Universidade de Aveiro
2022

Lucas Azevedo das Neves

Modular Concept of a Heat Pump focusing on Acoustic Impact Reduction

Conceito Modular de uma Bomba de Calor com foco na Redução do Impacto Acústico

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Trabalho de projeto apresentado à Universidade de Aveiro para cumprimento dos requisitos necessários à obtenção do grau de Mestre em Engenharia Mecânica, realizada sob orientação científica de Rui António da Silva Moreira, Professor Auxiliar do Departamento de Engenharia Mecânica da Universidade de Aveiro e de Doutor Jürgen Herbst, Engenheiro Acústico do Departamento TT-RHP/Eng-Av da Bosch Termotecnologia S.A..

Este trabalho teve o apoio financeiro dos projetos UIDB/00481/2020 e UIDP/00481/2020 - FCT - Fundação para Ciência e Tecnologia; e CENTRO-01-0145-FEDER-022083 - Programa Operacional Regional do Centro (Centro2020), no âmbito do Acordo de Parceria Portugal 2020, através do Fundo Europeu de Desenvolvimento Regional.

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Agradecimentos / Acknowledgements

I am in great gratitude with my supervisors, Prof. Dr. Rui Moreira, Dr. Jürgen Herbst and Dra. Idalina Alcântara for all the support, valuable suggestions and guidance during the development of this master thesis work. I would like to warmly thank to the BOSCH development teams, with a special thank to the Acoustic, ERM and Super-Structure teams for making this project possible. With a special thank to João Duarte, Diogo Oliveiro, Fábio Pereira and Artur Cruz. I am also extremely grateful to my friends who made this journey more than just an academic course but rather an unforgettable and valuable fraternal experience. Finally, a huge thank to my family, specially my parents and brothers for all the precious support and unconditional love shared during these last years.

Keywords Noise; Modular Heat Pump; Acoustic Enclosure; Acoustic Impedance; Structure-Borne Noise; Wood-Plastic Composite; Sound Power; Tonality; User Experience

Abstract

The aim of this work is to develop a prototype of a modular heat pump with focusing on noise reduction by separating the two main noise sources of a heat pump: the compressor and the fan. The first part of this master thesis work consisted of developing two modules, the compressor and the fan modules from a monobloc BOSCH heat pump. The compressor, the refrigerant circuit, the inverter, the expansion valves and the electronic board were installed into the compressor module while the evaporator and the fan were installed in the fan module. Different rapid prototyping technologies were used in to order to obtain the needed components, such as, laser cut, hot wire cut and metal bending. As a result, a functional modular heat pump was developed presented as a solution that allows the end user to manage the preferred location to install both modules and consequently the two main noise sources. The second part included the development of a second sound barrier around the compressor module. Hence, an acoustic enclosure was built. The development of the enclosure was based in three acoustic principles: high acoustic impedance gradient between materials, sound leakage insulation and absence of structure-borne noise. Consequently, a study was conducted about the best material to use taking into account several high density materials. The material chosen was WPC (Wood plastic composite), a composite of rice husk and PVC (Polyvinyl chloride) scraps. In addition, an air gap between the enclosure and the compressor module was designed in order to avoid structure-borne noise. Sound measurements were performed to the compressor module with and without the acoustic enclosure around it. The compressor module solely, presented values of sound power of 50,8dB and 53,3dB in night and max day modes, respectively, correspondingly a reduction of 6,7dB and 10,7dB when compared with the initial monobloc unit. However, the most significant result was obtained with the WPC enclosure, which presented values of 35,2dB and 35,8dB in night and max day modes, respectively. These values demonstrate an excellent behavior of the developed enclosure with regard to sound reduction since they are 22,3dB and 28,2dB lower than the values of the initial monobloc unit. On the other hand, the developed solution presented some tonality problems which may increase the level of sound annoyance. Lastly, a user experience was realised what proved to be very useful since it provided valuable information about market chances, preferred places to install the modules, positive and negative aspects and improvement suggestions.

Palavras-chave

Ruído; Bomba de Calor Modular; Invólucro Acústico; Impedância Acústica; Ruído Estrutural; Composto Madeira-Plástico; Potência Sonora; Tonalidade; Experiência do Utilizador

Resumo

O objetivo deste trabalho é desenvolver um protótipo de uma bomba de calor modular com foco na redução de ruído através da separação das duas principais fontes de ruído: o compressor e a ventoinha. A primeira parte deste trabalho de dissertação de mestrado consistiu no desenvolvimento de dois módulos, o módulo do compressor e o módulo do ventilador, partindo de uma bomba de calor BOSCH do tipo monobloco. O compressor, o circuito refrigerante, o inversor, as válvulas de expansão e a placa eletrónica foram instalados no módulo do compressor enquanto que o evaporador e o ventilador foram instalados no módulo do ventilador. Diferentes tecnologias de prototipagem rápida foram utilizadas para obter os componentes necessários, como corte a laser, corte com fio quente e quinagem de metal. Como resultado, uma bomba de calor modular e funcional foi desenvolvida e apresentada como uma solução que permite ao utilizador final gerir o local preferido para instalar ambos os módulos e, consequentemente, as duas principais fontes de ruído. A segunda parte contempla o desenvolvimento de uma segunda barreira de som em torno do módulo do compressor. Assim, foi construído um invólucro acústico. O desenvolvimento deste baseou-se em três princípios acústicos: gradiente de alta impedância acústica entre materiais, isolamento de fugas de ondas sonoras e ausência de ruído transmitido por componentes rígidos (structure-borne noise). Consequentemente, foi realizado um estudo sobre qual o melhor material a utilizar tendo em conta diversos materiais de alta densidade. O material escolhido foi o WPC (*Composto Plástico-Madeira*), um composto de casca de arroz e restos de PVC (*Policloreto de vinila*). Além disso, um espaço de ar entre o invólucro e o módulo do compressor foi projetado a propagação de ruído proveniente de componentes rígidos (structure-borne noise). Foram realizadas medições de som ao módulo do compressor com e sem o invólucro acústico. O módulo do compressor apresentou valores de potência sonora de 50,8dB e 53,3dB para os modos noturno e máximo diurno, respectivamente. O que significou uma redução de 6,7dB e 10,7dB quando comparado com a unidade monobloco inicial. No entanto, o resultado mais significativo foi com a utilização do invólucro em WPC que apresentou valores de 35,2dB e 35,8dB para os modos noturno e máximo diurno, respectivamente. Estes valores demonstraram um excelente comportamento do invólucro desenvolvido no que diz respeito à atenuação de ruído, uma vez que apresenta valores 22,3dB e 28,2dB menores que os valores da unidade monobloco inicial. Por outro lado, a solução desenvolvida apresentou alguns problemas de tonalidade o que poderá significar um aumento do nível de incómodo sonoro. Por fim, foi realizada uma experiência de utilizador que provou ser batante útil, pois forneceu informações valiosas sobre oportunidades de mercado, locais preferenciais para instalação dos módulos, aspectos positivos e negativos e sugestões de melhorias.

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List of Acronyms

ANSI - American National Standards Institute
ASTM - American Society for Testing and Materials
CAD - Computer Aided Design
CLD - Constrained-layer Damping
CM - Compressor Module
CO2 - Carbon Dioxide
DIN - German Institute for Standardisation
EEA - European Environment Agency
EN - European Standards
END - Environmental Noise Directive
EPP - Expanded Polypropylene
EU - European Union
FM - Fan Module
HDPE - High Density Polyethylene
HMI - Human Machine Interface
HP - Heat Pump
HPP - High Price Product
IDU - Indoor Unit
ISO - International Organization for Standardization
MDF - Medium Density Fiberboard
NOISE - Noise Observation and Information Service for Europe
ODU - Outdoor Unit
OSB - Oriented Strand Board
PP - Polypropylene
PS - Polystyrene
PVC - Polyvinyl chloride
SEA - Statistical Energy Analysis
SI - International System
SPL or L_p - Sound Pressure Level or Sound Power Level
STL - Sound Transmission Loss
UV - Ultraviolet
UX - User Experience
WHO - World Health Organization
WPC - Wood Plastic Composite

Part I

Framework and Objectives

Chapter 1

Introduction

In this chapter, a brief introduction to the research work is presented, encompassing motivation and main goals. Moreover, a short description of what a heat pump consists of and a description of the work plan.

1.1 Context

The research work was developed in an industrial context. It was part of a project with BOSCH Termotecnologia S.A.'s acoustic team to reduce acoustic emissions.

The majority of the work was done at BOSCH facilities more specifically at the thermodynamic and acoustic laboratories at *Bosch Termotecnologia Aveiro*.

1.2 Motivation

In the 21st century, one of the greatest challenges is the anthropogenic climate change due to the wasteful consumption of fossil fuels.

Global climate is changing faster than ever, and the european citizens are more aware about it. Appropriate measures must be taken and act with urgency in order to manage this challenge in the best interest of future generations.

Energy used for heating and cooling represents around 50% of the EU's absolute essential energy interest [1]. Therefore, switching from inefficient to environmentally friendly heating systems can result in significant savings in primary energy resources.

Heat pumps are outlined as the most effective solution for the future of the heating industry because of the pursuit of low emissions of carbon compounds and derivatives notably aspired to by developed countries. Green electricity combined with heat pump technology can lead to virtually zero CO_2 emissions [2].

Even though heat pumps are among the most energy-saving household appliances, they also emit sound. Noise pollution from heat pumps is related to their components operation (fans, compressors and refrigerant circuits). With the use of heat pumps tending to increase, the amount of noise pollution produced by them is becoming significant.

In addition, households are where the human spends the majority of its time including the sleeping hours, so it is important to limit the noise levels in the surroundings of a residential facility.

Over daytime, the World Health Organization (WHO) recommends that noise values for outdoor residential areas should not exceed 50 dB A-weighted [3] in order to decrease the levels of sound annoyance. During night hours, WHO imposes a more restrict limit of only 30 dB A-weighted indoors with the purpose of guaranteeing healthy conditions during the sleeping cycle of a human being.

Health effects of noise pollution include auditory and non-auditory effects [4]. Noise-induced hearing loss and tinnitus are among the auditory health effects [4, 5]. Health effects associated with non-auditory effects include disturbances in the sleeping cycle; mental disorders such as excessive stress and irregular mood changes; physical symptoms, such as excessive fatigue and migraines; decrease of learning and cognitive ability; cardiovascular problems, among others [5].

This problem must be addressed by optimizing heat pump appliances in terms of sound emission in order to meet the needs of the user and lead to a less noisy world.

1.3 Heat pump

A heat pump consists of four main elements: a compressor, a condenser, an expansion valve, and an evaporator. This device transfers thermal energy in the opposite direction to where it would normally flow in nature. The exchange of energy occurs through a refrigerant contained inside of a closed circuit that undergoes through thermodynamic changes in order to achieve the desired temperatures of the system [6].

Heat pumps can run on three different energy sources: air, geothermal, and water. These energy sources can be used for central heating and domestic water heating (showers, tap water, among others) [6].

The majority of residential heat pump systems based on air energy sources can be divided into three main categories: full monobloc, semi monobloc and split units.

Full monobloc heat pumps consist of only one piece, the outdoor unit. This unit is characterised by including a plate heat exchanger responsible for transferring the heat from the refrigerant circuit into a water circuit which is directly connected to the central heating by water hoses.

A semi monobloc unit is similar to a full monobloc because it is composed of an outdoor unit with a plate heat exchanger just like the full monobloc. However the water circuit is not directly connected to the central heating but rather connected to an indoor unit which has one more heat exchanger separating the water line that comes from the outdoor unit from the central heating water circuit. Moreover, with this second heat exchanger it also possible to provide domestic water heating with the implementation of a water tank in the indoor unit.

In a split unit, which is composed of an outdoor unit and an indoor unit, the outdoor heat exchanger is absent, whereas the indoor unit has one. With this, the connection between these two units is made by a refrigerant line. Moreover, the indoor unit can also be equipped with a water tank in order to perform domestic water heating.

In the case of this study, a semi monobloc unit was used: AirX 90 BOSCH model (**Figure 1.1**), which is designed for the northern Europe market where the weather is more severe during winter.

Figure 1.1: Example of an AirX 90 used as basis for the modular heat pump concept.

1.4 Aim

Heat pumps generate a considerable amount of noise when they are used frequently in urban areas. Regulations concerning noise pollution and annoyance must be followed to keep residents safe.

In order to accomplish the previous statement, BOSCH has conducted different studies in their products, performing meticulous investigations about the noise generated by each component aiming to reach increasingly lower sound emission values. This master thesis work was developed with the aim of reducing the noise produced by a heat pump following a slightly different approach which is based in two major objectives.

The first consists of separating the two main noise sources: the compressor and the fan [7]. With the intention of creating a modular solution which would include two separated modules, denoted as compressor module and fan module. With this modularity, the user will have the flexibility to manage the placement of the two major sources of noise in different locations.

The second objective is to develop a second sound barrier around the compressor module in order to further reduce the sound emission coming from the compressor.

1.5 Work plan

A work plan is defined in order to achieve the objectives of this Master's Thesis.

BOSCH Termotechnology has a working philosophy based on team work, mutual aid and standardised procedures and therefore the first task was the integration in this context, getting to know the team members and their skills, the available tools and technologies and how to proceed to make use of them.

Secondly, a first contact was made with the heat pump (AirX model) which would be used in the project, to learn about its structure, components, and features by disassembling it.

Afterwards, a modular heat pump was designed and prototyped using the same type of materials, components and following the aesthetic design from the initial heat pump (AirX).

The next step was the development of a second sound barrier for the compressor module, conducting a study about different materials to be used, designing the structure and prototyping the final concept.

In the next task, sound measurements were taken for the compressor module and the second sound barrier. Lastly a user experience was fulfilled with technicians, installers, experts and professors in order to gather feedback about the proof of concept developed.

A summary of the working plan is presented in the **Figure 1.2**.

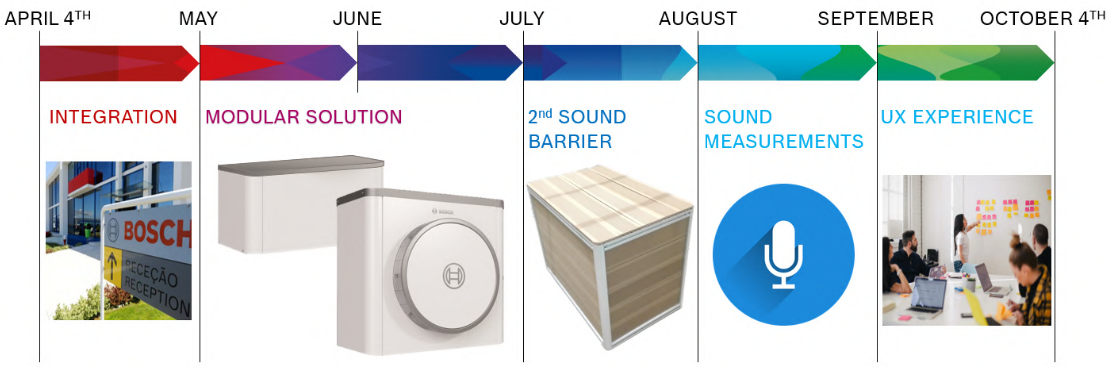


Figure 1.2: Master thesis working plan.

Chapter 2

State of the Art

In the following chapter, a review about acoustic legislation applied to heat pumps is presented. Followed by a revision of different studies about sound reduction approaches carried in the past. Lastly, it is presented some literature about the performance of different materials in relation to sound reduction.

2.1 Heat pumps - standards overview

The population has been increasingly exposed to noise in recent years, what has made sound a priority issue on local, national, and European authorities priority lists [8].

With more awareness about the negative consequences of noise, more policies have been implemented to attenuate excessive levels of sound. These measures aim to restrict the noise from transport vehicles, generators, home appliances, heat pumps, among many other types of machinery [8].

