when lifted. A more detailed 3D model of the Acustilab modified lid and detailed drawings can be found in Annex A.

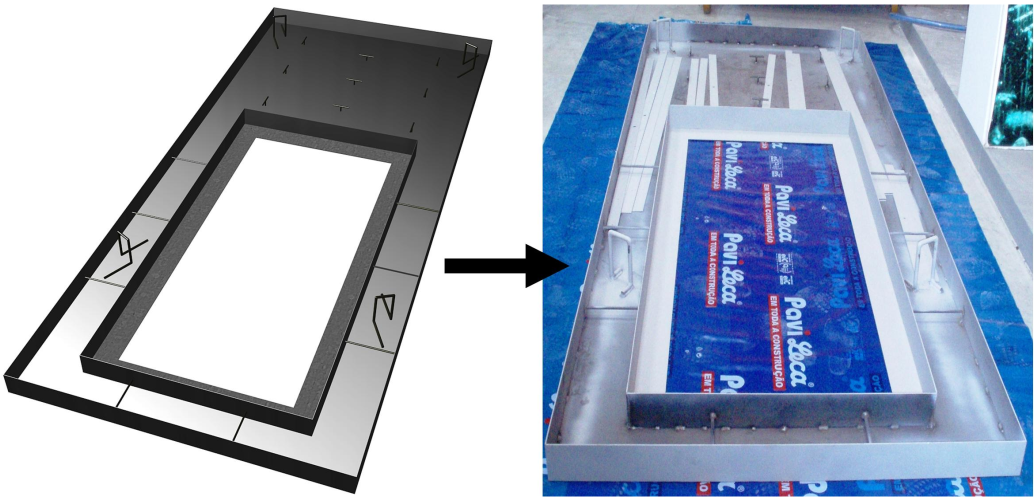

Fig. 4.12: Acustilab lid steel formwork – 3D model (left) to finished product (right)

After building the steel frame, it was filled with steel fibre-reinforced concrete, as shown by Fig. 4.13 and Fig. 4.14.

Fig. 4.13: Making fibre-reinforced concrete

Fig. 4.14: Applying the concrete on the steel formwork

The concrete was left to set for one week, and then a resilient layer of foam (Aglomex^{TM} 120 kg/m^3 foam, 10 mm thick) was applied on the base of the opening in order to acoustically separate the Acustilab lid from the test sample, as shown in the following pictures.

Fig. 4.15: Fitting the base plate opening with the resilient layer

The final result is the following:

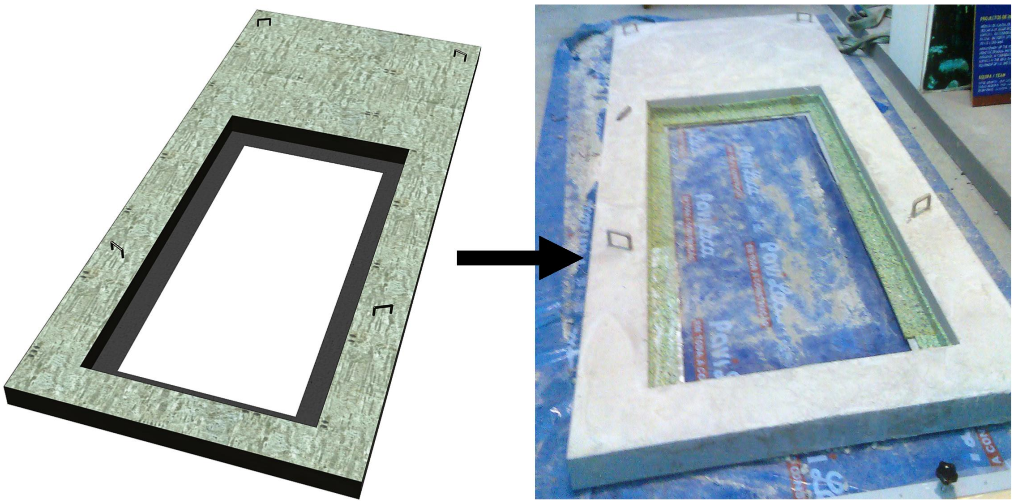

Fig. 4.16: Acustilab modified lid – 3D model (left) to final result (right)

4.3.2 Test Sample

The test sample in itself is basically a rectangular wooden frame (1,0 x 2,0 m^2) composed of four wooden bars (cross section 80 mm x 100 mm), with a 2 mm thick steel grill with square holes (20 x 20 mm) and a thin LDPE (low density polyethylene) sheet placed on top (0,15 mm thick). This plastic sheet is used to hold the liquid, while both the steel grill and wooden frame are used for support. The wooden frame is fitted with steel plates (2 mm thick) that are screwed into the wood to connect the steel grill to the wooden frame on the bottom, and to hold the plastic sheet to the frame on top. Four levelling screws are fitted in the centre of each side of the frame so that the test sample can be easily levelled if necessary. The wooden frame is marked on its inner facing with the various heights required for testing (20, 40 and 80 mm high from the base of the sheet).

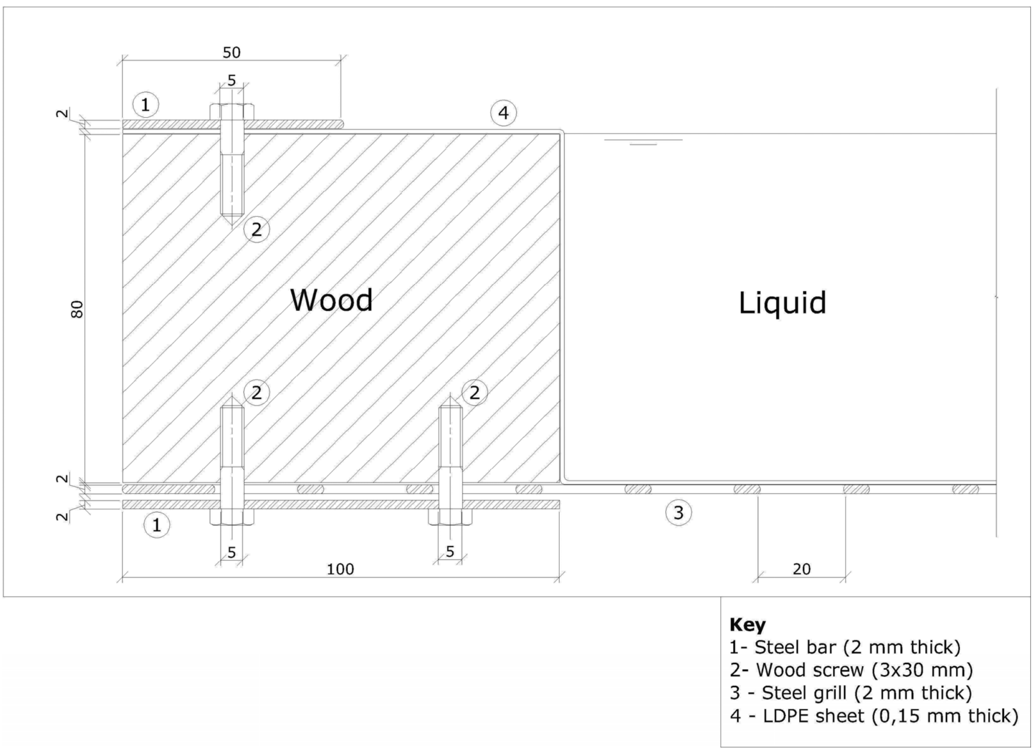

Fig. 4.17: Detailed drawing of the test sample

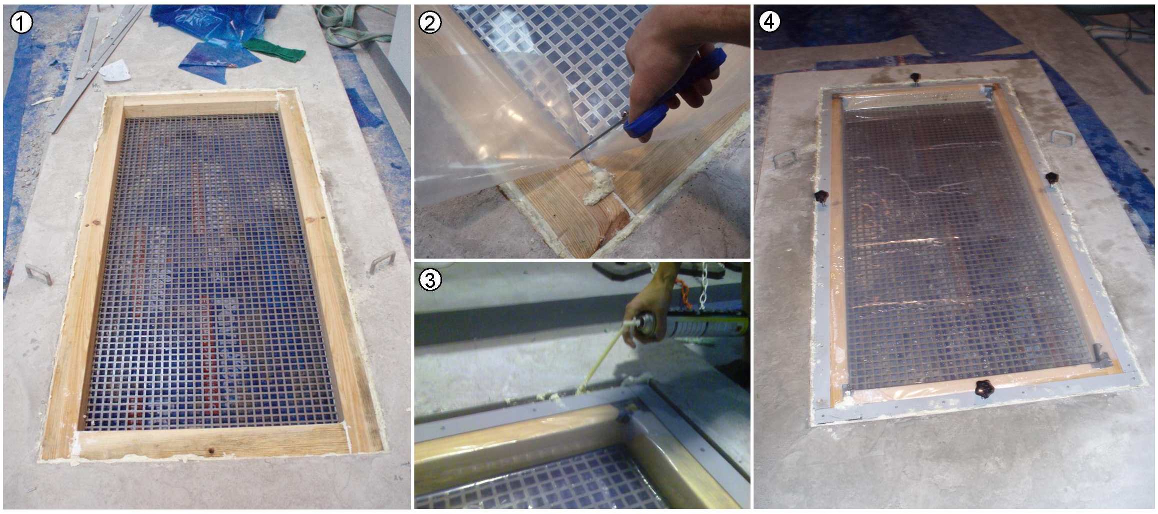

Fig. 4.18: Fitting the test sample into the Acustilab lid

Fig. 4.18 shows how the test sample was fitted into the Acustilab modified lid. First the wooden structure with the steel grill was placed in the opening (1) and then the LDPE sheet was fitted onto the test sample frame (2). Polyurethane foam was sprayed into the gap between the test sample and Acustilab lid opening (3) to avoid sound transmission leaks, the levelling screws were fitted in to adjust the test sample if necessary, and finally a small amount of water was poured into the sample to test for leaks (4).