The process of solving noise problems involves not only urban planning rules but also the creation and application of suitable policies on noise control. In the European Union, noise policy focuses primarily on noise reduction via mandatory technical standards for products. Legal instruments for regulating noise emissions involve a series of directives establishing sound emission limits for particular products.

Committed with fifty three Member States, the World Health Organization (WHO) Regional Office for Europe serves a vast geographical region stretching from the Pacific to the Atlantic. With the purpose of solving human health environmental challenges, the European Environment, the Health Process and its Ministerial Conferences are responsible for conducting regional methodologies [9].

In 2010, on the Fifth Ministerial Conference of the WHO European Region, the Parma Declaration on Environment and Health made implicit the need to reduce exposure to noise, and encouraged the WHO to develop appropriate guidelines [9].

For the European Union, the Environmental Noise Directive (END) 2002/49/EC provides an approach in line with the WHO guidelines in order to reduce harmful noise levels and preserving quiet areas by avoiding and preventing their exposure to environmental noise [10]. In addition to collecting noise exposure data and maintaining the Noise Observation and Information Service for Europe (NOISE), the European Environment Agency (EEA) also gives support to the European Commission in the implementation of that directive [8].

In order to accomplish the Seventh Environment Action Programme (which was in action until 2020) and the new one: Eigth Environment Action Programme (up to 2030), currently in development, the Environmental Noise Directive, 2002/49/EC, becomes a crucial legislative tool [11]. As outlined in the directive, methods for collecting noise data are described in it. The END data output are supposed to serve as a basis for developing noise reduction measures at the source.

The Directive 2002/49/EC has been transposed into all member states, however just some countries have adopted it as the basis for environmental noise regulations. Others, in particular the ones with previous specific national laws, have used the past national descriptors.

Noise Directive 2002/49/EC was fundamental to define more specific standards at order to test and rate heat pumps in a noise level.

Those specific standards define a set of conditions to conduct sound measurements properly. For instance, the use of a anechoic, semi-anechoic or reverberant room will depend on the sound parameter to be measure. Moreover, the installation conditions are also standardised as well as the working conditions of the equipment to be tested.

The EN 12102 standard concerns about sound power measurements for space heating and cooling applications when using, air conditioners, heat pumps or liquid chilling packages.

The actual version of the EN 12102 standard was published in 2013 and revised in 2017 under the title: *"Air conditioners, liquid chilling packages, heat pumps and dehumidifiers with electrically driven compressors for space heating and cooling Measurement of airborne noise - Determination of the sound power level".*

CEN/TC113 - Heat pumps and air conditioning units is actual owner of the standard and was also responsible for the development of it.

2.2 Sound reduction approaches

There has been an increase of research work about sound reduction approaches.

A simple method was studied for designing soundproof sheets with high transmission loss as well as high coincidence frequency [12]. As a method for predicting the transmission of sound through lightweight double walls, it was presented a model for computing coupling loss factor [13]. These studies discuss about the exact solution of two-layered infinite plates excited by a plane acoustic wave, for the coincidence frequency and sound transmission loss.

The sound transmission loss of an individual metal panel was determined by the sound intensity method and the conventional method declared by the American Society for Testing and Materials (ASTM E90) and compared with a theoretical model of Statistical Energy Analysis (SEA) [14]. This comparison between two different experimental procedures and a theoretical model allows to sustain small systematic differences between the results obtained from the two experimental methods of measurement. The experimental results and SEA model were found to be in agreement.

The sound intensity method was used for determining the field sound transmission loss [15]. Moreover, in this study a proposal of sound transmission loss data is given for elements of construction walls, such as windows and doors.

It was showed how sills and reveals can affect sound transmission through windows and panels. New data was presented and a design guide for sills and reveals was developed to help designing light panels with higher sound transmission loss values [16].

The Statistical Energy Analysis (SEA) algorithms were applied to calculate the transmission of sound between different acoustic power sources placed in an arbitrary structure with separating panels and cavities representing multiple rooms [17]. In an analysis of sound transmission through lightweight double leaf partitions, it was used the same method of Statistical Energy Analysis (SEA) [18].

They determined that the best SEA model depends on the frequency range assumed and assembly process. A single subsystem of the wall was modeled for low frequencies while for high frequencies the wall was modeled as a number of interconnected subsystems. In a three-dimensional model, the structural connection between the two panels could be modeled as a set of independent points or as a line connection, accordingly to the nail spacing.

By developing a sound enclosure, structural discontinuities may appear and should be take into account due to leakage of sound through grooves and holes. This might lead to significant decreases in insertion losses, mainly in mid frequencies. Depending on the type and size of the leakage paths, some sound frequency singularities can occur, such as resonances which are negative in sound attenuation [19, 20].

2.3 Sound blocking materials

The acoustic properties of rice straw-wood particle composite boards were studied and was found that the sound absorption coefficient of this composite boards, in the frequency range from 500 to 8000 Hz, was higher comparing with other pure wood materials [21].

It has been demonstrated that the acoustic properties of materials can be improved when specific geometrical details are designed, for instance, panels perforated with circular holes.

Studies about the impact of perforated plates, porosity and plate airspace layers on the acoustic properties of materials were carried out [22]. It was found that the porosity of the perforated plate and the density of the porous material significantly influence the acoustic impedance and sound absorption coefficient of the panel.

Thereafter, two research works about sound transmission loss of two different materials are describe in more detail.

2.3.1 Wood plastic composite (WPC)

A study was carried out about improving the sound insulation performance of wood plastic composite panels by using a numerical and experimental approach [23].

In the article, the authors present a numerical and experimental vibro-acoustic analysis of a WPC panel providing several conclusions about the dynamic behavior under vibration and acoustic excitations. In order to evaluate the predictive results, the numerical and measured data were compared [23].

It was used WPC boards obtained from extrusion processing composed of high density polyethylene (HDPE) mixed with 50% of wood fibers derived from pine sawdust, performing a final density of $\rho = 1316$ kg/m^3 [23]. The cross section of the boards can be visualised in **Figure 2.1**.

An elastic putty with a density of about $\rho = 2400$ kg/m^3 was used to connect the boards and avoid sound leakages. The dimensions of the panel were the following ones: $Lx = 0.73$ m, $Ly = 0.73$ m and a thickness of $h = 22$ mm [23]. The final setup of panel can be visualised in **Figure 2.2**.

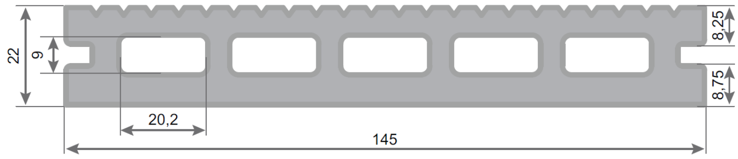


Figure 2.1: Cross section of the used WPC boards [23].

Figure 2.2: Experimental setup of the WPC panel composed by 6 boards [23].

The rectangular panel composed by six WPC boards was inserted between a reverberant and a semi-anechoic chamber. Three omnidirectional speakers were placed inside the reverberant chamber emitting a stationary white noise.

In the semi-anechoic chamber, six microphones were placed to measure the average sound pressure level. Moreover, a manual scan was performed in this chamber using a sound intensity probe.

Therefore, following the Standard ISO 15186-1:2003 the sound transmission loss was calculated from the measured data.

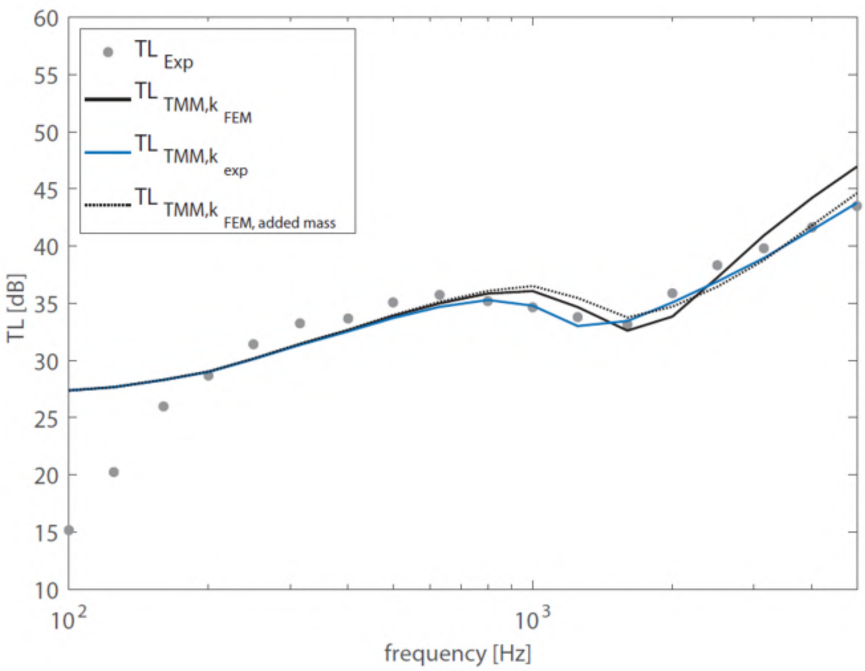


Figure 2.3: Transmission loss in 1/3-octave bands - numerical and experimental results of the WPC panel [23].

The grey dots depicted in the chart represent the sound transmission loss obtained from the experimental analysis. It is possible to visualise that in the lower frequencies the transmission loss presents smaller values ranging from 15dB to approximately 26dB. Thereafter, form a frequency of 200Hz to 1000Hz the values of transmission loss vary between 28dB and 35dB. In the higher frequencies there are also higher values of TL up to almost 45dB at 5000Hz.

Therefore, taking into account the magnitude of TL values presented in this study it is possible to conclude that WPC behaves relatively well regarding sound blocking.

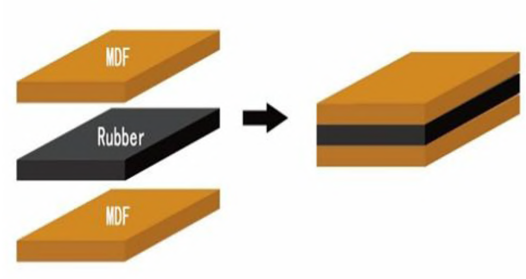
2.3.2 Constrained-layer damping (CLD)

Medium density fiberboard (MDF) and multilayer panels following the constrained-layer damping (CLD) technique were evaluated regarding sound reduction and mechanical properties [24].

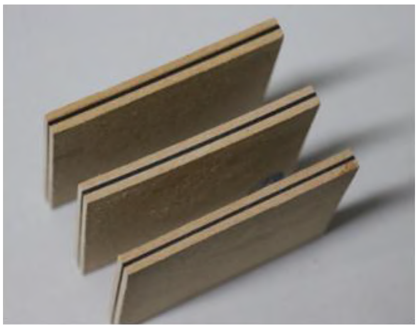
The sound transmission loss (STL) was calculated for three different cases: using only MDF boards varying it thickness, using only rubber material varying it thickness and combining both materials with 2mm MDF boards and different thickness of the rubber layer.

Two samples of MDF boards were tested: one with a thickness of 2mm and another one with a thickness of 6mm.

The CLD samples were composed of two MDF boards having one layer of a rubber damping material between them with a density of 2,300 kg/m^3 illustrated in **Figure 2.4a** and **Figure 2.4b**. It was used the methylene diphenyl isocyanate (MDI) adhesive to glue the three layers. The dimensions of the samples were: 500 mm × 500 mm.



(a) Structure of the composite material [24].



(b) Samples of the composite material [24].

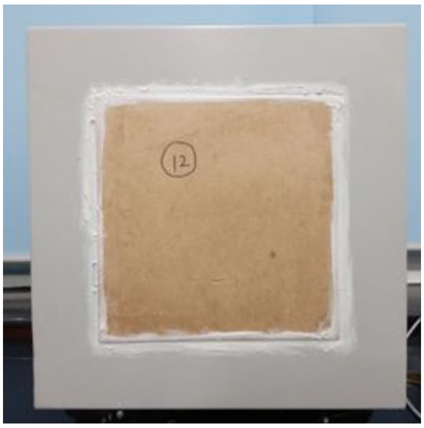
Figure 2.4: CLD material.

In order to calculate the sound transmission loss (STL) of the prepared samples, the authors used a method denominated as *"reverberation chamber-muffler box method"*.

This method consisted of building a small reverberation chamber as pictured in **Figure 2.5a** having a small window where the samples were installed. One sample installed ini the window is marked with number 12 in **Figure 2.5a**.

As illustrated in **Figure 2.5b** a microphone was placed inside the chamber in order to capture sound pressure and send the generated signal into a analyzer. On the outside of the chamber it was used an omnidirectional speaker to emit white noise [24].

ISO 717 was used to formulate the experimental results obtained.



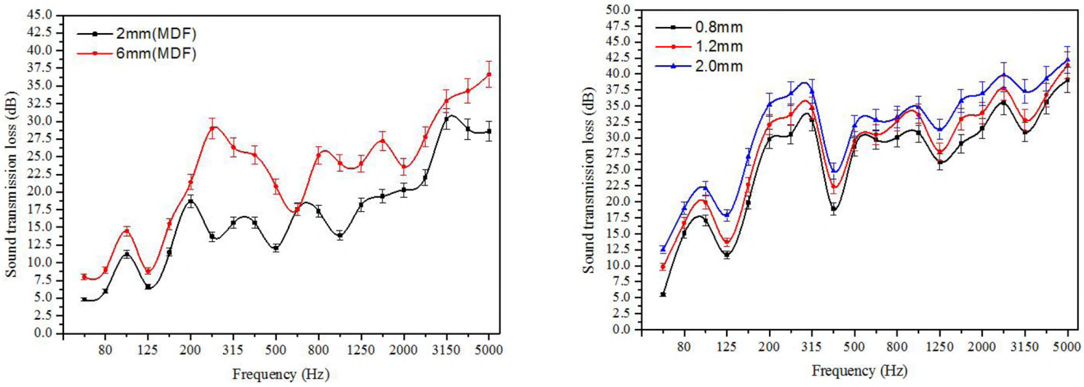
(a) Acoustic test box with sample installed [24].

(b) Inner space of the acoustic test box [24].

Figure 2.5: Acoustic test box and set-up used to perform the measurements.

Analysing the data displayed in the chart of the **Figure 2.6a** it is possible to conclude that the increase of the thickness of the MDF board produced a rise in sound transmission loss. The same behaviour occurred with the increase of the thickness of the rubber layer, depicted in the chart of the **Figure 2.6b**.

This can be justified by the fact that increasing the thickness of the MDF implies an addition of mass into the system what leads to a higher acoustic impedance. Moreover, with the increase of the rubber thickness some resonances were attenuated due to the increase of the performance of the damping material.



(a) STL values for MDF boards varying it thickness [24].

(b) STL values of the CLD samples varying the rubber layer thickness [24].

Figure 2.6: Sound transmission loss (dB) of the CLD boards varying MDF and rubber thicknesses.

In conclusion, the CLD samples performed better than the MDF boards at all frequencies providing a sound transmission loss ranging from 17.5dB to approximately 42.5dB within 100Hz to 5000Hz, whereas, for the MDF this result was comprised between 10dB and 37.5 dB.

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Chapter 3

Theoretical Background

3.1 Fundamentals of acoustics

Humans commonly define noise as sound that is perceived as annoying or harmful for their health. There are many different ways to perceive and react to noise, and one person's "music" may be another's "noise". However, some noises, such as the whistle on a train, are beneficial in order to warn people of potential threatening situations.

Sound can be defined as a pressure disturbance that travels through a material at a particular speed that depends on the material [25].

3.1.1 Sound waves

When solid surfaces vibrate in fluids, they produce sound waves as show in **Figure 3.1**. In response to the vibration, the fluid adjacent to the surface is compressed when it moves to the right and becomes rarefied as the surface moves to the left. This behaviour causes pressure variations above and below the fluid bulk pressure (atmospheric pressure).

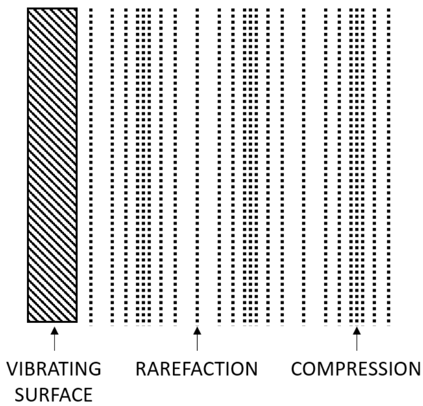

Figure 3.1: Sound wave propagation in materials.

In order to characterise sound waves there are three crucial parameters to consider: wavelength, frequency and wave number.

Simple harmonic waves or sinusoidal waves have a specific frequency (f), which only depends on the excitation of the sound source regardless the medium where it travels through, considering a non-dissipative medium. *Hertz* (Hz) is the unit of frequency named in honor of the German physicist Heinrich Rudolph Hertz, who pioneered research studies in electromagnetism and elasticity [26]. The unit *Hertz* means the same as the unit cycle/sec.

In order to understand the magnitude of the frequency of sound waves considered in noise control, it should be noted that a healthy human hearing is susceptible to frequencies between about 20 Hz and about 20 kHz [27]. Infrasound is defined as frequencies below 20 Hz, and ultrasound is defined as frequencies above 20 kHz.

The period of a wave (τ) is another important parameter which defines the time elapsed between two successive peaks for a simple harmonic wave or the time elapsed to realise a complete cycle, as shown in **Figure 3.2**. Its unit, accordingly to the international system of units (SI), is the second (s).

The period is the reciprocal of the frequency:

$$\tau = \frac{1}{f} \tag{3.1}$$

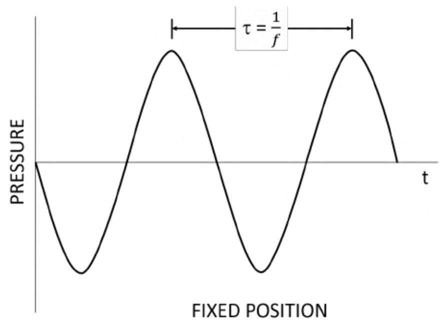

Figure 3.2: Period for a simple harmonic wave.

Wavelength (λ) is another important parameter to describe the behavior of sound waves. If we would take a "picture" of the wave at a specific instant in time, as shown in **Figure 3.3**, we are able to define the wavelength as the distance between two successive peaks of the wave. Its unit, accordingly to the international system of units (SI), is the meter (m).

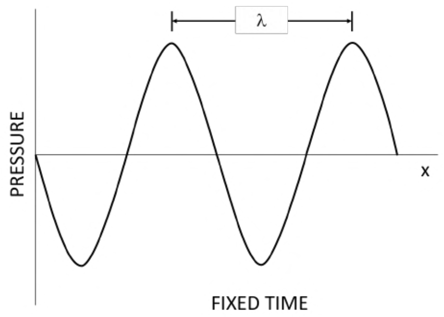

Figure 3.3: Wavelength for a simple harmonic wave.

The relation between the speed of sound and the wavelength can be described accordingly to the following equation:

$$\lambda = \frac{c}{f} \tag{3.2}$$

One more parameter used to characterise a sound wave is the wave number (k), which is defined by:

$$k = \frac{2\pi}{\lambda} = \frac{2\pi f}{c} \tag{3.3}$$

where (λ) is the wavelength; f is the frequency; c is the speed of sound (m/s - SI).

3.1.2 Speed of sound

Considering an ideal gas, the speed of sound can be expressed following the equation below, which depends on the absolute temperature of the gas:

$$c = (\gamma RT)^{\frac{1}{2}} \tag{3.4}$$

where γ is the specific heat ratio, $\gamma = c_p/c_v$; R is the specific gas constant, $R = 287$ J/kg·K for air; T is the absolute temperature, in K.

In general, for a fluid (liquid or gas), the speed of sound is expressed by the following relation:

$$c^2 = \frac{\gamma \beta}{\rho} \tag{3.5}$$

where γ is the specific heat ratio; B is the isothermal bulk modulus; ρ is the fluid density (kg/m^3 - SI).

In a solid material, the speed of sound for transverse sound waves is given by [28]:

$$c^2 = \frac{(1-v)E}{(1+v)(1-2v)\rho} \tag{3.6}$$

where E is *Young*'s modulus; v is *Poisson*'s ratio for the material.

In the case of a thin bar, the speed of sound expression reduces to:

$$c^2 = (\frac{E}{\rho})^{\frac{1}{2}} \tag{3.7}$$

3.1.3 Sound pressure and particle velocity

Sound waves moving through a material are defined as having an acoustic pressure p which is determined by the instantaneous difference between the local pressure P and the ambient pressure P_o. For a simple harmonic wave propagating throughout a plane along one direction, x, the acoustic pressure is expressed accordingly to the next formula:

$$p(x,t) = p_{\max} \sin(2\pi ft - kx) \tag{3.8}$$

The variable $p_{\max}$ represents the amplitude of the acoustic pressure wave. Sound level meters, normally, do not measure the amplitude of a sound wave but rather they measure the root-mean-square pressure, p_{rms}, which is proportional to the amplitude, this actually happens with almost every acoustic instrument. That relation between the p_{rms} and the pressure wave amplitude is described in equation **Equation 3.9**.

Considering $\theta = 2\pi t/\tau$, thus $\mathrm{d}\theta = 2\pi \mathrm{d}t/\tau$. The p_{rms} pressure can be described as the square root of the average of the square of the instantaneous acoustic pressure for one period of vibration τ:

$$(p_{\mathrm{rms}})^2 = \frac{1}{\tau}\int_0^\tau p^2(x,t)\mathrm{d}t = \frac{(p_{\mathrm{max}})^2}{2\pi}\int_0^{2\pi} \sin^2(\theta - kx)\mathrm{d}\theta \tag{3.9}$$

Resolving the integration, we get:

$$\begin{aligned} (p_{\mathrm{rms}})^2 &= \frac{(p_{\mathrm{max}})^2}{2\pi}\left[\frac{1}{2}(\theta - kx) - \frac{1}{4}\sin(2\theta - 2kx)\right]_0^{2\pi} \\ (p_{\mathrm{rms}})^2 &= \frac{1}{2}(p_{\mathrm{max}})^2 \end{aligned} \tag{3.10}$$

For a simple harmonic wave, the relation between the *rms* pressure and the pressure amplitude is given by:

$$p_{\mathrm{rms}} = \frac{p_{\mathrm{max}}}{\sqrt{2}} \tag{3.11}$$

When a sound wave is transmitted through a material, the instantaneous acoustic particle velocity (u) describes the local motion of the particles that constitute the medium where the wave passes through. In engineering analysis, the velocity of acoustic particles is the quantity relevant for measuring energy and intensity of a sound wave.

3.1.4 Acoustic impedance

The specific acoustic impedance is the relation between the rms acoustic pressure and the rms acoustic particle velocity accordingly to the following formula:

$$Z_s = \frac{p_{rms}}{u} \tag{3.12}$$

Pa-s/m is the SI unit for specific acoustic impedance. As a tribute to Lord Rayleigh, who wrote the famous book on acoustics, "*The theory of sound*" [29,30], that combination of units has been named rayl: i.e., 1 rayl ≡ 1Pa-s/m.

The specific acoustic impedance of plane acoustic waves depends only on the fluid properties. It is called the characteristic impedance (Zo) for plane waves, and is calculated as follows:

$$Z_o = \rho c \tag{3.13}$$

where ρ is the specific mass of the fluid and c is the sound velocity in the medium

In cases where the impedance refers to an interface between different media, velocity is properly represented by a vector having the direction normal to the interface boundary. At this point the associated impedance is named as boundary impedance, Z_n [35].

Acoustic impedance is defined in complex notation, it expresses both the magnitude of the pressure-velocity ratio as well as the phase angle between the pressure and velocity waves, having a real part, representing the resistance and an imaginary part related to the reactance.

$$Z_n = r_n + jx_n \tag{3.14}$$

The acoustic impedance can be normalised by dividing this value by the real part of the specific acoustic impedance of the medium.

$$\frac{Z_n}{\rho c} = \frac{r_n}{\rho c} + j\frac{x_n}{\rho c} \tag{3.15}$$

3.1.5 Sound intensity and sound power

Sound intensity (I) can be defined as the sound power (W) transmitted across a unit area or as the average amount of energy transmitted across a unit area per unit time . The SI units for sound intensity are W/m^2.

For plane waves (**Figure 3.4**), the relation between the sound intensity and the sound power per unit area (S) can be represented by:

$$I = \frac{W}{S} \tag{3.16}$$

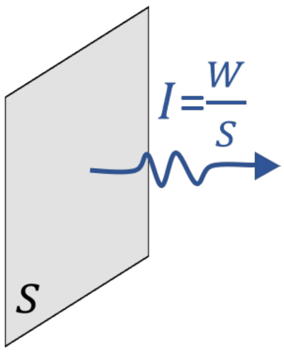

Figure 3.4: Sound wave through a plane.

Assuming a spherical wave (acoustic wave which travels uniformly from the source in all directions - cross section represented in **Figure 3.5**), the acoustic energy is transmitted through a cross area defined as $4\pi r^2$, and the distance from the sound source is r, thus we have the following formula:

$$I = \frac{W}{4\pi r^2} \tag{3.17}$$

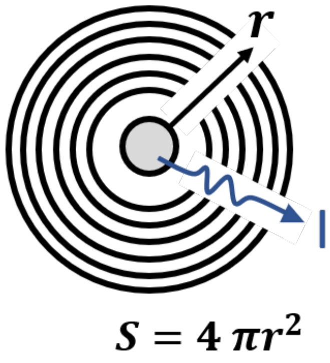

Figure 3.5: Spherical wave cross section.

For general cases where the sound intensity varies with direction because the sound cannot be radiated uniformly from the source, the intensity may be found by:

$$I = \frac{QW}{4\pi r^2} \tag{3.18}$$

Variable Q is designated as directivity factor, which is a dimensionless quantity related to the frequency and direction of the sound wave.

It is important to know that the sound intensity can be also represented as depending on the rms sound pressure. The average sound power per unit area, averaged through one period of time, is represented by:

$$I = \frac{1}{\tau}\int_0^{\tau} p_r ms(x,t)u(x,t)\mathrm{d}t = \frac{1}{2\pi}\int_0^{2\pi} p_r ms(x,t)u(x,t)\mathrm{d}\theta \tag{3.19}$$

where $\theta = 2\pi f t = (2\pi/T)t$.

Using the following equations for the acoustic pressure and for the acoustic particle velocity:

$$p(x,t) = \sqrt{2}p_{\mathrm{rms}}\sin(2\pi t - kx) \tag{3.20}$$

$$u(x,t) = \left(\sqrt{2}p_{\mathrm{rms}}/\rho c\right)\sin(2\pi t - kx) \tag{3.21}$$

and replacing them into the **Equation 3.2**, we obtain:

$$I = \frac{1}{2\pi}\int_0^{2\pi} \frac{2\left(p_{\mathrm{rms}}\right)^2}{\rho c}\sin^2(\theta - kx)\mathrm{d}\theta \tag{3.22}$$

$$I = \frac{2\left(p_{\mathrm{rms}}\right)^2}{2\pi\rho c}\left[\frac{1}{2}(\theta - kx) - \frac{1}{4}\sin(2\theta - 2kx)\right]_0^{2\pi} \tag{3.23}$$

Therefore final expression for the acoustic intensity takes the following form:

$$I = \frac{p^2}{\rho c} \tag{3.24}$$

where $p = p_{rms}$.

The sound power radiated by a source is defined as the integral of the normal component of the intensity over a surface surrounding the source.

Sound power is another important quantity used in acoustics. It is defined as the integral of the normal component of sound intensity along the cross area where the sound wave passes through.

$$W = \int \vec{I} \cdot d\vec{s} \tag{3.25}$$

3.1.6 Decibel scale

Acoustics quantities, such as sound power, sound pressure and intensity, can assume a wide range of values. For instance, a healthy human ear is sensible to sound waves with an acoustic pressure starting at 20μPa, on the other hand the ear can withstand sound waves with a sound pressure as high as 20 Pa during a few minutes without damaging [31]. Moreover, the perception of sound for a human ear does not have a linear relation with the quantities we use to characterise sound waves.

Therefore, a logarithmic scale called the level scale was developed for these reasons, since human hearing is more sensitive to ratios of intensity than to differences in intensity. A quantity level is determined as the logarithm to the base 10 of the ratio between an energy-like quantity and a reference value of that quantity.

Its symbol is usually L followed by a subscript for the quantity it describes. For instance, the sound intensity is defined by L_{I}. Thus, the sound intensity level is designated by:

$$L_{\mathrm{I}} = 10 \log_{10} (I/I_{\mathrm{ref}}) \tag{3.26}$$

This level quantity is dimensionless, however it is attributed the "unit" bel to it, in honor of Alexander Graham Bell. Normally engineers and physics prefer to use the decibel (dB) scale, where 1 decibel is equal to 0.1 bel. To convert from bels to decibles the logarithmic function is multiplied by 10.

The reference quantity for sound intensity assumes the following value:

$$I_{\mathrm{ref}} = 10^{-12}\ \mathrm{W/m^2} = 1\mathrm{pW/m^2}$$

The reference quantities were not defined randomly but rather based on the human perception of each physical quantity. For instance, a healthy ear almost do not hear a sound wave with an acoustic pressure of 20μPa at a frequency of 1000 Hz. Due to this fact, the reference quantity for acoustic pressure assumes that value:

$$p_{\mathrm{ref}} = 20\mu\mathrm{Pa} = 20 \times 10^{-6}\ \mathrm{Pa} \tag{3.27}$$

Considering that the characteristic impedance ρc for air is approximately $Z_0 \approx 400$ rayl at ambient conditions, the sound intensity (**Equation 3.24**) for a sound pressure of 20μPa traveling through ambient air is approximately:

$$I_{\mathrm{ref}} = \left(20 \times 10^{-6}\right)^2 /(400) = 10^{-12}\ \mathrm{W/m^2} = 1\mathrm{pW}.$$

Moreover, the sound power (**Equation 3.25**) obtained from the reference intensity per unit area of 1 m^2 is:

$$W_{\text{ref}} = \left(10^{-12}\right)(1) = 10^{-12}\ \text{W} = 1\text{pW}.$$

Sound pressure is not linearly proportional to the sound intensity (**Equation 3.24**) but rather quadratically proportional, p^2. Therefore sound pressure level is describe as:

$$L_{\text{p}} = 10\log_{10}\left(p^2/p_{\text{ref}}^2\right) = 20\log_{10}\left(p/p_{\text{ref}}\right) \tag{3.28}$$

The expressions for the various "levels" and the reference quantities, according to ISO (International standardization) and ANSI (American National Standards Institute), are given in **Table 3.1**.

Table 3.1: Reference quantities for acoustic levels [32].