4.4 Laboratory Measurements and Procedures

4.4.1 Laboratory Equipment

The following equipment was used to carry out the test measurements:

- Three Brüel & Kjaer ½" 4189 microphones with stands;
- Brüel & Kjaer Type 4224 sound source;
- Brüel & Kjaer Type 4226 Multifunction Acoustic Calibrator;
- Brüel & Kjaer PULSE data acquisition system with Type 7700 CPB Analysis software

4.4.2 Test Method

4.4.2.1 Calibration and Mounting

The three available microphones captured sound pressure levels that were then recorded using the PULSE system linked to them. Two microphones were placed inside the Acustilab and one other was placed 0,70 m above the lid, in three different positions for each test. The test was set up as shown in Fig. 4.19.

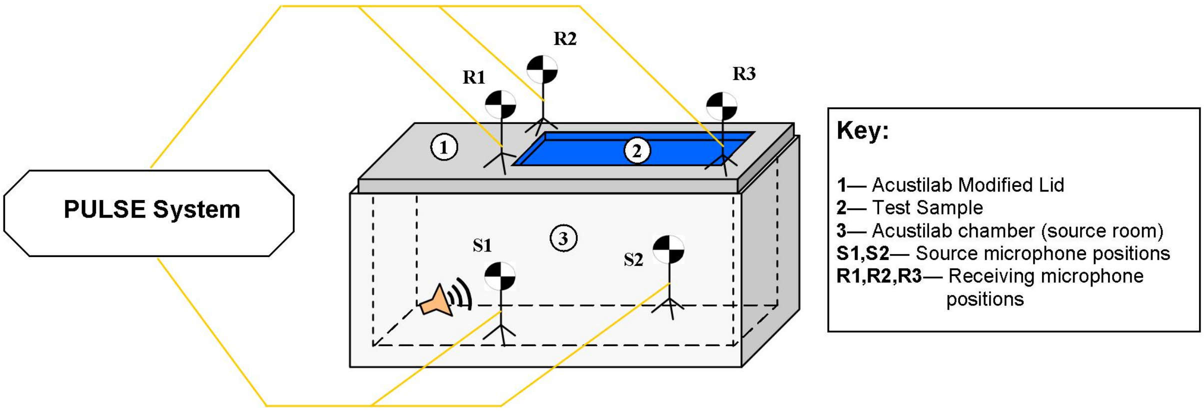

Fig. 4.19: Test setup with the Acustilab modified lid

The three microphones were calibrated using the Brüel & Kjaer Multifunction Acoustic Calibrator (B&K Type 4226), and the calibration itself was done through the PULSE system.

The microphones were carefully positioned inside the Acustilab chamber order to get the most reliable results, since standing waves may form for lower frequencies and therefore influence the measurements. Various measurements were taken inside the lid until satisfactory results were recorded, having assumed that a relatively small dispersion of results between the microphones would provide more reliable data. After these microphone positions were found, actual testing could then begin.

4.4.2.2 MEASUREMENT OF BACKGROUND NOISE

The background noise was measured in the laboratory in various positions so that corrections could be made for the measurements taken outside the Acustilab. Laboratory tests were done at night, when the background noise is lowest so as to not influence the test results. The laboratory lights were also turned off since they were also a significant noise source.

The reverberation time of the laboratory facility was measured and it was concluded that the volume was too large to consider the receiving room as a reverberant space. Therefore only the direct field around the panel was measured, and the correction for reverberant spaces was not considered.

4.4.2.3 MEASUREMENT OF SOUND PRESSURE LEVEL DIFFERENCE

Various laboratory measurements were done, using two Acustilab lids: both are made of concrete and have the dimensions 1,40 x 3,50 x 0,11 m^3. The first is the Acustilab solid concrete lid with no opening and a steel base, and the second is the Acustilab modified lid fitted with the test sample, both of which are described earlier on in this work.

With the Acustilab modified lid, seven tests were done:

1. Empty test sample fitted in the lid;
2. Test sample filled with 20 mm of water;
3. Test sample filled with 40 mm of water;
4. Test sample filled with 80 mm of water;
5. Test sample filled with 20 mm of vegetable oil;
6. Test sample filled with 40 mm of vegetable oil;
7. Test sample filled with 80 mm of vegetable oil.

The sound pressure level differences were recorded for each of the microphone positions both inside and out of the Acustilab chamber. The empty test sample was fitted with both the steel grill and the same LDPE sheet during all the laboratory tests. A small leak was detected and promptly dealt with using adhesive tape.

The test sample was filled with water using a hose until the relevant markings on the wood frame were reached (the different heights were marked on the inside of the wooden frame). After the tests were completed the water was removed with a hose, by taking advantage of the height difference between the test sample and the drainage system to flush it out. The LDPE sheet was subsequently dried so that the next liquid could be used.

Fig. 4.20: Filling the test sample with water

The vegetable oil was directly poured into the test sample from its containers until the relevant markings in the wooden frame were reached. The same method was used as the water tests, only this time the vegetable oil was collected in other containers to be properly disposed of.

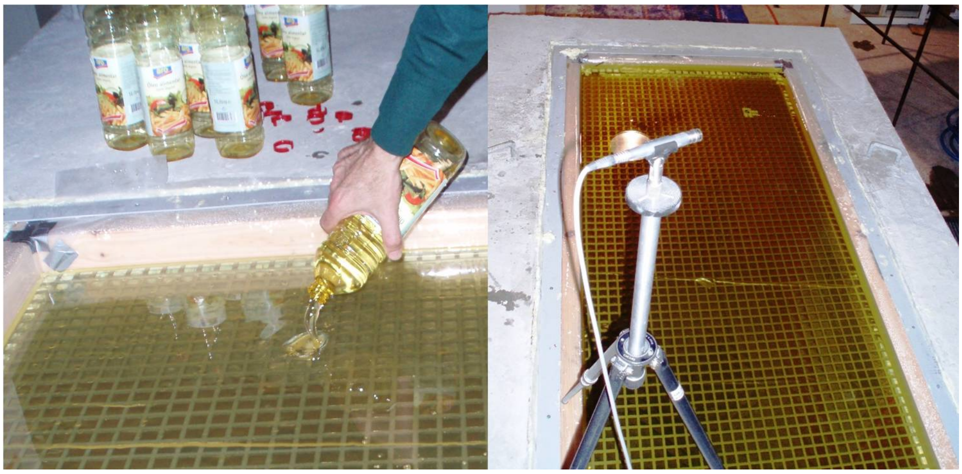

Fig. 4.21: Filling the test sample (left) and then testing the vegetable oil (right)

5
Results and Discussion

5.1 Introduction

The following Chapter presents the laboratory test results and their analysis. Various combinations are tested as are different thicknesses of liquid sheets, in order to get some insight of the behaviour of liquids in terms of sound insulation. As shall be seen in this Chapter, the frequency range which gives adequate results is 315 to 3150 Hz, confirming the illustration in Fig. 4.10.

5.2 Results

5.2.1 Validation of the Test Method (using the conrete lid)

Looking at the results with more detail, the first thing that was done was to compare the concrete lid with prediction models, as shown in Fig. 5.2. This was done in order to help validate the test method. The input data used for the prediction models was that of the known properties of concrete such as density, thickness and total loss factor.