Quantity	**Definition**	**Reference**
Sound pressure level	$L_{\text{p}} = 20\log_{10}(p/p_{\text{ref}})$	$p_{\text{ref}} = 20\mu\text{Pa}$
Intensity level	$L_1 = 10\log_{10}(I/I_{\text{ref}})$	$I_{\text{ref}} = 1\text{pW/m}^2$
Power level	$L_{\text{W}} = 10\log_{10}(W/W_{\text{ref}})$	$W_{\text{ref}} = 1\text{pW}$
Energy level	$L_{\text{E}} = 10\log_{10}(E/E_{\text{ref}})$	$E_{\text{ref}} = 1\text{pJ}$
Energy density level	$L_{\text{D}} = 10\log_{10}(D/D_{\text{ref}})$	$-D_{\text{ref}} = 1\text{pJ/m}^3$
Vibratory acceleration level	$L_{\text{a}} = 20\log_{10}(a/a_{\text{ref}})$	$a_{\text{ref}} = 10\mu\text{m/s}^2$
Vibratory velocity level	$L_{\text{v}} = 20\log_{10}(v/v_{\text{ref}})$	$v_{\text{ref}} = 10\ \text{nm/s}$
Vibratory displacement level	$L_{\text{d}} = 20\log_{10}(d/d_{\text{ref}})$	$d_{\text{ref}} = 10\text{pm}$
Vibratory force level	$L_{\text{F}} = 20\log_{10}(F/F_{\text{ref}})$	$F_{\text{ref}} = 1\mu\text{N}$
Frequency level	$L_{\text{fr}} = 10\log_{10}(f/f_{\text{ref}})$	$f_{\text{ref}} = 1\ \text{Hz}$

3.1.7 Octave bands

Human ear is sensitive to acoustic frequencies ranging from about 20 Hz to 20 kHz. Acoustic measuring instruments measure acoustic energy in a range of frequencies instead of measuring the sound level at each of the 15,984 frequencies since it would be unfeasible to process all of these frequencies.

Additionally, humans are also more sensitive to frequency ratios instead of frequency differences, thus frequency ranges generally have terminal frequencies: lower (f_1) and upper (f_2) frequencies which are related by the same ratio. Bandwidth is often use to represent a frequency range.

Acoustic bandwidths are usually expressed as octaves, where an octave is a frequency interval where the upper frequency (f_2) is twice the lower frequency (f_1)

Some frequency range measurements use a more refined division of the range, described as 1/3 octave bands, where:

$$\frac{f_2}{f_1} = 2^{1/3} = 1.260 \tag{3.29}$$

Center frequency of the band f_0 is calculated as the geometric mean between the upper and lower frequency of each bandwidth:

$$f_0 = (f_1 f_2)^{1/2} \tag{3.30}$$

The standard octaves used in acoustics are presented in **Table 3.2**.

Table 3.2: Standard octave bands [33].

Band No.	**Frequency, Hz**		
	f_1	f_0	f_2
12	11	16	22
15	22	31.5	44
18	44	63	88
21	88	125	177
24	177	250	355
27	355	500	710
30	710	1,000	1,420
33	1,420	2,000	2,840
36	2,840	4,000	5,680
39	5,680	8,000	11,360
42	11,360	16,000	22,720

3.1.8 Weighted sound levels

The A-, B-, and C-scales are the common weighting scales found on the vast majority of sound level meters [34]. In the A-weighted scale, low frequencies and high frequencies are ignored in a manner similar to the human ear perception. In this case, frequencies between 500 to 10,000 Hz are the most significant.

The B- weighted scale is used for intermediate levels, similar to A, excepting the attenuation of low frequencies which is much less extreme. The C- scale metric does not discriminate between low and high frequencies, measuring uniformly throughout the 30 to 10,000 Hz range.

The A-scale is widely used in the assessment of possible hearing damage, human annoyance associated with noise, as well as compliance with various noise regulations. C-weighted scale is used in part for measuring sound pressure levels of the broadband and also for recording purposes. On the other hand, the B-scale is seldom used nowadays. In the A-scale, sound levels are described as L_{A} and the units are denoted by dBA. The same occurs for B-scale and C-scale, being: L_{B}, dBB; L_{C}, dBC, respectively.

The weighting factors for the A-scale and C-scale are presented **Table 3.3**. These values are also illustrated in (**Figure 3.6**).

Table 3.3: Weighting values for the A- and C-scales [34].

Center frequency (Hz)	**A-Weighting (dB)**	**C-Weighting (dB)**
31.5	−39.4	−3.0
63	−26.2	−0.8
125	−16.1	−0.2
250	−8.9	0
500	−3.2	0
1,000	0.0	0

Center frequency (Hz)	A-Weighting (dB)	C-Weighting (dB)
2,000	+1.2	−0.2
4,000	+1.0	−0.8
8,000	−1.1	−3.0
16,000	−6.6	−8.5

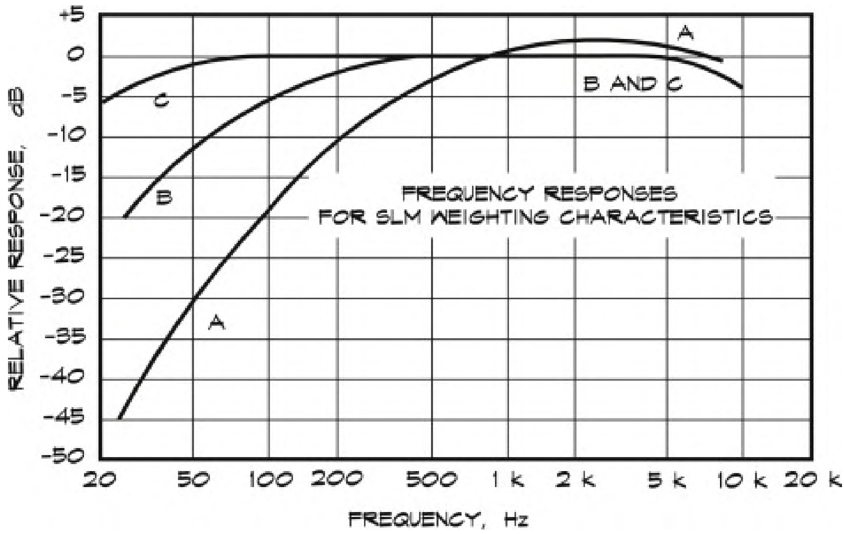

Figure 3.6: Weighting factors for the A-, B- and C-scales [34].

3.2 Insertion loss

The sound power insertion loss, usually defined just by insertion loss, is the most common measurement used to determine the efficiency of a sound enclosure.

It is similar to transmission loss but instead of representing the reduction of sound power level through one specific object (such as a panel, a board or a membrane), it defines the difference between the sound power level emitted by the noise source without any enclosure and the sound power level emitted by noise source with the enclosure. Thereby insertion loss measures the sound power level reduction due to the insertion of an enclosure as a whole.

Mathematically, the definition of insertion loss is equal to the one for transmission loss. However, it is good practice to use different notation:

$$IL_w = 10\log\left(\frac{W_o}{W_E}\right) = L_{Wo} - L_{WE} \quad \text{(dB)} \tag{3.31}$$

An illustration of insertion loss is depicted in Figure 2.4.

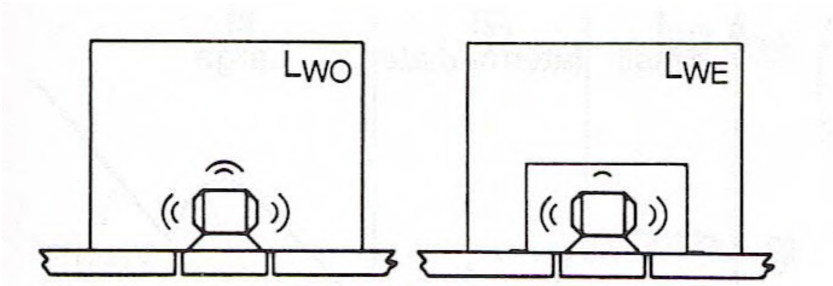

Figure 3.7: Definition of insertion loss [25].

In contrast to transmission loss, the insertion loss is installation sensitive, which means it accounts any effects that are created by installing the enclosure, such as varying source power, changes in the fluid flow or even varying the clearance due to different tightening torques during the installation [35]. Consequently, it provides the most realistic indicator of an enclosure performance.

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Part II

Proof of concept

Chapter 4

Modular Heat Pump

As stated previously in **Section 1.4**, **Chapter 1**, the first objective of this master thesis work was the development of a modular heat pump by separating the two main noise sources: the fan and the compressor.

In this chapter, it is presented the heat pump used to develop the prototype, as well as the development procedure of both modules.

4.1 AirX 90 - Air/Water heat pump

It was used an air to water heat pump as basis for the modular concept. This heat pump collects energy from the outside air and delivers it into the water heating system of the house. Moreover, if combined with an indoor unit equipped with a water tank, it is also capable for deliver heat into the domestic water heating system.

The model used was an AirX 90, which has the following characteristics [36]:

- **Heating power**: 9kW
- **Refrigerant**: R410A
- **Product Segment**: Semi-Monobloc
- **COP @ 7º/35º**: 5,09
- **COP @ 2º/35º**: 4,23
- **Weight**: 115kg

An identification of the main components of the AirX 90 is presented below in **Figure 4.1** and **4.2**.

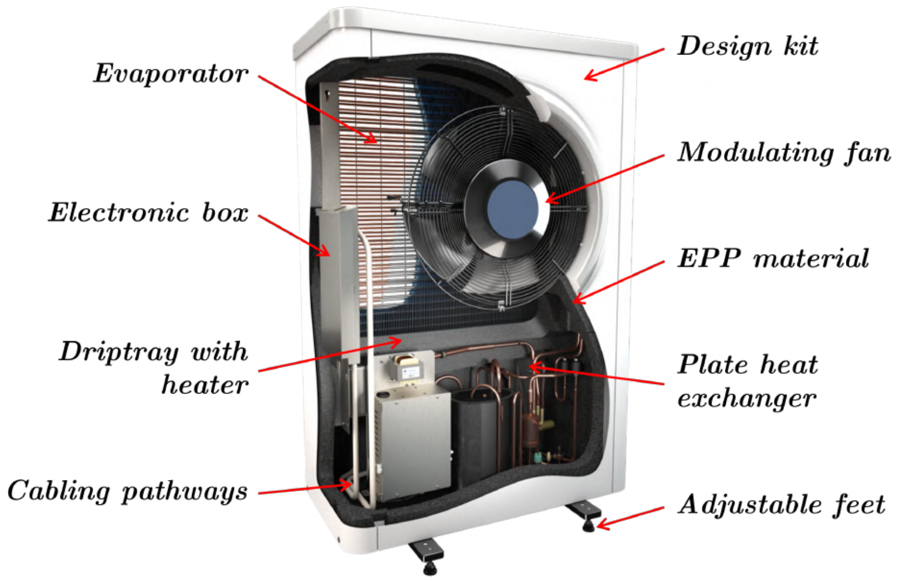


Figure 4.1: AirX 90 components - full assembly [36].

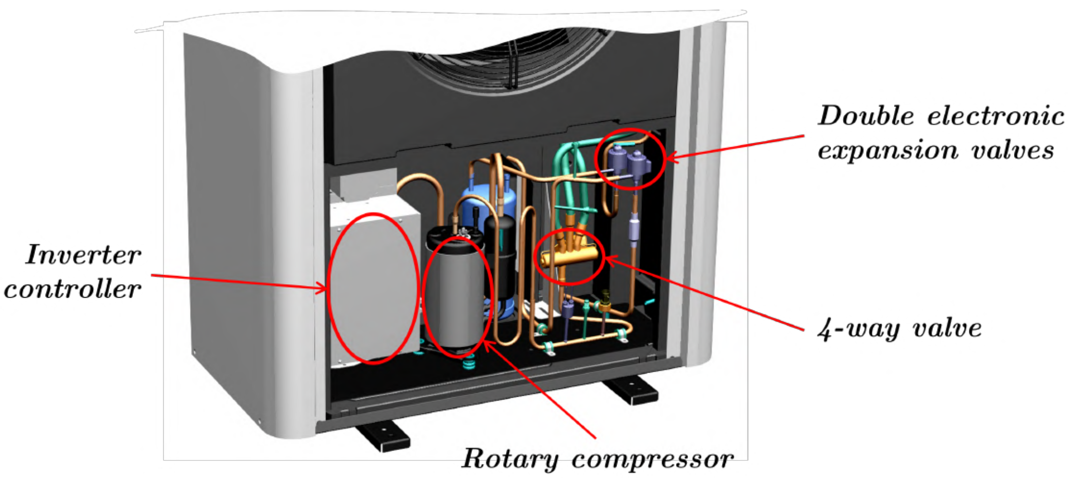


Figure 4.2: AirX 90 components - refrigerant circuit [36].

The heat pump used is classified as a HPP product within BOSCH nomenclature. HPP stands for High Price Product. This category of products is classified as the top-end products within BOSCH portfolio regarding heating appliances. The market for these type of appliances is essentially located at the northern and central Europe where citizens are exposed to more extreme weather conditions compared to the southern Europe.

4.2 CAD design

A CAD design was made before starting developing the proof of concept. The software used was the Siemens NX [37].

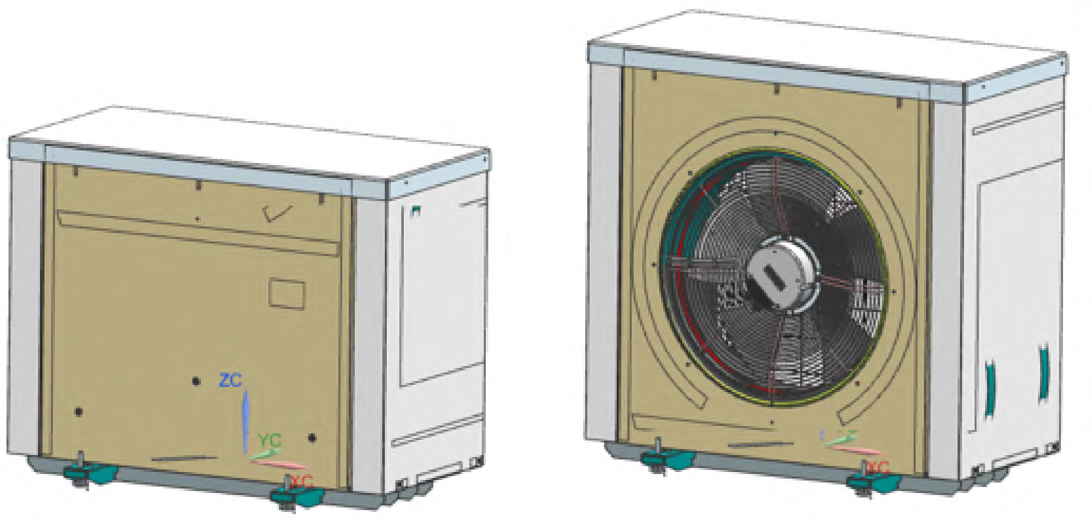
The AirX 90 has two different design kits: BUDERUS and BOSCH. The design kit is basically the exterior panels of the heat pump, composed by two lateral panels, one frontal panel and a top panel. BUDERUS kit presents squared corners in the front edges of the heat pump while BOSCH kit presents filleted round corners. This is the unique difference between them. All the rest is identical (fittings, dimensions, thickness, holes, added bitumen, sound insulation strips). The heat pump previously presented in **Figure 4.1** and **Figure 4.2** is equipped with a BOSCH design kit.

In the CAD designs it was used the AirX 90 equipped with the BUDERUS design kit however the prototype developed is featured with panels following the lines of the BOSCH design kit.

Working on the CAD files provided and drawing some new parts helped to know about all the components constituting the AirX 90, how the heat pump operates and how far the heat pump could be modified without compromising its normal operation.

Therefore, with the CAD tool it was possible to estimate which EPP parts must be cut along with designing new expanded polypropylene (EPP) parts to be prototyped. Moreover, the CAD project was also useful to predict where the Electronic box should be placed as well as to design new outer panels (design kit) for both modules.

The final result of the compressor and fan module CAD designs is illustrated in **Figure 4.3**.



(a) CAD design - compressor module. (b) CAD design - fan module.

Figure 4.3: CAD designs - fan and compressor modules.

4.3 Structural development

4.3.1 EPP structure

The AirX heat pump structure is composed of modular EPP parts. EPP was developed by JSP Company after the development of Polystyrene foam (PS). Nowadays is one of the most widely used foamed polymers in multiple industries, such as: automotive, thermal appliances, construction, packaging, home appliances, among others.

It not only features all the benefits of polypropylene (PP), but also has its own unique properties [38, 39]. PS foam has a working temperature below 80°C, while PE foam is 70 − 80°C but EPP withstands temperatures up to 130°C.

EPP presents excellent mechanical properties, as well as a stable thermal conductivity characterised by homogeneous and closed cell structures, being able to keep these characteristics even in moist environments. As a result, EPP is a good thermal and sound insulator material. A good performance in vibration absorption is also a feature inherent to EPP [40].

Comparing to PS foams, EPP offers a good chemical resistance as well as burning with low toxic gas release, having a higher degradation performance level than PS and PE foams [40].

In the AirX, all components are embedded in EPP modular parts, these parts are the structure of the heat pump. In **Figure 4.4a** it is possible to see the assembly of all EPP parts.

The AirX is characterised by having a fully modular structure. Therefore, it was possible to disassemble the majority of the parts. **Figure 4.4b** illustrates the disassembling process.

(a) AirX 90 - EPP structure.

(b) AirX 90 - partially disassembled EPP structure.

Figure 4.4: AirX 90 - EPP structure and disassembling.

In order to separate the fan and the evaporator from the compressor side, it was necessary to cut the copper pipes that connects the refrigerant circuit to the evaporator, as well as disconnecting the electrical connections coming from the controller board to the fan and to the ambient temperature sensor (which is placed in the top-rear part of the evaporator).

Subsequently, it was possible to start working on the compressor module. As illustrated in **Figure 4.5**, there was the need of prototyping an EPP lid with the aim of closing the top part of the compressor module.

Figure 4.5: Compressor module - EPP structure without lid.

A hot wire was developed to cut EPP parts. Made from a power supply of 19 Volts and 4 Amps, therefore 76 Watts of power.

It was used a steel wire with a diameter of approximately 0,8 mm, re-used from a steel cable used in bicycle brakes. Having a cross section of only 2,01 mm^2 and a length of 1 m, by Joule effect it was possible to achieve considerably high temperatures through the steel wire.

It was designed a sliding structure made of aluminum profiles and wood parts which allowed to operate the hot wire along different heights and different angles, as presented in **Figure 4.6**.

The EPP foam has a melting point of approximately 170ºC [41]. Hence, it was possible to perform clean and precise cuts in EPP parts with the developed hot wire system.

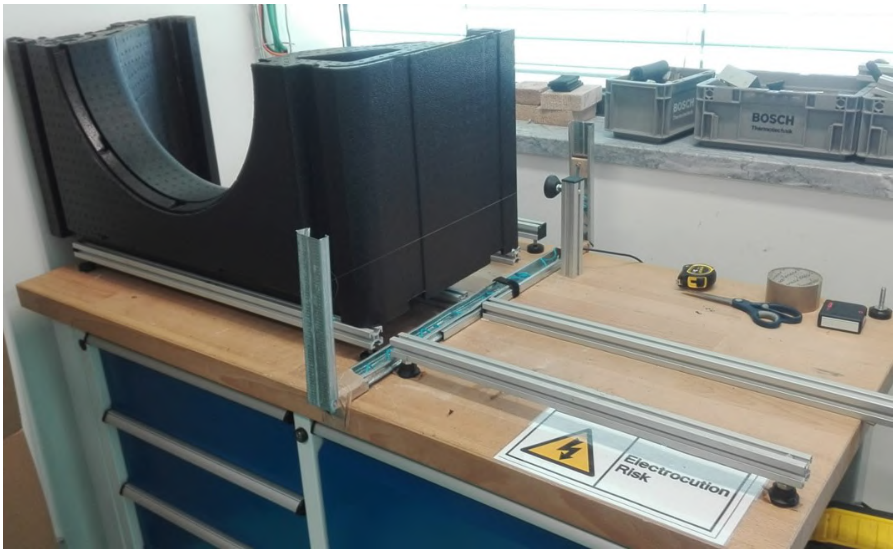


Figure 4.6: Hot wire foam cutter.

With this tool it was possible to prototype the lid for the compressor module. Old EPP parts were used from other similar units. To start with, a mid EPP part from a 90 AirX unit was cut in order to maintain the fittings between the parts, these fittings (**Figure 4.7a**) are essential to avoid sound leakage since they create a more complex path for sound to propagate. This first layer of the EPP lid can be visualised in **Figure 4.7b**.

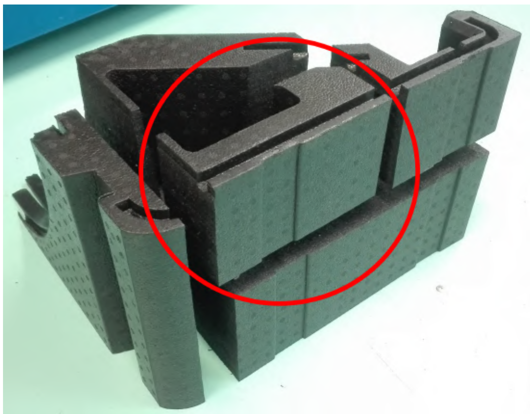
(a) EPP fitting detail.

(b) EPP lid - first layer.

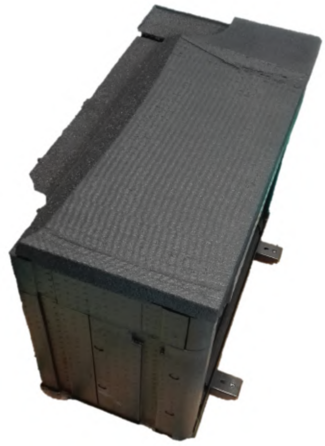
Figure 4.7: Reworks performed on the EPP structure.

As explicit in **Figure 4.7b**, there are some big holes through which sound can travel easily. These holes exist due to the the controller board that used to be located in the lateral part of the EPP structure (marked with a red circle) and for material saving purposes (marked with yellow circles).

Hence, it was necessary to prototype a lateral EPP part (**Figure 4.8a**) and a second EPP layer (**Figure 4.8a**) with 5 centimeters in order to close these holes, using the hot wire foam cutter.



(a) EPP lateral cover.



(b) EPP lid - second layer.

Figure 4.8: Reworks performed on the EPP structure.

The second layer of the lid presents a rectangular shape along its outline, not following the curved geometric lines of the EPP placed below it. To enable the future assembly of the design kit it is crucial that the entire lid follows the geometrical design of the fittings of the panels.

With the purpose of cutting the second layer of the lid following the curved geometrical design it was developed a smaller hot wire foam cutter using the same power supply device, represented in **Figure 4.9a**.

The final result of the compressor module (CM) regarding the EPP structure finalised can be visualised in **Figure 4.9b**.

(a) Small hot wire foam cutter.

(b) CM - EPP structure and final lid.

Figure 4.9: Small hot wire and final EPP structure.

Moving forward to the EPP structure of the fan module (FM), it was used two top EPP parts from the AirX model and assembled together in order to be able to place the fan and the evaporator embedded on it. In **Figure 4.10a**, it is possible to visualise the assembly of the fan and the evaporator and in **Figure 4.10b** the final EPP structure with the second EPP part assembled on top.

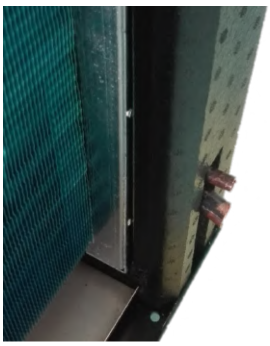
(a) Fan module - assembly.

(b) Fan module - EPP structure.

Figure 4.10: Fan module assembly and EPP structure.

There are two refrigerant lines coming out from the evaporator, the liquid and the gas lines. These lines need to be connected to the refrigerant circuit placed inside the compressor module. To enable this connection, an opening in the bottom EPP part was made, as pictured in **Figure 4.11a**. Therefore, these two copper pipes leave the fan module from the back.

To allow the installation of the fan module at the installation site it was added a baseplate below the EPP structure. **Figure 4.11b** shows the baseplate used which is equal to the one used in the compressor module.



(a) Evaporator refrigerant lines an EPP opening.

(b) Baseplate.

Figure 4.11: Fan module - evaporator refrigerant lines and baseplate.

4.3.2 Design kit

A new design kit was prototyped to be assembled to the EPP structure of both modules. It follows the same design of the BOSCH kit, therefore two pairs of lateral panels were re-used, being cut with new dimensions and new holes were drilled to assemble the rubber pop-nuts. These rubber pop-nuts guarantee the coupling between the lateral panel and the top panel through bolts. **Figure 4.12a** shows the marking lines through which the panels were cut as well as the location of the drilled holes.

On the other end of the panels, there are also two holes where bolts pass through, threading onto the threaded holes of the lateral side of the baseplate. Between the head of the bolt and the panel exists a rubber washer to absorb vibrations.

Figure 4.12b shows one lateral panel used. At the top it is possible to visualise the rubber pop-nuts which enables the coupling to the top panel.

The black material glued inside the panel is a layer of bitumen placed there with the purpose of increase the mass of the panel. With this increase of mass, the overall value of acoustic impedance of the panel is higher, increasing the effectiveness of sound blocking by reflecting more the incident sound waves.

The black material glued inside the panel is a layer of bitumen placed there used to increase the mass of the panel and subsequently the overall value of the acoustic impedance. Increasing the effectiveness of sound blocking by reflecting more the incident sound waves.

(a) Lateral panel - cutting mark.

(b) Lateral panel with bitumen layer.

Figure 4.12: Re-work of the lateral panels.

Regarding the front panel of the compressor module, it was necessary to design and prototype it from scratch. The best process found to prototype it was the bending metal forming process.

First of all it was designed a 3D model of the panel as a sheet object. Afterwards, the flat pattern of the panel was created in *.dxf* format with the purpose of being cut by laser.

Subsequently, using a manual bending press it was possible to shape the panel. The last step was the painting process through spray painting in white color.

Regarding the front panel for the fan module, it was possible to re-use the panel from the initial AirX unit cutting the bottom part with a sheet metal scissor following the correspondent flat pattern. Afterwards, it was used the manual bending press to shape the bottom part of the panel following the details illustrated in **Figure ??**. Finally it was necessary to re-paint the panel in white.

All the panels are made of a zinc coated metal sheet with 0,7mm of thickness.

The final assembly of the design kit on the compressor module can be observed in **Figure 4.13a** and the assembly of the fan module in **Figure 4.13b**.

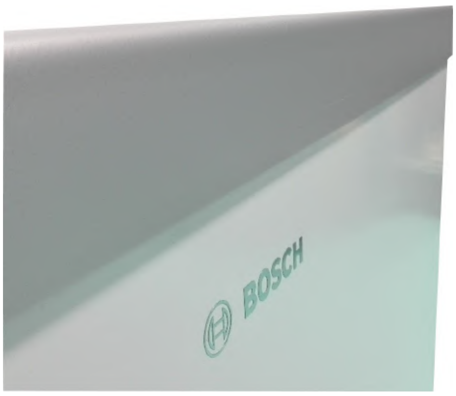
(a) Compressor module - design kit assembly.

(b) Fan module - design kit assembly.

Figure 4.13: Final assembly of the prototyped design kits.

On the front side of the fan a diffuser was installed with the objective of decreasing the noise produced by the air flow created by the fan, (**Figure 4.13b**).

In **Figure 4.14a** and **4.14b** it is possible to observe another detail: the BOSCH logo attached to the frontal panel of the compressor and fan modules.



(a) Compressor module - BOSCH logo.

(b) Fan module - BOSCH logo.

Figure 4.14: Application of the BOSCH logos.

4.4 Drip tray and protection grid

4.4.1 Drip tray

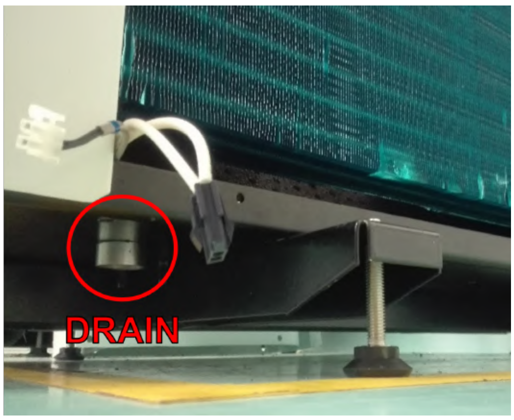
From time to time the heat pump enters in a defrost cycle to melt the ice accumulated in the fins of the evaporator. The defrost cycle is achieved by reversing the heating cycle by changing the state of the 4-way valve.

Therefore, the heat pump changes from heating mode to cooling mode and the evaporator behaves like a condenser, heats up and allows the accumulated ice to melt. Consequently, a considerable amount of water starts dripping from the bottom part of the evaporator.

Aiming the evacuation of the water from the heat pump, a drip tray was installed in the base plate underneath the evaporator. For this, using a step drill, it was possible to drill a hole with 32mm of diameter in the baseplate shown in **Figure 4.15a**.

With this the drain of the drip tray can pass through the drilled hole of the baseplate as presented in **Figure 4.15b** and the user can install a water hose on it to drain the dripped water. The assembly of the drip tray in the baseplate is depicted in **Figure 4.16a**.

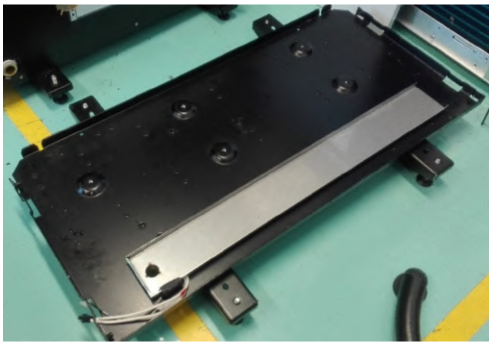
(a) Baseplate hole for drainage.



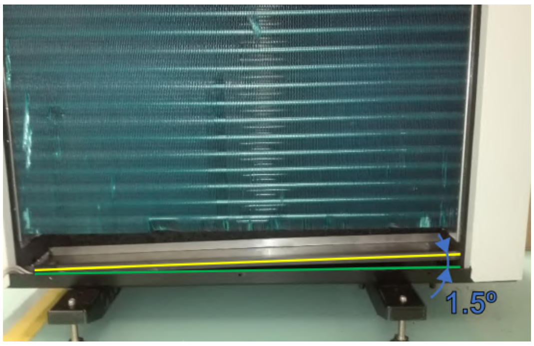
(b) Drip tray drain.

Figure 4.15: Installation of a drain to leak the water that drips from the accumulated ice in the fins of the evaporator.

The drip tray was installed with a slope of approximately 1.5 degrees to allow the flow of the water to the drain, as illustrated in **Figure 4.16b**.



(a) Drip tray assembly in the baseplate.



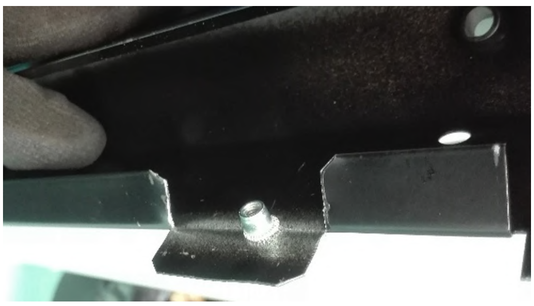
(b) Drip tray slope.

Figure 4.16: Installation of the drip tray under the evaporator.