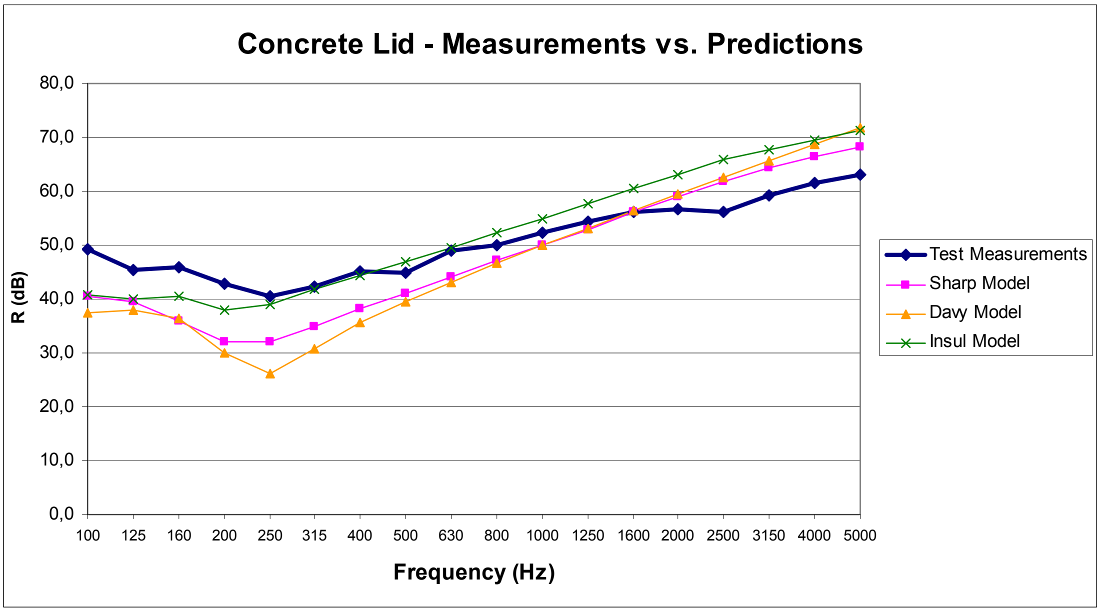

Fig. 5.1: Sound Insulation of the Acustilab concrete lid – comparing test measurements with various prediction models

Calculation of correlation coefficients gives very good correlation between the test results and the prediction models – R^2 value of 0,96 for the Insul model, 0,98 for Sharp and 0,99 for the Davy model. The test measurements give a weighted sound reduction index R_w of 51 dB, which the Insul model also predicted. The Sharp and Davy models calculated smaller values – 45 and 43 dB respectively. The critical frequency is also very similar in all of the results (around the 250 Hz third octave frequency band). From these results one can admit that the test method gives adequate results for analysis, although with some limitations that have been already discussed.

5.2.2 WATER-FILLED PANEL

The following graph shows the test results for the Acustilab modified lid (fitted with the empty test sample) and the same lid and test sample filled with 20 mm of water. The relevant prediction models are also included for comparison.

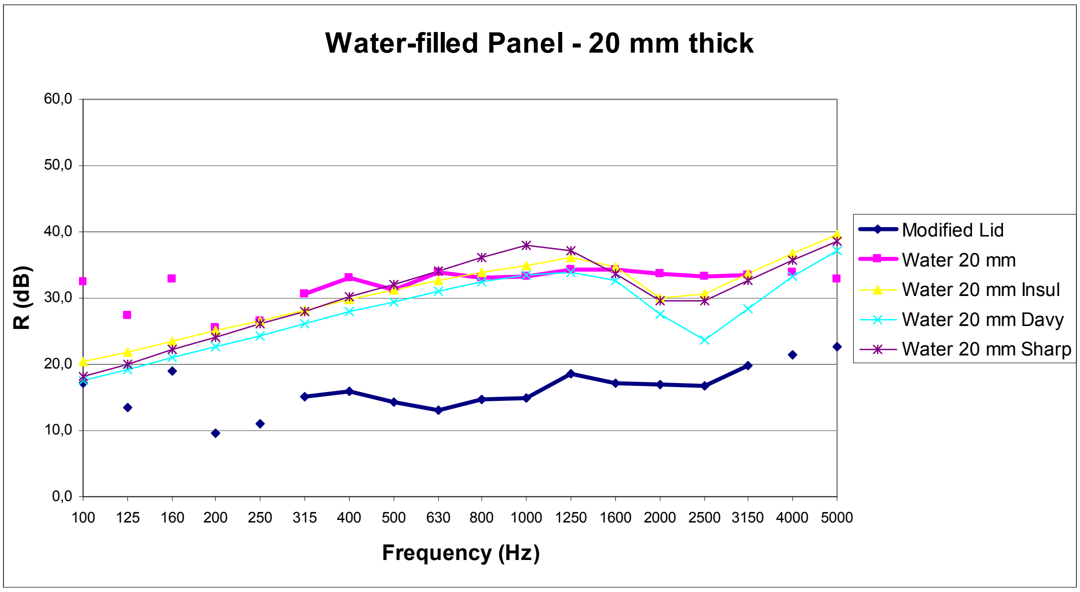

Fig. 5.2: Test results for the Acustilab modified lid and the same lid filled with 20 mm of water, together with the relevant prediction model calculations

There is quite a significant gain when the water is added to the empty test sample, in which the sound insulation increases by a more or less constant value; from 315 Hz to 3150 Hz, the proposed frequency range of analysis, there is an average gain in R of 17 dB in each third octave frequency band. From 630 Hz onwards the R values smoothen out and remain more or less constant, around 33 dB (± 1 dB). The Acustilab modified lid has an R_W of 16 dB, and when filled with 20 mm of water the sound reduction index increases to 34 dB – an 18 dB difference, which is quite significant.

There is a very weak correlation between the prediction models and the test results, for which the R^2 values for the Insul, Davy and Sharp models in relation to the test results are 0,47 , 0,30 and 0,34 respectively. The input data and relevant calculations are referred to in Chapter 4.2.2. An interesting aspect inferred by the test measurements is that the critical frequency phenomenon doesn't occur where expected (in the 2500 Hz third octave band), which confirms the expected behaviour of the liquid layer.

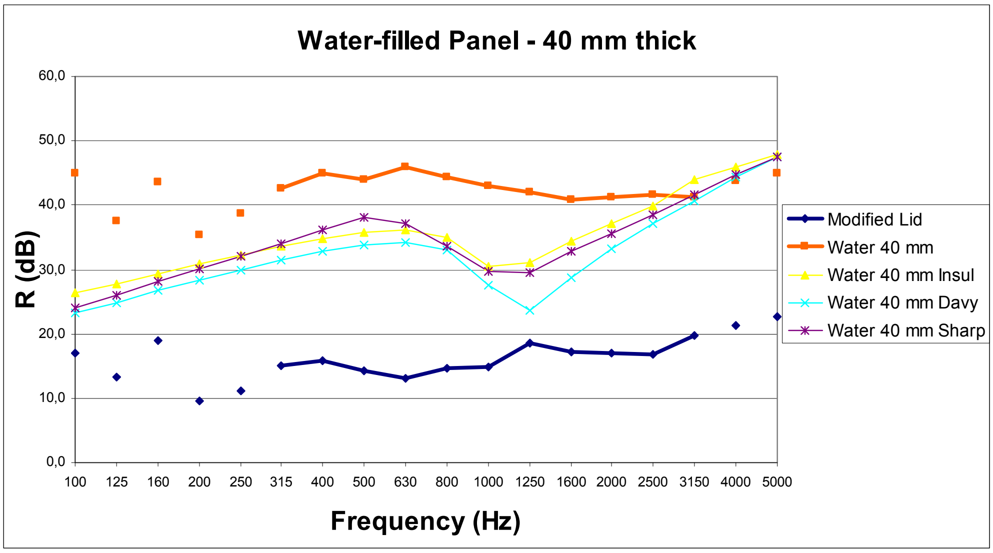

Fig. 5.3: Test results for the Acustilab modified lid and the same lid filled with 40 mm of water, together with prediction model calculations

The graph presented above (Fig. 5.3) now shows the gain in sound insulation when the test sample is filled with 40 mm of water. The R values in the analysed frequency range (315-3150 Hz) show a relative drop from 1000 Hz to 1600 Hz and then maintain relatively constant until 3150 Hz. The average gain from the empty test sample is 27 dB in each third octave band. The maximum R value is at 630 Hz which is 46,0 dB and the lowest is 41,3 dB at 1600 Hz and 3150 Hz. The sound reduction index (R_W) of the test sample with 40 mm of water is 43 dB – an increase of 27 dB from the empty test sample. The prediction model calculations give very different results than the test measurements, as there is no coincidence frequency phenomenon where would be expected for a rigid panel of identical characteristics. Correlation coefficient between the measurements and the Insul, Davy and Sharp models are: -0,23 , 0,05 and 0,05 respectively – no correlation or a weak negative correlation. The input data and relevant results for the calculations are specified in Chapter 4.4.2.

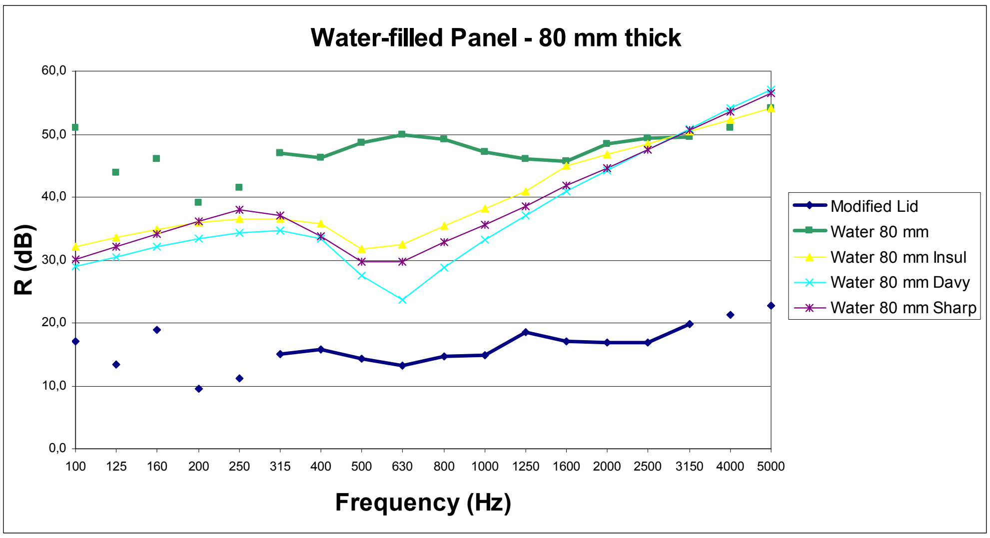

Fig. 5.4: Test results for the Acustilab modified lid and the same lid filled with 40 mm of water, with prediction model calculations

Fig. 5.4 shows a comparison between the Acustilab modified lid fitted with the empty test sample and the same lid filled with 80 mm of water, together with the prediction model calculations for a rigid panel of identical characteristics to water (same density, surface mass, thickness, etc.). The sound insulation values for the water-filled test sample in the analysed frequency range are just under 50,0 dB, although there is a dip between 1000 Hz and 2000 Hz, in which the lowest value is 45,7 dB. R_W for the 80 mm water panel is 48 dB – an increase in 32 dB from the empty test sample. The test measurements show a very clear difference from the prediction models, as all the sound insulation values are 10,0-20,0 dB higher than the latter. Just as with the previous tests, there is no coincidence frequency phenomenon – in fact, the sound insulation increases instead of dropping in the frequency band indicated by the prediction models (630 Hz third octave band), which is quite interesting. Correlation R^2 values between the Insul, Davy and Sharp models compared to the test measurements are respectively: -0,14 , -0,31 and -0,10 – a very weak (and negative) correlation.