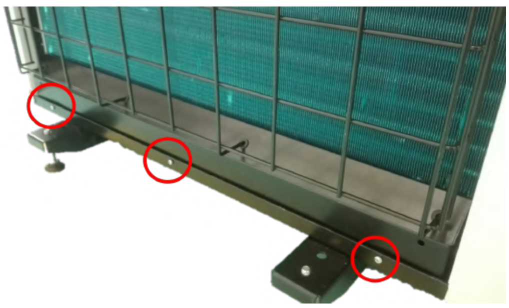
In extreme weather conditions, where negative temperatures can be reached, heating the evaporator is not enough to melt and drain the ice out of the heat pump since when it drips into the drip tray it freezes again. Thereby, the installed drip tray has a heating resistance in the underneath face, which is activated when the heat pump enters in defrost mode, avoiding the accumulation of ice over it.

4.4.2 Protection grid

In order to protect the evaporator and the users from being injured in the fins of the evaporator, the grid from the initial AirX was installed on the back of the module. To accomplish this, it was necessary to re-work the support of the grid (see **Figure 4.17b**) cutting and adding three pop nuts (see **Figure 4.17a**) to allow the assembly to the baseplate of the fan module.



(a) Re-work of the grid support.



(b) Application of pop nuts in the grid support to attach he protection grid.

Figure 4.17: Fan module - re-work of the grid support.

The final result of the assembly of the protection grid can be observed in **Figure 4.18**.

Figure 4.18: Fan module - evaporator with the protection grid installed.

4.5 Electrical connections

In order to keep the heat pump working properly it was necessary to re-work the electrical parts of it.

4.5.1 HP board and power supply

First of all, it was decided to install the controller board (denominated as HP board) in the compressor module which was initially placed on the mid EPP block of the AirX unit. It was placed in the compressor module due to the fact that the majority of the electrical connections coming out from the HP board go to components placed inside the compressor module.

In order to accomplish this, the metal frame where the HP is installed was re-used and consequently re-worked. **Figure 4.19** shows the cuts done to it using a sheet metal scissor.

Moreover, two holes were drilled at the lateral part of the metal frame, as shown in **Figure 4.20a**, with the purpose of fixing the metal frame into the EPP structure by using two plastic bolts. These bolts, shown in **Figure 4.20b**, are characterised by having a big height of the fundamental triangle what is good to guarantee a good tightening torque in foams like EPP.

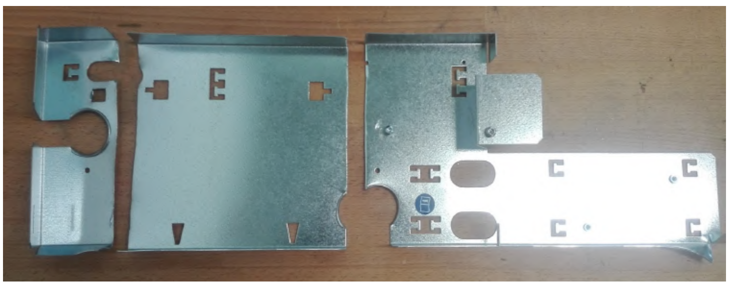
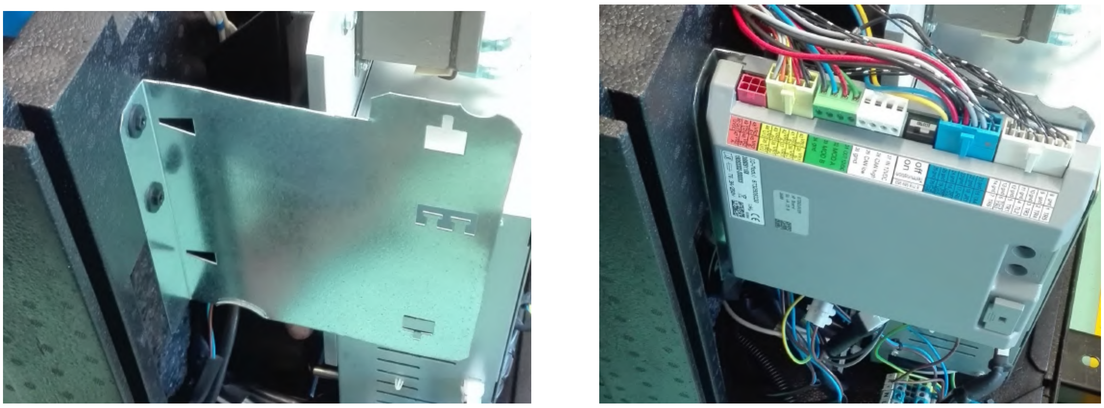


Figure 4.19: Metal frame re-work

(a) Metal frame holes. (b) Bolts for EPP foams.

Figure 4.20: Re-work of the HP board support.

Figure 4.21a shows the assembly of the metal frame into the EPP structure, while **Figure 4.21b** illustrates the final assembly of the HP board in the metal frame. The HP board can be easily taken out and installed back to the frame since it has two snap-fits on its back that clamps into two square holes on the metal frame.

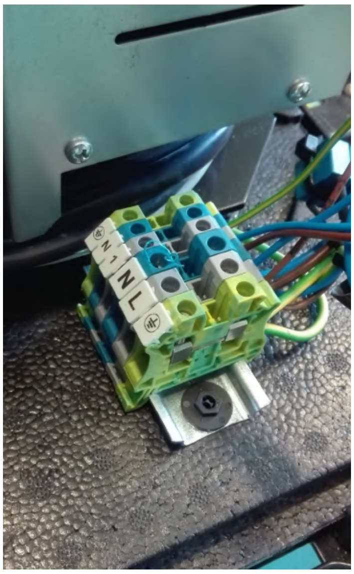


(a) Metal frame assembly. (b) HP board assembly.

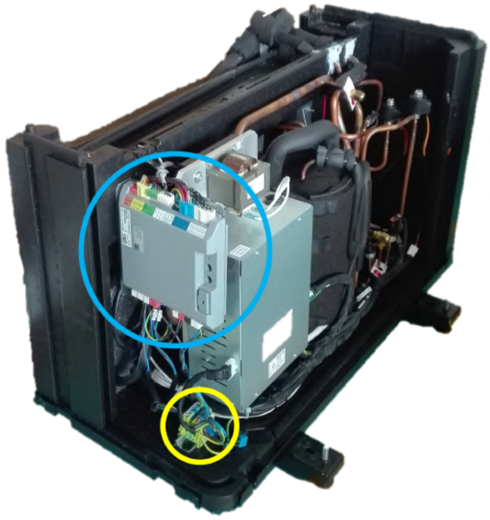
Figure 4.21: HP board assembly.

Moving to the power supply, it was installed a DIN rail in the bottom EPP of the compressor module with 6 terminal blocks (see **Figure 4.22a**) to protect against overcurrent and short-circuiting.

As the unit is single phase, there are a blue terminal for the neutral line, a grey for the phase line and a green for the grounding. After these 3 terminal blocks the current is divided into two, going to the inverter and into other 3 terminals, which are smaller (less current allowed to pass through). These smaller terminals deliver the needed energy to the HP board and the green terminal serves as grounding for it.



(a) DIN rail terminal blocks.



(b) HP board and terminal blocks.

Figure 4.22: HP board assembly.

Figure 4.22b shows the positioning of the HP board marked with a blue circle and the terminal blocks marked with a yellow circle. Both are placed on the left side of the compressor module when viewed from the front side.

4.5.2 Electrical extensions to the fan module

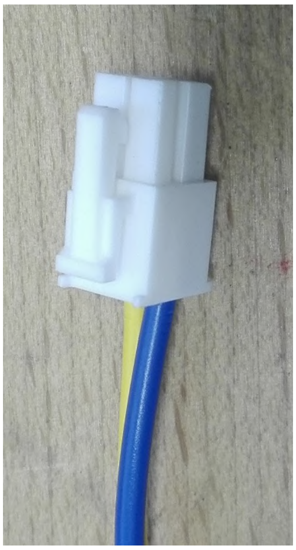
Some components of the fan module need to be connected to the HP board to operate properly. To accomplish this, electrical extensions with five meters long were made.

To operate the fan, two extensions were required, one to supply energy and another one to control its velocity.

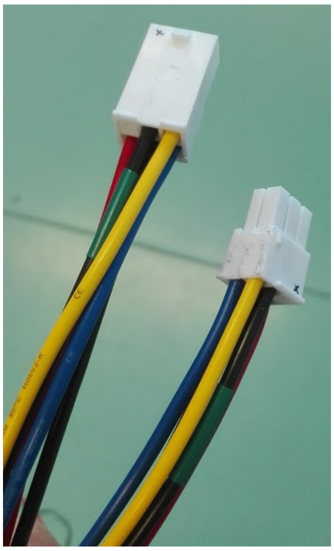
There is an ambient temperature sensor installed on the top part of the fan module which is crucial for the correct operation of the heat pump. This sensor must also be connected to the HP board so another extension were made for this sensor.

Regarding the drip tray, as refereed previously, it has a heating resistance which also needs to be connected to the HP board as well as the temperature sensor attached to the drip tray. Hence two more extensions were made.

With this, five electrical extensions were made. The connectors used were the standard *Mini-fit JR* of 2, 4, and 6 pins depending on the number of wires as illustrated in **Figure 4.23a** and **Figure 4.23b**: a male *Mini-fit JR* of 4 Pins and male and female *Mini-fit JR* of 6 Pins, respectively.



(a) Mini-fit JR - four pins.



(b) Mini-fit JR - six pins.

Figure 4.23: Electrical extensions connectors.

In the fan module all the cables were guided and extended through the EPP structure with the purpose of leaving it out from the back side and from the bottom, as illustrated in **Figure 4.24a**.

Inside the compressor module there are three holes on its back. The one from the left is used to pass the communication cable that goes to the IDU, the one to the right is used to pass the power supply cables form the power grid and there is a spare one in the middle.

Taking advantage of the spare hole, all the electrical cables that go to the fan module were passed through this aperture and connectors were installed on their ends to make it easier to connect and disconnect the wire extensions. **Figure 4.24b** depicts this arrangement.

(a) Fan module - wiring exit.

(b) Compressor module - wiring exists.

Figure 4.24: Electrical cables exits.

4.6 Refrigerant circuit

As the AirX is a semi monobloc heat pump it does not have a pump down cycle. A pump down cycle can be described as an autonomous routine that collects all the refrigerant into a determined part of the refrigerant circuit normally separated by shut-off valves.

The pump down cycle finishes when the heat pump gives a sonorous signal or displays a message in the computer software or in the unit HMI, at this time the technician closes the shut-off valve.

For instance, in a split unit there is refrigerant going from the outdoor unit into the indoor unit. Therefore with the pump down cycle it is possible to collect all the refrigerant into the outdoor unit.

This is very useful for maintenance and transport purposes. For example, if a technician needs to work in refrigerant parts of the indoor unit. With the pump down it is possible to retain the refrigerant in the outdoor unit, avoiding the necessity of evacuating all the refrigerant out of the heat pump.

The prototype developed is not a split unitm however it has two separated modules which separate components of the refrigerant circuit; more precisely the evaporator is separated from the rest of the circuit.

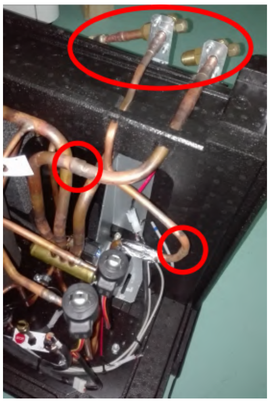
For the heat pump to work it is mandatory that the refrigerant circuit is connected to the evaporator. In line with the split unit concept, it would be very useful if the developed concept would have a pump down system. Hence it was decided to install two shut-off valves at the exits of the refrigerant circuit that goes to the evaporator in the compressor module.

There are two lines coming out from the refrigerant circuit from the compressor module: the liquid line, which is smaller (3/8 inch of diameter), and the gas line (5/8 inch). Therefore, it was used a 5/8 inch shut-off valve and due to the lack of 3/8 inch valves it was used a 1/4 inch instead.

Consequently it was necessary to use copper reductions in the liquid line at both sides of the valve. **Figure 4.25a** shows the reductions, the shut-off valves and the refrigerant flares used. **Figure 4.25b** depicts the refrigerant circuit after brazing all components.

Two copper extension pipes were made in order to have the valves placed outside the compressor module. The union of the extensions with the refrigerant circuit is marked with small red circles while the shut-off valves and the surrounding brazed components are marked with a red ellipse in **Figure 4.25b**.

(a) CM - refrigerant circuit.



(b) CM - refrigerant circuit re-worked.

Figure 4.25: CM - re-work of the refrigerant circuit.

Two metal brackets were prototyped to which the shut-off valves were bolted through two pop-nuts. **Figure 4.26a** shows the brackets and the valves assembled. In **Figure 4.26b** is possible to observe the final assembly of the shut-off valves in the compressor module.

(a) Shut-off valves and bracket.

(b) CM - shut-off valves installed.

Figure 4.26: Compressor module - installation of refrigerant valves.

In the proof of concept developed, the refrigerant connection between both modules was accomplished through two refrigerant flexible hoses for easier maneuverability of the modules while performing tests. However, ordinary copper pipes could be used.

The flexible hoses were five meters long just like the electrical extensions. **Figure 4.27a** shows the flexible hoses connected to the valves with a white insulation foam around them.

To connect the flexible hoses to the evaporator it was necessary to braze two brass flares in the copper pipes exists, as shown in **Figure 4.27b**.

(a) Compressor module - flexible hoses.

(b) Fan module - refrigerant flares.

Figure 4.27: Flexible hoses and refrigerant flares.

4.7 Installation remarks

4.7.1 Pump down

As stated above, the unit used in this work does not have a pump down cycle. However with the implementation of the two shut-off valves it became possible to collect all the refrigerant into the side of the refrigerant circuit that is placed inside the compressor module. Therefore, it is possible to transfer all the refrigerant from the evaporator and flexible hoses into the compressor module.

Nevertheless, the developed prototype does not have a routine implemented at the software level to do the pump down, therefore this procedure must be done manually. The steps to perform the manual pump down are the following ones:

1. Close the liquid line valve;
2. Wait until pressure drops close to zero;
3. Close the gas line valve.

To perform the pump-down, the heat pump needs to be in the heating mode, thus the compressor will suck the refrigerant from the evaporator and hoses into the circuit inside the compressor module.

A manometer must be connected to the shut-off valve placed in the gas line. With this, it is possible to know the pressure between the compressor and the closed valve placed in the liquid line. This pressure valve corresponds to the pressure stated in the step number two. The pressure read in the manometer must be as close to zero as possible (vacuum) meaning that there is no refrigerant in the line.

When the pressure drops like this, the heat pump identifies an error in the system and stops the compressor because it is not prepared to do pump down. Therefore the step three must be done with the compressor running, otherwise the refrigerant will return to the evaporator. There are two ways to know if the compressor is running: by listening it or by checking its state displayed in the computer software.

4.7.2 Refrigerant fill-in

In the first operation of the heat pump it was necessary to fill the refrigerant circuit with R410A. The AirX 90 must be filled with a charge of 2,35kg of R410A. However, we must compensate the extra volume of the refrigerant circuit due to the addition of the flexible hoses between both modules (5 meters long). It was considered an addition of 50 grams per linear meter.

This value was based on the split concepts which defines the necessary refrigerant to be added when two parts of a refrigerant circuit are separated by copper pipes, depending on the extension and unit power.

Ignoring the gas line because it carries an insignificant amount of refrigerant, when compared with the liquid line, and considering a length of 5 meters, it gives a total 250 grams, what means a final weight of 2,6kg of R410A to be added.

4.8 Potential applications

The developed concept is mainly characterised by its modularity and therefore the possibility of separating the two main noise sources: the fan and the compressor.