All the water-related tests are compared in the following graph:

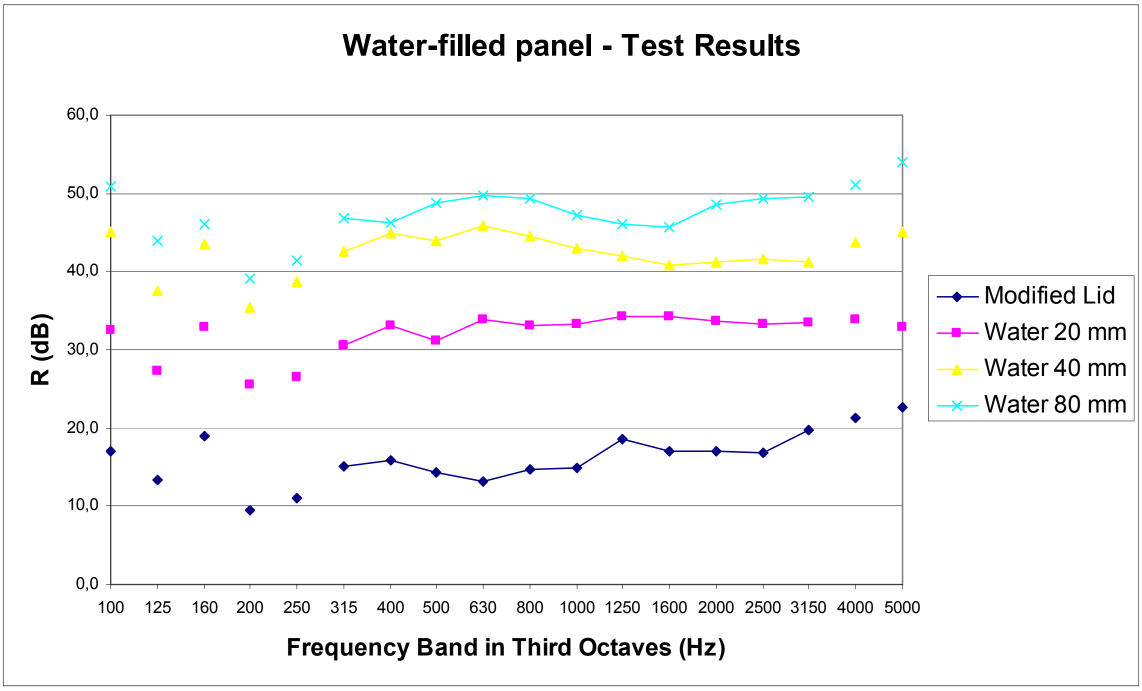

Fig. 5.5: Test results for the water-filled panel, with different thicknesses of water: 20, 40 and 80 mm

This final graph shows quite well how the water-filled panel has a relatively linear response in the analysed frequency range (315-3150 Hz) and that there is an increase in sound insulation of 9 dB in R_W when the thickness is doubled from 20 mm to 40 mm, and then there is an increase of 5 dB in R_W from 40 mm to 80 mm.

Note that R_W values presented aren't error-free, since the ISO 717-1 weighting includes frequency bands (100-5000 Hz) that are outside of the frequency range that gives reliable results in the Acustilab. In any case these values aren't omitted due to the fact that these errors won't change the final values in a very significant way and the results are comparable between themselves (the errors affect all the test results in the same way).

5.2.3 Panel Filled with Vegetable Oil

Now the vegetable oil test results are presented and analysed. The first graph shows both the Acustilab modified lid fitted with the empty test sample and the test sample filled with 20 mm of vegetable oil. The analysed frequency range maintains the same: 315 Hz to 3150 Hz.

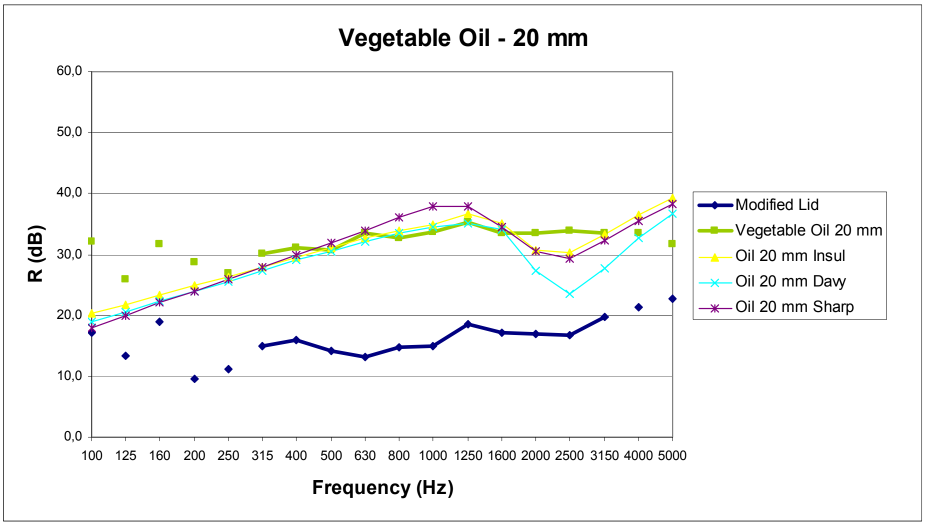

Fig. 5.6: Test results for the Acustilab modified lid and the same lid filled with 20 mm of vegetable oil, with prediction model calculations

Just like the test result with water, the vegetable oil presents constant R values from 630 Hz onwards (around 33,5 dB), will a small peak at 1250 Hz of 35,3 dB. From 315 to 630 Hz there is a small increase, from 30,2 dB to 33,6 dB respectively. R_W for the 20 mm oil-filled panel is 33 dB – 1 dB less than the equivalent test with water. The critical frequency phenomenon doesn't occur in the predicted third octave frequency band, since the liquid layer is non-rigid. The correlation coefficient was calculated in order to evaluate the correlation between the test measurements and the prediction model curves: R^2 for Insul, Davy and Sharp are, respectively, 0,75 , 0,31 and 0,61.

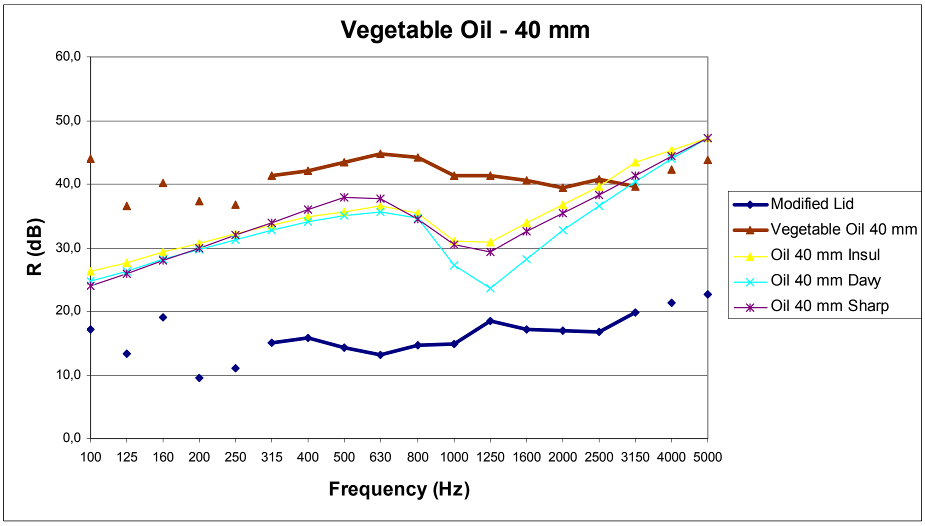

Fig. 5.7: Test results for the Acustilab modified lid and the same lid filled with 40 mm of vegetable oil, with prediction model calculations

After having doubled the thickness of the liquid layer from 20 to 40 mm, a very significant increase in sound insulation can be seen, and again the R values have small variations – in the analysed frequency spectrum, 315-3150 Hz, values vary from 40 to 45 dB. From 315 to 630 Hz the sound insulation increases steadily from 41,4 to 44,8 , only to decrease again until the 1000 Hz third octave band, and then varies only slightly (around 1 dB) until 3150 Hz, around 40,0 dB. R_W for the 40 mm oil-filled panel is 42 dB – also 1 dB less than the equivalent water-filled panel. The coincidence effect doesn't appear, and correlation values for the Insul, Davy and Sharp models in relation to the test measurements are: -0,23 , 0,11 and 0,02 , respectively, which show practically no relation between them.