Hence, it is possible to highlight different applications for this concept. Starting with the compressor module, it can be installed in different sites depending on the limitations of the user. Some possible places are listed below:

- Basement or Garage;
- Wall integrated;
- Buried;
- Back yard;
- Front yard;
- House Rooftop;
- Garage Rooftop.

The possibilities of installation for the fan module are in part equal to the compressor module with exception of the wall integrated and the buried applications. These possibilities are not possible due to the necessity of ensuring an exterior air flow through the evaporator.

The modularity associated to the developed concept plays an important role because it allows the user to manage the placement of the main noise sources. For instance, if there are complaints from the neighbors concerning noise, the user can place the compressor module inside the garage or the basement just leaving the fan outside.

On the other hand, if the noise is annoying the owner itself, it is possible to place both modules in distinct places and far away from the house for example, placing the compressor module beyond a tree in the back yard and the fan module on the rooftop of the garage.

Moreover, the separation of these 2 main components enables the application of optimized sound barriers to the compressor module since, when isolated from the fan and the adjacent evaporator, the compressor module may be enclosed inside a closed box.

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Chapter 5

Acoustic Enclosure

As presented previously in **Section 1.4**, the second objective of this master thesis work was the development of a second sound barrier to enclose the compressor module.

In this chapter, it is described the theoretical strategy adopted in order to build an efficient enclosure. A brainstorm about possible materials to be used in the enclosure is also presented.

Lastly, the final concept is described in terms of characteristics and assembly.

5.1 Theoretical strategy

In order to ensure an effective sound encapsulation of the compressor module, the development of the acoustic enclosure was based in three crucial acoustic principles:

- High acoustic impedance gradient between materials;
- Sound leakage insulation;
- Absence of structure-borne noise.

In sound propagation, acoustic impedance is very important. Engineers can control the wave propagation by selecting materials with the appropriate impedance. If the acoustical impedance remains unchanged, the wave will tend to continue unchanged. For instance, in ultrasonic testing, engineers can ensure that waves propagate well by choosing materials with similar impedance characteristics [35].

In contrast, engineers can also suppress sound by incorporating materials with significantly different values of impedance. As an example, when a wave traveling through a low impedance material (in the case of air) meets a high impedance material (in the case of a high density wall), the wave will significantly reflect back on itself, being transmitted through the wall only in a small part.

Acoustic impedance does not only depends on the material properties. It is also defined by geometrical characteristics, such as bends, cross sectional area, junctions, cavities, among others.

Therefore, it was decided to design an air chamber between the compressor module and the walls of the acoustic enclosure. In addition, the wall material must have a considerably high mass with the purpose of achieving a high impedance value at the boundary between the air chamber and the wall material. Consequently, to avoid large volume walls the focus was on choosing materials with high specific mass in order to obtain a high characteristic impedance (Z_o).

Regarding the sound leakage insulation principle, the presence of openings can severely compromise the performance of the enclosure regardless its efficiency in sound blocking. The presence of a small opening is sufficient to significantly reduce the insertion loss of the enclosure [47]. Hence, ensuring the sealing of the acoustic enclosure became a priority during its development phase.

Structure-borne noise can be described as sound waves which travel through solid state materials or which are radiated by solid state materials. It plays an role in noise propagation since the velocity of sound in solid state material is usually higher than in gases [59].

Therefore it becomes fundamental to separate the compressor module from the enclosure avoiding any physical contact between them. However, there will exist some contact between the water and refrigerant hoses and the walls of the enclosure as well as with the electrical cables. This can significantly reduce the performance of the enclosure, therefore it is important to introduce some damping and isolating material in those locations.

5.2 Materials

Considering the statements referred in the previous section, a survey of materials that present a considerably high specific mass was conducted. In addition, its behaviour regarding waterproof and workability were also considered.

Table 5.1 summarises these three characteristics for three different material categories: pressed wood, brittle and composites. Hence, thirteen different materials were considered as a starting point in the selection of the material to be used in the construction of the enclosure.

The enclosure must be waterproof to allow the placement of the module in the outside of the building. The workability of the material was also evaluated depending on the type of tools needed to work the material, three different levels were considered: easy, medium and hard.

The easy level is associated to the use of simple woodworking tools, such as: hand saws, chisels and wood drills among others. The medium level includes the same type of tools described for the easy level with the addition of some specific tools such as, the rabot and the Stanley knife. The hard level is related with brittle materials for which it is necessary to use diamond tips to cut them, such as diamond drills and diamond cutting disks.

Table 5.1: Overview of possible materials to construct the sound enclosure. Materials that present a considerably high specific mass with reference to its waterproof capability and ease of working the material.

Category	**Material**	ρ **[kg/m^3]**	**Waterproof**	**Workability**
Pressed Wood	MDF [48]	700	No	Easy
	MDF Water-repellent [48]	840	Yes	Easy
	OSB [49]	590	Yes	Easy
	Plywood [50]	670	No	Easy
	Platex (HDF) [51]	950	No	Easy
	Aglomerate [48]	550	No	Easy
Brittle materials	Slate [52]	2600	Yes	Hard
	Basalt [52]	2600	Yes	Hard
	Ceramic tiles [54]	2000	Yes	Hard
	Fibro cement cedral [55]	1300	Yes	Hard
	Fibro cement equitone [56]	1580	Yes	Hard
Composites	Perforated plasterboard [57]	800	No	Medium
	Wood plastic composite [58]	1000	Yes	Easy

The brittle materials presented the highest values of specific mass when compared with the other three categories; however, they are considered to be the hardest to work with. On the other hand the pressed wood category presented materials which are easier to work, although they present considerably lower values of specific mass and just two of them are waterproof.

Two different materials were presented in the composite categories. Both have a relatively high value of specific mass but only WPC is waterproof and is considered a easier material to work with since its behaviour is similar to pressed wood materials. Moreover, WPC presents a higher specific mass when compared to any pressed wood material.

5.3 Concept development

5.3.1 Choice of material

Taking into account the materials considered in the previous section and the literature review carried out in **Section 2.3** in the **State of the art Chapter**, about sound performance of different materials, it was decided to use WPC to build the enclosure around the compressor module.

Firstly, as presented in the literature review, WPC presented better values of STL when compared with the remaining studies. However, this correlation must not be assumed as absolutely correct due to the fact that not all studies used the same test procedure in the acquisition and calculation of the STL values. Moreover, the dimensions and the profile section of the samples were different from study to study.

Another reason that justify the choice of WPC was it simplicity since it is a one piece composite material obtained from extrusion processing while CLD (presented in the literature review) is more complex to obtain since they involve different materials which need to be connected together.

In addition, WPC is characterised by being a sustainable solution when compared with convectional composites [42]. Moreover, WPC performs well when exposed to weather conditions such as: rain and UV radiation [43, 44].

The final reason to choose WPC was the possibility of having a customisable appearance. WPC can be easily produced in different colors having different profile sections with different finishing patterns. **Figure 5.1** depicts the available colors and finishes produced by *TwinWood*

[45]. This company provided the necessary WPC for this study. The provided boards were made of shredded PVC scraps and rice husk.

Figure 5.1: Different colors and finishes of WPC boards by *TwinWood* [45].

5.3.2 Structural development

To be able to assemble the WPC boards together it was developed a structure using BOSCH structural profiles [46]. Two types of profiles were selected: 45X45L and 45X45R, illustrated in **Figure 5.2a** and **Figure 5.2b** respectively. Both types have a 10 mm slot and a cross-section width of 45 mm.

To be able to assemble the profiles together, a quick BOSCH connector was used, depicted in **Figure 5.2c**. An important feature of this type of connection is that it is only necessary a hexagonal key to fasten the connection.

(a) Structural profile 45X45L [46].

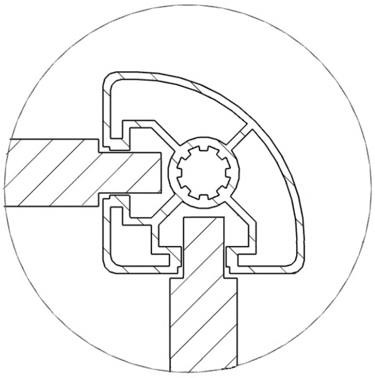
(b) Structural profile 45X45R [46].

(c) Quick connector, 8/10 mm slot, 0° [46].

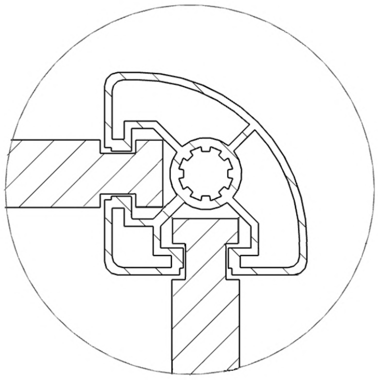
Figure 5.2: BOSCH components used as structural parts of the enclosure.

The purpose of using BOSCH profiles was to allow the WPC boards to easily slide within the slots of the profile. **Figure 5.3a** and **Figure 5.3b** depict two cross-section views of the assembly of the boards through the slot of the BOSCH profiles illustrating two different approaches.

This type of profiles were also chosen due to the fact that they are obtained through extrusion processing just as the WPC boards. Therefore, if moving forward to production the aluminium profiles can easily be replaced by the same WPC material used in the boards.



(a) Proposal 1.



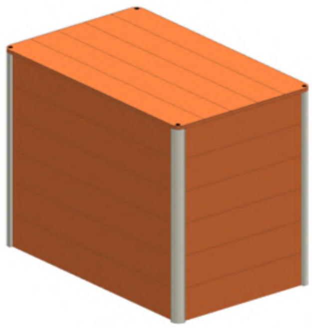
(b) Proposal 2.

Figure 5.3: Fitting solutions between WPC boards and BOSCH profiles.

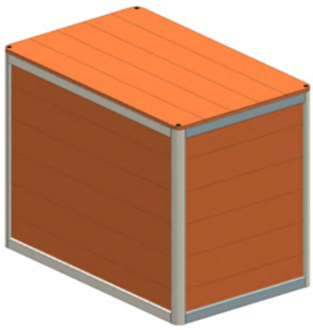
Figure 5.4 illustrates three different structural designs. The first solution is only possible if using the fitting solution, proposal number 2 (**Figure 5.3b**).

As is possible to visualise, the round profiles were used as vertical bars while the square profiles were used as horizontal bars. This structural design with curved vertical edges was chosen to be in line with the design kit used in the developed modules.

The three designs were discussed with BOSCH development engineers and the second proposal was chosen for development due to two reasons: more aesthetically appealing and greater ease in machining fitting detail number one (**Figure 5.3a**).



(a) Proposal 1.



(b) Proposal 2.

(c) Proposal 3.

Figure 5.4: WPC enclosure structural design proposals.

It was necessary to machine the WPC boards to be possible to assemble them within the BOSCH profile slots following the fitting solution 1. The corners of the top boards were also machined in order to obtain the rounded shapes.

5.3.3 Assembly and final solution

Figure 5.5a illustrates an exploded view of the final CAD concept. It is possible to visualise all the components as well as it assembly.

The modularity feature inherent to this concept is important insofar as it allows to the installer to transport the enclosure in smaller parts, assembling it in site.

Moreover, it becomes easier for the technician to access to the compressor module since it is not necessary to remove all the boards of the enclosure, but only those corresponding to the side to be accessed.

(a) 3D CAD design - exploded view.

(b) Final solution - prototype built.

Figure 5.5: 3D CAD design and final solution of the developed acoustic enclosure.

Figure 5.5b shows the final prototype built in the laboratory. The final dimensions are the following ones: width: 116 cm; depth: 73 cm; height: 84 cm.

In order to allow the water hoses to pass through the WPC boards, two holes were drilled at the joint of the lateral bottom WPC board with the ground BOSCH profile. In addition, to pass the refrigerant hoses and the electrical extensions, 3 more holes were drilled at the joint of the same WPC board with the upward one.

It was used a circular hole saw drill to produce the circular holes illustrated in **Figure 5.6a**. The final assembly of the hoses passing through the holes is shown in **Figure 5.6b**.

(a) Detail of the holes drilled in the WPC boards to allow the water and refrigerant hoses and the electrical extensions to pass through.

(b) Refrigerant (white) and water (black) hoses passing through the drilled holes in the WPC.

Figure 5.6: Drilled hoses to allow the passage of the holes and electrical cables.

Part III

Results, Discussion and User Experience

Chapter 6

Sound Measurements

This chapter begins with a description of the set-up and preparation of the modular heat pump in order to realise the sound measurements, along with the set-up of the semi-anechoic chamber and the microphones.

To evaluate the influence of different sound improvements implemented into the compressor module and the WPC enclosure, a set of 5 sound measurements was realised. Those improvements are described in detail in the second section of this chapter.

The last section of this chapter presents the acoustic measurements results obtained regarding sound power and tonality for different operating conditions. Moreover, it is also presented a comparison between the developed solution and the different full monobloc BOSCH units.

6.1 Set-up and preparation

Starting with the set-up and preparation, the compressor module was placed inside a semi-anechoic chamber, while the fan module was placed outside of it.

Subsequently, the refrigerant and the water lines were installed. It was also required to connect the modules to the power grid, as well as, to establish the communication between them and with the computer.

Finally, inside the semi-anechoic chamber, several microphones were properly placed for the sound measurements.

All these procedures are described in the next subsections.

6.1.1 Refrigerant circuit set-up

It was necessary to connect the evaporator placed inside the fan module with the refrigerant circuit of the compressor module. To accomplish this connection, a pair of flexible hoses were used as described in **Section 4.6**, **Chapter 4**,

After the installation of the refrigerant circuit, it was filled with an inert gas (nitrogen) to detect any possible leakage in the circuit. With nitrogen loaded up to a pressure of 15 bar and after a period of 30 minutes, it was noticed that the pressure did not dropped, meaning that the circuit was sealed.

To fill the circuit with the refrigerant, one important step to be carried out is the evacuation of any particles inside this circuit. To accomplish this task, a vacuum pump was used until the pressure of the circuit reached values lower than 0,3 mBar. After this, the circuit was ready to be charged with refrigerant.

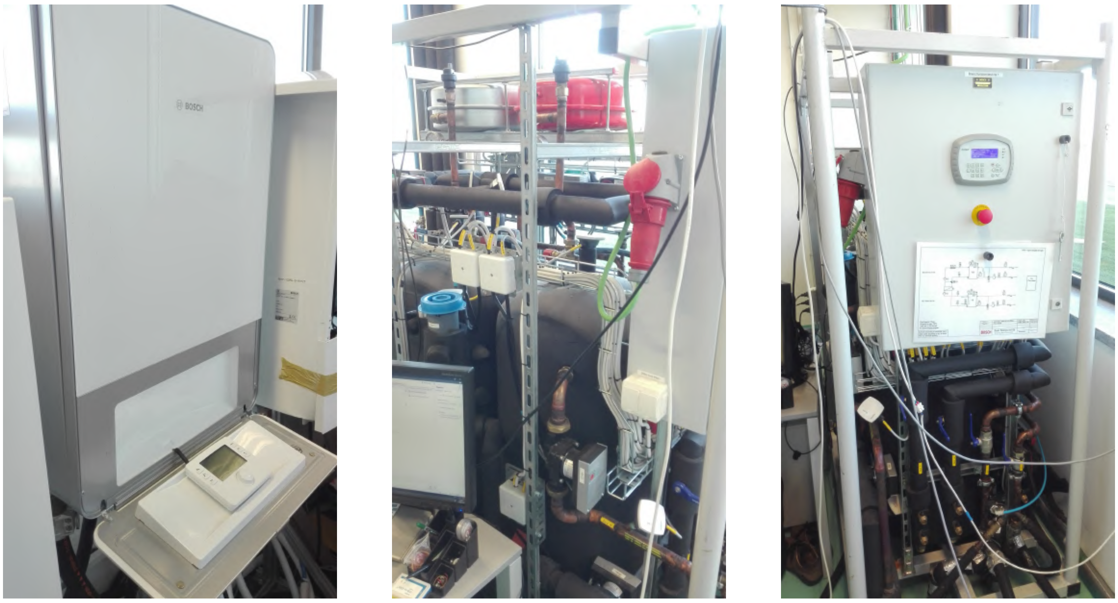
The refrigerant circuit was charged with 2,6kg of R410A accordingly to the statement presented in **Subsection 4.7.2**, **Chapter 4**.