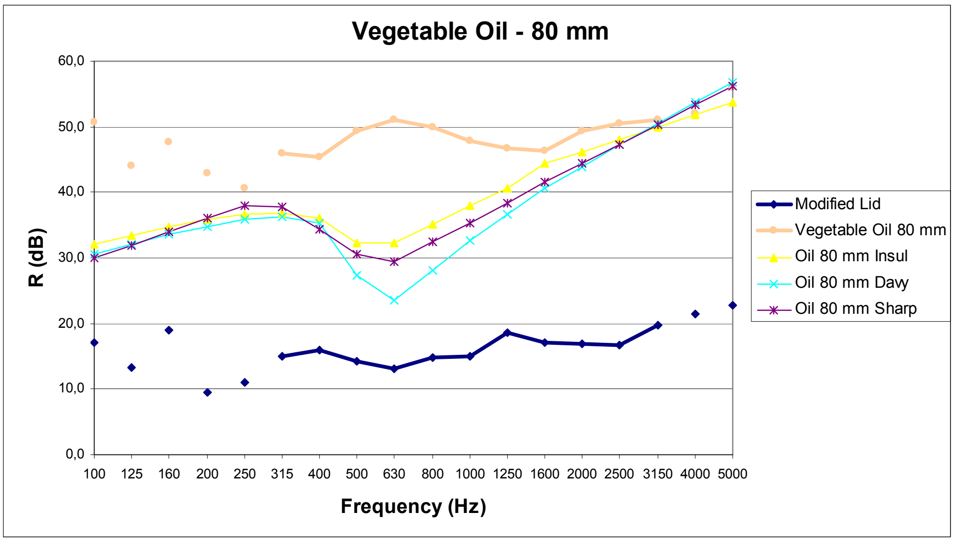

Fig. 5.8: Test results for the Acustilab modified lid and the same lid filled with 80 mm of vegetable oil, with prediction model calculations

Results for the test sample filled with 80 mm of vegetable oil show an increase and then decrease of R values, from 45,9 dB at 315 Hz to 51,1 dB at 630 Hz, then 46,3 at 1600 Hz and increases steadily until 3150 Hz at 51,0 dB. There is a very significant gain for the vegetable oil in relation to the empty test sample – R_w increases from 16 dB to 49 dB (ΔR = 33 dB). The sound reduction index for the vegetable oil test is 1 dB higher than the equivalent water test. Correlation coefficients for the Insul, Davy and Sharp models: 0,17 , 0,04 , 0,16 respectively – insufficient for adequate predictions. Another interesting observation: in this case the sound insulation actually increases where it was ‘supposed’ to drop (according to the prediction models).

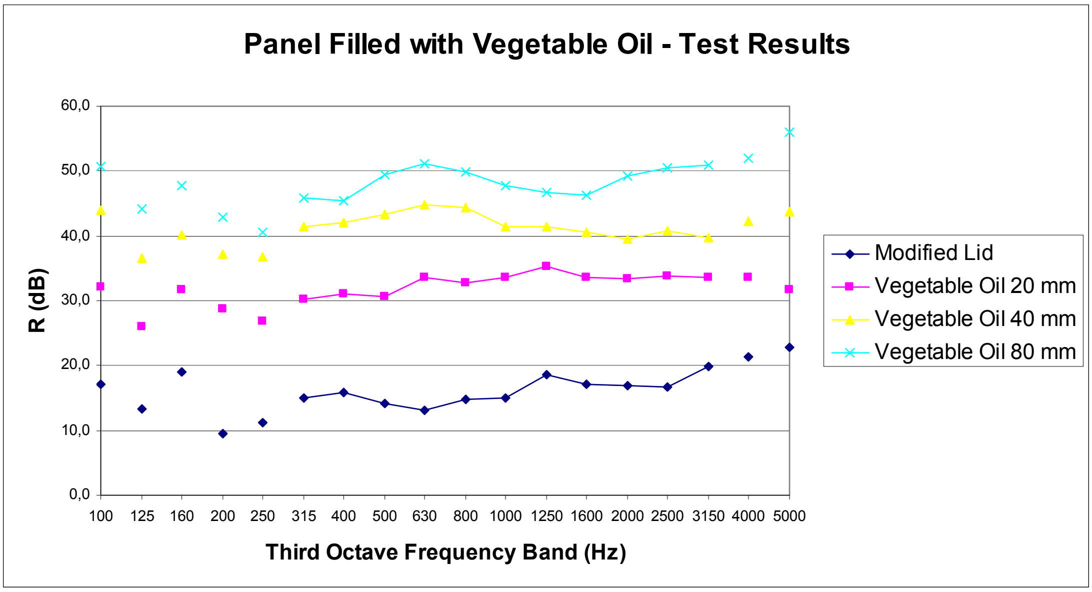

Fig. 5.9: Test results for the panel filled with vegetable oil, with different amounts of vegetable oil: 20, 40 and 80 mm

Just as with the water, the vegetable oil has a more or less linear response in the analysed frequency range, and the liquid also shows a definite increase in R values when the thickness is doubled. This increase in R_W is 9 dB from 20 to 40 mm, and 7 dB from 40 to 80 mm.

5.2.4 Water vs. Vegetable Oil

The graph below shows all of the results for the liquids and the empty test sample:

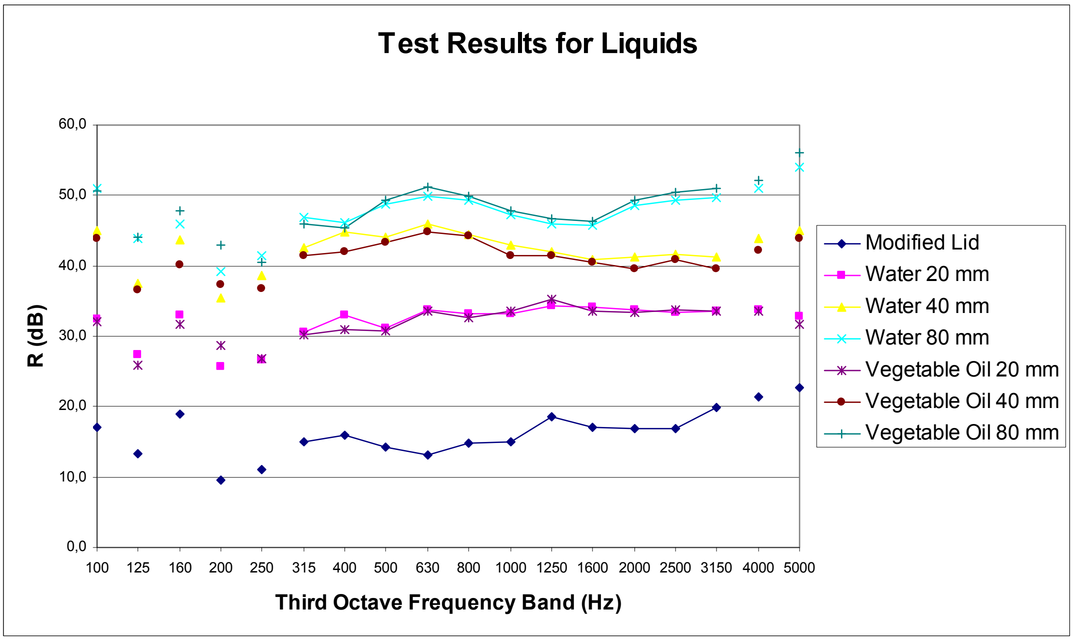

Fig. 5.10: Test results for the test sample filled with vegetable oil and the test sample filled with water

As can be seen above, the results for sound insulation are almost identical, with small variations in one or two frequency bands.

5.3 Final Results

All the test results are represented in the following graph:

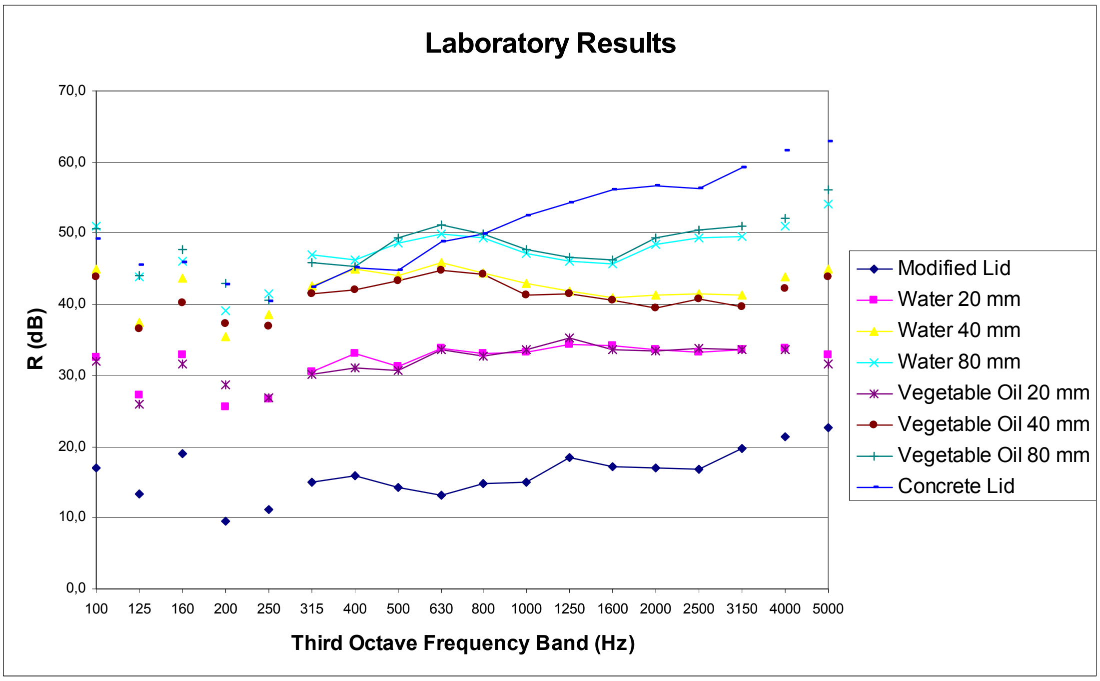

Fig. 5.11: Complete test results – Sound insulation values at third-octave frequency bands

5.4 DISCUSSION

A test method has been proposed, as a first (to the author's knowledge) approach to the evaluation of sound insulation of liquid-filled panels. It is somewhat limited due to the relatively small frequency range that gives adequate data, although a small window of light has been shed onto these materials, which in fact may encourage more testing.

The first observation is that both water and the vegetable oil perform in an identical way. By comparing the physical properties of each material in Tables 4.3 and 4.4, the two parameters that affect the air-liquid interface (density ρ and sound propagation speed c) are also very similar. Another important aspect is that the gains in sound reduction index R_W when the thickness doubles seem to follow a logarithmic law.

The first aspect helps to validate the use of Snell's law, of which the density and sound propagation speed are the main variables. The second observation leads an attempt to describe the gains in R_W. Therefore, the following expression is proposed:

$$R_{W,liq} \approx \overline{R_{Snell}} + 20\log h = \frac{1}{n}\cdot\left[\sum_{j=1}^{n} 10\log\left(\frac{1}{a_{t,j}}\right)\right] + 20\log h \qquad (5.4.1)$$

Where:

$R_{W.liq}$ – weighted sound reduction index prediction for a liquid, in dB;
$\overline{R_{Snell}}$ – average sound insulation due to the air-liquid interface, according to Snell's law, in dB (See Chapter 3.5);
n – number of discrete angles until the critical angle;
a_t – sound power transmission coefficient (Note: if the angle of incidence is greater than the critical angle, $a_t = 0$);
h – thickness of the liquid layer, in centimetres (cm).

And:

$$a_t = \frac{4Z_0 Z_L \cos\theta_i \cos\theta_t}{\left(Z_0\cos\theta_t + Z_L\cos\theta_i\right)^2} \qquad (5.4.2)$$

Where:

a_t – sound power transmission coefficient (Note: if the angle of incidence is greater than the critical angle, $a_t = 0$);
Z_0 – specific acoustic impedance of air (in rayl or Pa.s/m), where $Z = \rho c$ and $Z_0 \approx 400$ rayl;
Z_L – specific acoustic impedance of the receiving liquid (in rayl or Pa.s/m);
θ_i – angle of incidence of the sound wave on the interface;
θ_t – angle of transmission of the sound wave from the interface.

The following table was built in order to ascertain the accuracy of the proposed expression:

Table 5.1: Testing $R_{W,liq}$ by correlation with the available test results

	R_W (dB)	Average R_{Snell} (dB)	R_W - R_{Snell} (dB)	20*log(h) (dB)	R_{Snell} + 20*log(h) (dB)	R^2
Water 20 mm	34	30,9	3,1	6,0	37	0,987
Water 40 mm	43		12,1	12,0	43	
Water 80 mm	48		17,1	18,1	49	
Vegetable Oil 20 mm	33	30,5	2,5	6,0	37	0,997
Vegetable Oil 40 mm	42		11,5	12,0	43	
Vegetable Oil 80 mm	49		18,5	18,1	49	

The first component of (5.4.1) is the mean value of sound insulation calculated by using all the whole number angles of incidence that give a valid sound insulation value, according to Snell's law (See Chapter 3.5 and Expression (3.5.9)) including the critical angle of incidence. The second component of the formula is the correction that takes into account the sound insulation gain induced by the thickness of the panel (6 dB per doubling of thickness).

It cannot be stressed enough that this expression is completely preliminary and surely not error-free. The correlation coefficient R^2 is 0,987 between the calculated R_w from the test results and $R_{W,liq}$ for water, and 0,997 for vegetable oil, but the latter expression is based on six tests and so must be tested not only with different liquids but also on a broader frequency range.

6
Conclusions and Future Projects

6.1 CONCLUSIONS

After having analysed the test results, a few characteristics of the tested liquids become quite clear:

In the analysed frequency spectrum (315-3150 Hz) the sound insulation curve for the tested liquids is almost linear, a very interesting characteristic that may suggest high sound insulation for low frequencies which are quite problematic in construction;

The coincidence effect doesn't appear in liquids around the expected frequency band for a rigid panel, and might not appear at all, confirming that liquid-filled panels don't function in the same way as rigid panels;

The existing sound insulation prediction models are not adequate to describe the behaviour of liquid-filled panels, so new tests need to be done with different facilities so that broader frequency range can be analysed and new prediction models produced (or adapted from existing underwater acoustics models);

There is an almost fixed value increase in sound insulation when the layer thickness is doubled, which is about 6 dB per doubling of thickness.

There is a lot of potential in liquids as sound insulators, although further testing must be carried out. In fact, this was the main driving force of this work – to ascertain the potential of liquids, namely water, as sound insulation materials to be used in construction or other relevant applications.

Using both the information taken from these observations and by applying Snell's law for sound transmission between fluids, the following expression to obtain a prediction for the weighted sound reduction index R_W has been developed:

$$R_{W,liq} \approx \overline{R_{Snell}} + 20\log h = \frac{1}{n}\cdot\left[\sum_{j=1}^{n} 10\log\left(\frac{1}{a_{t,j}}\right)\right] + 20\log h \qquad (6.1.1a)$$

Where:

$R_{W.liq}$ – weighted sound reduction index prediction for a liquid, in dB;
R_{Snell} – sound insulation due to the air-liquid interface, according to Snell's law, in dB (See Chapter 3.5);
n – number of discrete angles until the critical angle;
a_t – sound power transmission coefficient (Note: if the angle of incidence is greater than the critical angle, $a_t = 0$);
h – thickness of the liquid layer, in centimetres (cm).

And:

$$a_t = \frac{4Z_0 Z_L \cos\theta_i \cos\theta_t}{\left(Z_0 \cos\theta_t + Z_L \cos\theta_i\right)^2} \tag{6.1.1b}$$

Where:

a_t – sound power transmission coefficient (Note: if the angle of incidence is greater than the critical angle, $a_t = 0$);
Z_0 – specific acoustic impedance of air (in rayl or Pa.s/m), where $Z = \rho c$ and $Z_0 \approx 400$ rayl;
Z_L – specific acoustic impedance of the receiving liquid (in rayl or Pa.s/m);
θ_i – angle of incidence of the sound wave on the interface;
θ_t – angle of transmission of the sound wave from the interface.

6.2 FUTURE PROJECTS

Since the means available weren't enough to do extensive testing, more tests in larger reverberant chambers should be carried out so that a larger frequency spectrum may be properly analysed. More liquids should be used, with different densities and sound propagation velocities, so that the importance of these parameters may be established. Also, more thicknesses should be included – 10 mm, 160 mm and 320 mm, to evaluate the gains in sound insulation with the variation of thickness. This means that a larger version of the test sample should be constructed, with the appropriate adaptations.

The proposed formula should also be tested as to its accuracy and corrected (or rejected) if necessary.

A structure-borne sound insulation method should also be proposed, by using the referred 'flexible containers', for example, since these may later be used as acoustic 'cushions'.

6.3 POSSIBLE APPLICATIONS

Having confirmed that in fact liquids are adequate sound insulators, another challenge presents itself: to find applications for these in the real world. In fact, the possibilities are endless, and one only requires some imagination and ingenuity to be able to apply liquids to solve various different problems.

One use for liquids as sound insulators would be to apply them in double walls, although a relatively elaborate system would have to be built in order to prevent leaks into the rest of the construction and other potential problems could arise due to thermal and ventilation issues. The liquid in the wall gap could be rainwater reused for sound insulation. The gap could instead have an enclosed container that would serve as a water reservoir for the household utilities, for example.

An important study that could be done is the sound insulation of aquariums, when introduced in a wall, for example, or a study on the sound transmission from outside to the inside of a public aquarium, since it is known that fish are sensitive to exterior vibrations caused by sound and therefore special care should be taken as to not disturb them.

A simple solution could be to replace traditional instrument practice booths that have very thick walls (up to 40 cm) to provide high sound insulation especially at low frequencies – a box-in-box solution, where a water-proof box would be inside another larger box (made of resistant plastic), and the gap between these would be filled with water (or another liquid), could probably reduce the wall thickness and increase portability, since the box wouldn't need to be full all the time, and the materials it was made of would be lightweight. This principle could also be applied to other uses, such as simultaneous translation booths in conference halls.

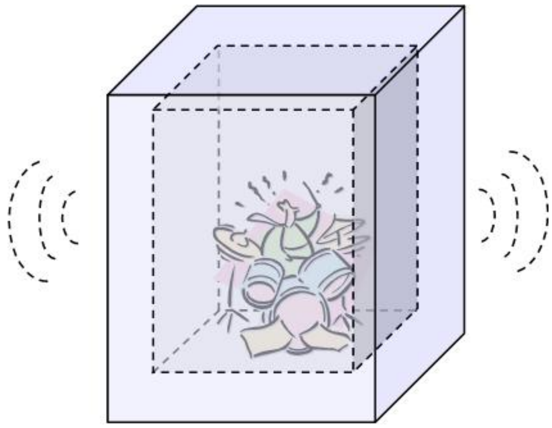

Fig. 6.1: A possible box-in-box solution with a liquid-filled gap to insulate an instrument practice booth

Liquids could also be used in construction as 'resilient layers' on floating floors, for example, where a resistant and flexible container, functioning as a 'bag', filled with liquid could greatly reduce structure-borne and airborne sound being transmitted from one room to another. These 'bags' could have other uses in reduction of airborne and structure borne sound. This solution could be applied in machine noise control, for example.

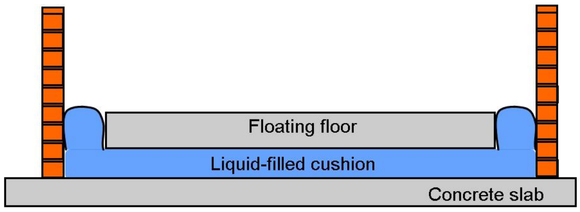

Fig. 6.2: A liquid-filled cushion acting as a resilient layer for a floating floor

Machines could be surrounded by water-filled enclosures – a simple example is a vacuum cleaner with a water enclosure around its engine, or a box-in-box solution for industrial machines could be proposed.

Many more solutions may be found and many more uses put to the test, but before they are actually implemented more testing is needed for us to fully understand the advantages and shortcomings of using liquids for these purposes.

It will all come down to the ingenuity of designers and engineers to find the best applications that may efficiently and cost-effectively unlock the potential of liquids as airborne (or possibly structure borne) sound insulators.

7
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8
Annex A – Diagrams and Drawings

8.1 DIAGRAMS AND DRAWINGS

Single Panel Sound Reduction Index R Prediction Model (Sharp, 1973)

Input:

Surface mass - m (Kg/m^2)

Panel thickness - h (m)

Material longitudinal speed - c_L (m/s)

Loss factor - η

Calculate:

Critical Frequency:

$$f_c = \frac{c^2}{1{,}8 \cdot c_L \cdot h} \; (Hz)$$

Point A:

$$R_A = 20\log_{10} f_c m - 54 \quad (dB)$$

Point B:

$$R_B = 20\log_{10} f_c m + 10\log_{10}\eta - 45 \quad (dB)$$

$f > f_x$

$$R = R_A + 6\log_{10}\left[\frac{f}{f_c}\right] \quad (dB)$$

$f_c < f < f_x$

$$R = R_B + 9\log_{10}\left[\frac{f}{f_c}\right] \quad (dB)$$

$0{,}5 f_c < f < f_c$

$$R = R_A + \left(\frac{R_B - R_A}{f_c - 0{,}5 f_c}\right) \cdot \log_{10}\left[\frac{f}{0{,}5 f_c}\right] \quad (dB)$$

$f < 0{,}5 f_c$

$$R = R_A + 6\log_{10}\left[\frac{f}{0{,}5 f_c}\right] \quad (dB)$$

Single Panel Sound Insulation Index R Prediction Model in Third-Octave Bands (Davy, 1990)

Input:
Surface mass - m (Kg/m^2)
Panel thickness - h (m)
Material longitudinal speed - c_L (m/s)
Loss factor - η
Panel Height - H (m)
Panel Width - W (m)

Calculate:

Critical Frequency:

$$f_c = \frac{c^2}{1{,}8 \cdot c_L \cdot h} \; (Hz)$$

Panel Area:

$$A = HV \; (m^2)$$

$0 < f < 0{,}8 f_c$

$$R = 20\log_{10}\left[\frac{\pi f m}{\rho c}\right] + 20\log_{10}\left[1-\left(\frac{f}{f_c}\right)^2\right] - 10\log_{10}\left[\log_e\left(\frac{1-a^2}{1+a^2\cos^2\theta_L}\right)\right] (dB)$$

$$a = \left(\frac{\pi f m}{\rho c}\right)\left[1-\left(\frac{f}{f_c}\right)^2\right] \qquad \theta_L = \cos^{-1}\sqrt{\frac{\lambda}{2\pi\sqrt{A}}} \; (º) \qquad \rho c = 400 \; (Rayl)$$

$0{,}8 f_c < f < f_c$

$$R = \max\left\{\begin{array}{c} 20\log_{10}\left[\pi f m/(\rho c)\right] + 10\log_{10}\left[2\eta f/(\pi f_c)\right] \\ ; \\ 20\log_{10}\left[\pi f m/(\rho c)\right] + 10\log_{10}\left[2\eta\Delta_b/\pi\right] \end{array}\right\}$$

$\Delta_b = 0{,}236$ $\qquad$ $\rho c = 400 \; (Rayl)$

$f_c < f < 1{,}2 f_c$

$$R = \max\left\{\begin{array}{c} 20\log_{10}\left[\pi f m/(\rho c)\right] + 10\log_{10}\left[\left(\frac{2\eta}{\pi}\right)\left(\frac{f}{f_c}-1\right)\right] \\ ; \\ 20\log_{10}\left[\pi f m/(\rho c)\right] + 10\log_{10}\left[2\eta\Delta_b/\pi\right] \end{array}\right\} (dB)$$

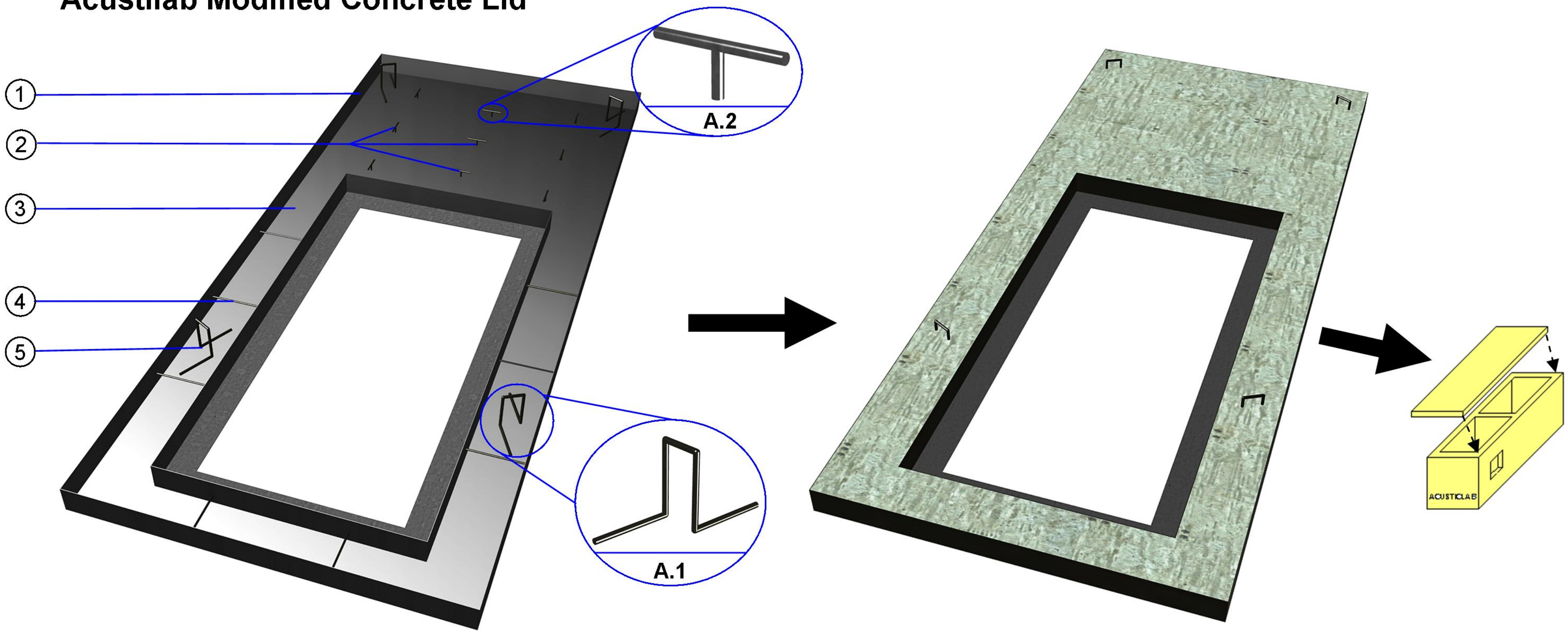
$\Delta_b = 0{,}236$ $\qquad$ $\rho c = 400 \; (Rayl)$

$f > 1{,}2 f_c$

$$R = 20\log_{10}\left[\pi f m/(\rho c)\right] + 10\log_{10}\left[\left(\frac{2\eta}{\pi}\right)\left(\frac{f}{f_c}-1\right)\right] (dB)$$

$\rho c = 400 \; (Rayl)$

Acustilab Modified Concrete Lid

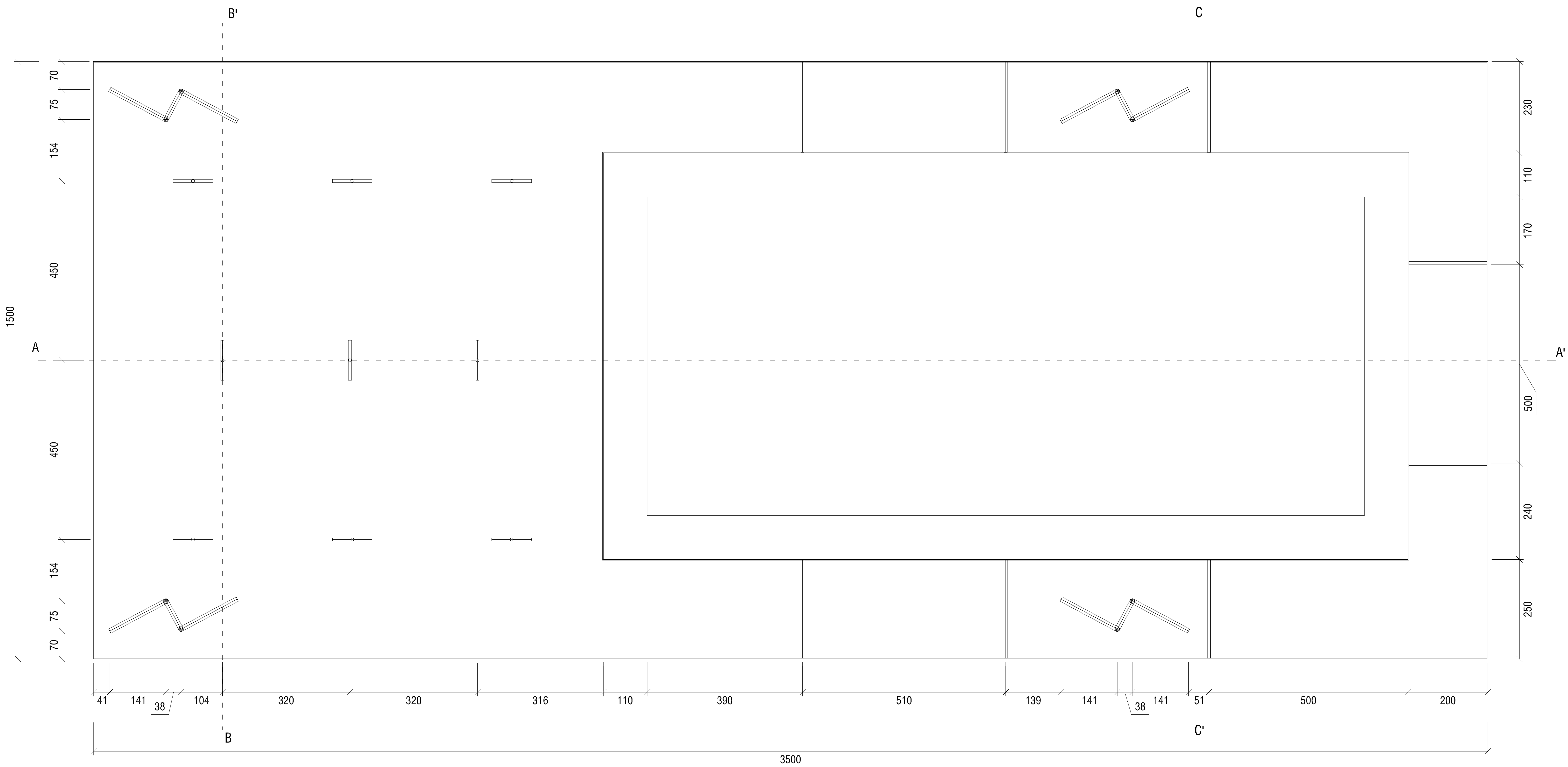


a) Steel formwork

b) Steel formwork with fibre-reinforced concrete

Key:

1 —Steel plate (2 mm thick)
2 —T-shape pieces, made of two 8 mm diameter steel rods, welded together (see zoom-in **A.2**)
3 — 5 mm thick steel plate (1500x3500 mm^2) which serves as the base of the metal frame, with 800x1800 mm^2 opening
4 — Steel bar (8 mm diameter) soldered to the inner walls of the metal frame
5 — Steel handle composed of a bent steel bar of 20 mm in diameter, welded onto the floor of the metal frame (see zoom-in **A.1**)

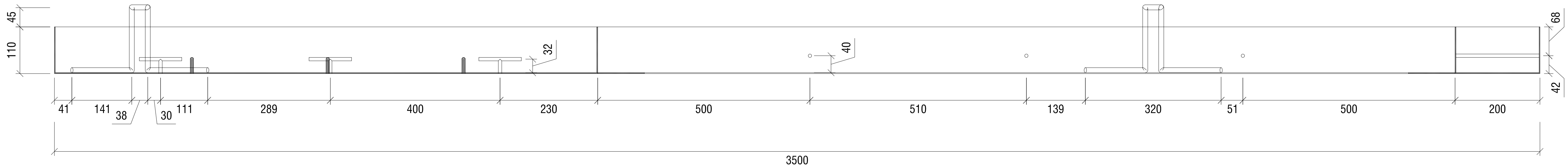


Acustilab Modified Lid - Steel Formwork Design

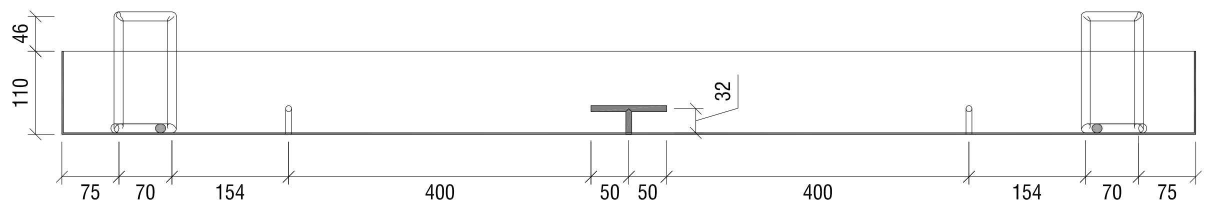
Plan

Scale 1:10

Cross-section A-A'



Cross-section B-B'



Cross-section C-C'

Acustilab Modified Lid - Steel Formwork Design

Cross-section A-A'
Cross-section B-B'
Cross-section C-C'

Scale 1:10

8.2 TABLE OF SYMBOLS

A – arbitrary sound wave amplitude
A_r – surface area (m^2)
A_2 – equivalent absorption area (m^2)
a_t – sound power transmission coefficient
c – sound wave speed of propagation in a medium (m/s)
c_L – speed of propagation of a longitudinal sound wave in a medium (m/s)
E – Young modulus (Pa)
f – frequency (Hz)
f_0 – first resonance frequency (Hz)
f_c – critical frequency (Hz)
f_s – Shear wave cross-over frequency (Hz)
h – thickness of a building element (m), Ch.3.6; thickness of a liquid layer (cm), Ch.6.1
k – wave number (m^{-1})
L – distance travelled by a sound wave (see 3.5)
L_p – sound pressure level (dB)
m - Surface mass of a building element (kg/m^2)
p – sound pressure (Pa)
P_0 – mean fluid pressure (Pa)
p_0 – static pressure or reference pressure (Pa)
p_i – incident sound wave pressure (Pa)
p_{max} – maximum pressure (Pa)
p_r – reflected sound wave pressure (Pa)
p_{rms} – root-mean-square sound pressure (Pa)
p_t – transmitted sound wave pressure (Pa)
R – gas constant (J/kg/K)
R – sound reduction index or sound insulation (dB)
R_N – sound reduction index at normal incidence (dB)
R_{Snell} – sound insulation or transmission loss due to a change in media with oblique incidence (dB)
R_W – weighted sound reduction index according to ISO 717-1 (dB)
$R_{W.liq}$ – weighted sound reduction index prediction for a liquid (dB)
T – period of a wave (s)
T_K – absolute temperature (Kelvin)
v – particle velocity (m/s)
Z – impedance (Rayl ≡ Pa.s/m)
γ – adiabatic bulk modulus
η – total loss factor
θ_{cr} – critical angle of incidence (º)
θ_i – angle of incident wave (º)
θ_L – limiting angle (º)
θ_r – angle of reflected wave (º)
θ_t – angle of transmitted wave (º)
λ – wavelength (m)
ρ – density (kg/m^3)
ρ_0 – mean density of air (kg/m^3)

τ – sound transmission coefficient
φ – phase angle (º)
ω – angular frequency (rad/s)