6.1.2 Water circuit set-up

The heat produced by the heat pump is transferred to the water circuit through the plate heat exchanger placed in the compressor module. A pair of water hoses were connected to this plate heat exchanger placed in the CM and to an indoor unit.

The indoor unit (IDU) used, represented in **Figure 6.1a**, has a water pump which controls the flow of the water that goes into the central heating system of the house. These two units, the IDU and the ODU, are responsible for the correct control of the temperature of the water that is supplied to the central heating system.

To simulate the central heating system, it was used a convectional test rig, depicted in **Figure 6.1b** and **Figure 6.1c**. The test rig delivers heat to the building chiller in order to maintain the preset temperature.



(a) Indoor unit (IDU). (b) Test rig - lateral view. (c) Test rig - Control panel.

Figure 6.1: Indoor unit and test rig used to run the heat pump and perform the sound measurements.

6.1.3 Communication connection

It is crucial that the IDU and the ODU communicate between themselves to be able to operate properly with the maximum efficiency. To accomplish this it is mandatory to connect the HP board of the compressor module with the controller board of the IDU.

The communication between both is established through a wired serial communication using an electrical wire. The serial communication is CAN which stands for Controller Area Network.

To be able to control the heat pump, a computer software was used. **Figure 6.2** depicts the interface of the software. Using this software, it becomes possible to set different compressor and fan velocities, as well as changing the heat pump operating mode: heating, cooling or defrost.

In addition, the software allowed to monitor multiple variables of the heat pump, such as, the temperatures and pressures of different parts of the water and refrigerant circuit including the compressor chassis temperature.

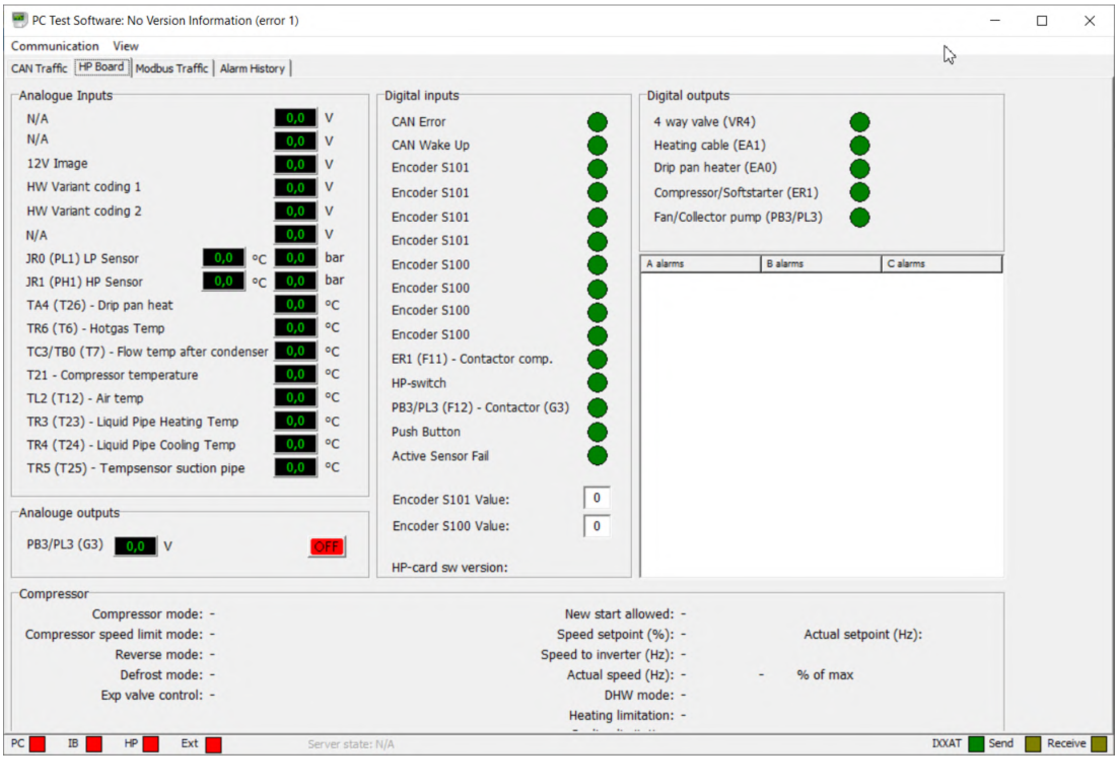


Figure 6.2: Computer software interface.

6.1.4 Sound measurements set-up

Aiming the determination of the sound power produced by the prototype, with the different improvements implemented, it was necessary to follow a standard methodology.

This is important since it guarantees a common measurement performed by different entities for multiple appliances, allowing a more credible comparison between products.

ISO 3744 was the used standard which is defined as: *"Determination of sound power levels and sound energy levels of noise sources using sound pressure — Engineering methods for an essentially free field over a reflecting plane"* [60].

Following this standard, 9 microphones were positioned at a distance of half a meter in a cubic envelop. As standardised, the measurements must be performed in a free field over a reflecting plane. Therefore the compressor module was placed in a semi-anechoic acoustic chamber, which is composed of 5 sound absorbing faces and a reflecting face (the floor). **Figure 6.3** shows the set-up of the 9 microphones in the semi-anechoic chamber.

Figure 6.3: Sound measurement set-up accordingly to ISO 3744.

6.2 Sound measurement procedure

To be able to evaluate the performance of the WPC enclosure and the effectiveness of certain sound insulation techniques, 5 sound measurements were realised.

The first measurement was performed onto the compressor module solely, without implementing any sound improvement, as described in **Chapter 4**. This analysis served as the baseline for the successive measurements. Afterwards, a set of sound improvements were implemented to the module and a second measurement was performed to it.

The third measurement was made with the WPC enclosure around the compressor module. Subsequently, some improvements were implemented into the enclosure and a new measurement was made. Finally, some foam was added between the module and the WPC enclosure and a fifth sound analysis was realised.

The next subsections describe, in detail, those 4 sets of improvements implemented to the compressor module, which correspond from the second to the fifth sound measurement configurations.

6.2.1 Compressor module with leakage treatment

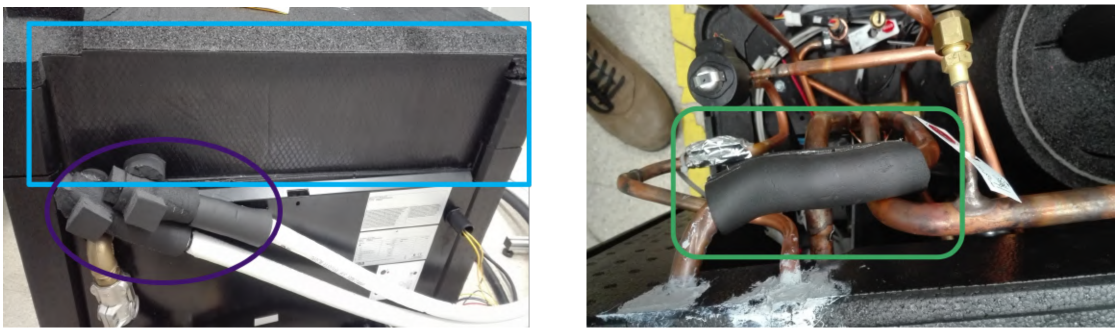
When listening to the compressor module whilst running it was possible to notice two major annoying noises: the compressor noise and the noise radiated by the circulation of refrigerant through the shut-off valves. The compressor was more audible than the valves above a compressor speed correspondent to a frequency of 220Hz. Otherwise the predominated noise came from the valves.

This inference is not precise due to the fact that the human hearing level varies from person to person, as well as the definition of noise. However, the microphone that presented the highest values of sound pressure was placed on the rear part of the module facing the shut-off valves.

Therefore, a bitumen layer was glued on the top rear EPP part depicted in blue on **Figure 6.4a** to block the compressor noise. Bitumen has a density of approximately 1000 kg/m^3 [61] allowing the addition of a significant amount of mass to the system, with a reduced volume of material.

To absorb the noise radiated by the valves, foam with a thickness of 15 mm was glued around them and around a small portion of copper piping immediately after it, as illustrated in **Figure 6.4a** marked in purple.

A portion of the gas line inside the compressor module was vibrating considerably more than the other parts of the refrigerant circuit. Consequently, a foam with 10 mm of thickness was glued around it, as illustrated in **Figure 6.4b**. This foam was also in contact with other fixed pipes of the circuit what allowed the damping effect of the foam.



(a) In blue - addition of a bitumen layer on the top rear EPP part of the compressor module; In purple - addition of foam around the shut-off valves.

(b) Addition of foam around the gas line inside the compressor module.

Figure 6.4: Addition of foam and bitumen.

Figure 6.5a shows the addition of a foam strip which guarantees tightness between the EPP lid and the remaining EPP structure of the module. Furthermore, a sealing glue was used to fill the gap created by the holes made in the EPP structure, through which the copper pipes run, as illustrated in **Figure 6.5b**.

(a) Implementation of a foam strip between the EPP lid and the remaining EPP structure of the module.

(b) Sealing of the EPP holes through which the refrigerant lines run.

Figure 6.5: Addition of foam strip and sealing glue.

6.2.2 Installation of the WPC enclosure

The third sound analysis was performed with the installation of the WPC enclosure around the compressor module without any improvement implemented to the enclosure. **Figure 6.6a** illustrates the set-up of the WPC.

Multiple leaks were found in the enclosure which compromise its performance in sound insulation. In **Figure 6.6b**, marked with a red arrow, is possible to see the gaps between the water hoses and the drilled holes on the WPC.

(a) WPC acoustic enclosure installation around the compressor module.

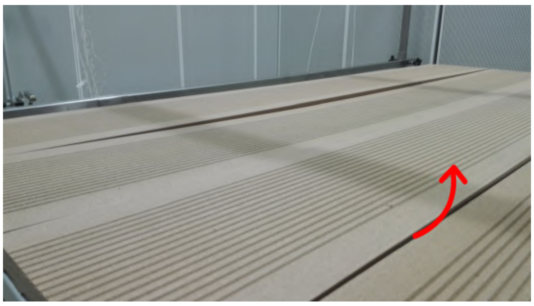
(b) Illustration of the sound leakage through the clearance around the water and refrigerant hoses.

Figure 6.6: Acoustic enclosure installation and illustration of sound leakages.

Figure 6.7a also depicts some sound leakages marked in red. The WPC boards present openings in the lateral side due to the fact that the standard boards used, were produced with hollow sections. However, it is possible to do completely solid boards in future designs since a WPC mold for extrusion is not a complex tool to be produced.

Another detected point of leakage were the gaps between the top WPC boards as illustrated in **Figure 6.7b**. Although the gaps were not significantly wide, there are four long gaps between the boards.

(a) Sound leakage through the lateral openings of the WPC boards.



(b) Sound leakage between WPC boards on the top face.

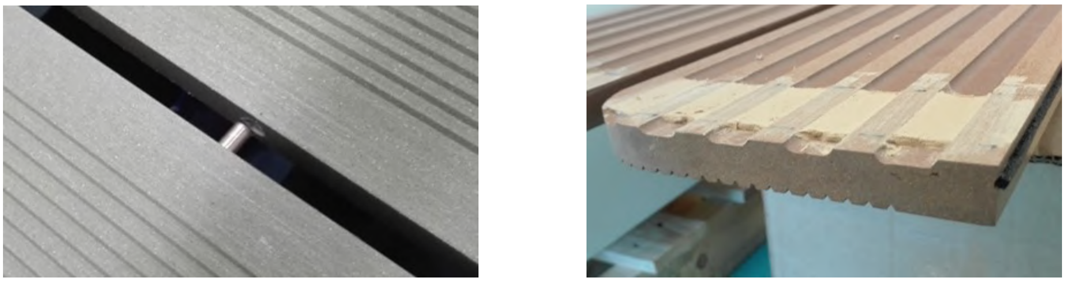
Figure 6.7: Acoustic enclosure sound leakages.

6.2.3 WPC Enclosure with leakage treatment

Since sound waves propagate through the movement of air particles as a fluid, as stated in **Section 5.1**, **Chapter 5**, it is crucial to avoid sound leakage in order to guarantee good sound isolation. To accomplish this, the leakages presented in the last subsection were sealed.

Some pins were installed between the top WPC boards to keep them leveled between each other as illustrated in **Figure 6.8a**. Moreover a foam strip were glued in the joint of the top WPC boards to seal it. **Figure 6.9b** depicts the glued strip.

In addition, to fill the lateral gaps of the top boards, a wood filler was used as illustrated in **Figure 6.8b**.



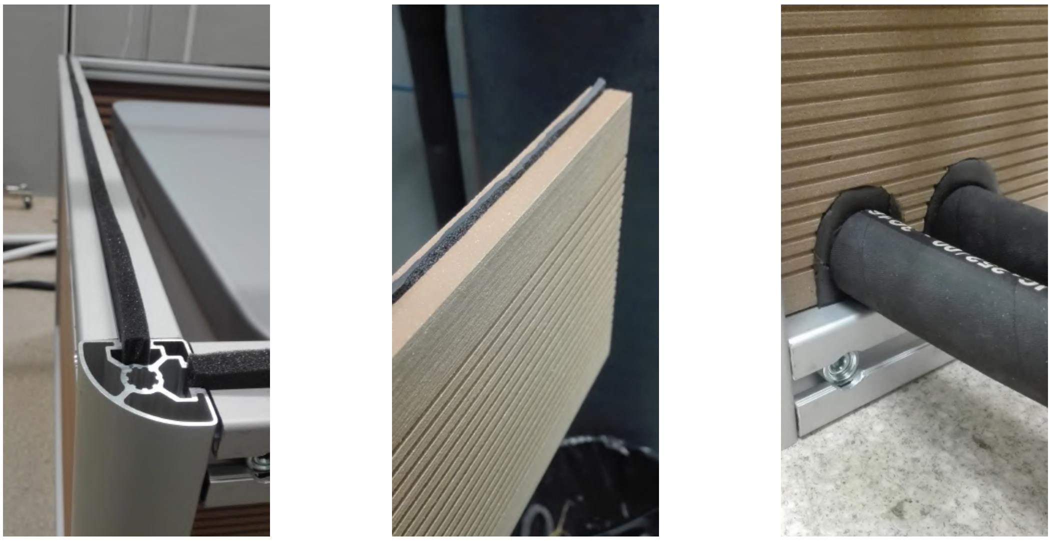
(a) Installation of pins between the top boards.

(b) Filling of the lateral openings of the WPC boards.

Figure 6.8: Installation of pins and openings filling.

Some foam strips were also placed inside the slots of the aluminium profiles, as presented in **Figure 6.9a**, to seal the junction between the top WPC boards and the profiles.

To fill the gaps around the water houses, a rod foam was used as illustrated in **Figure 6.9c**.



(a) Insertion of foam on the top face of the structural profiles.

(b) Insertion of a foam strip between the top WPC boards.

(c) Insertion of foam around the water hoses.

Figure 6.9: Addition of foam regarding sound leakage treatment.

6.2.4 WPC enclosure with leakage treatment and addition of foam

The last measurement to be performed was with the addition of foam to the sides that radiate more noise. Observing the values of sound pressure, the most significant sides radiating noise were the rear and the top faces of the enclosure.

Therefore, foam was installed between the WPC and the compressor module as depicted in **Figure 6.10a** and **Figure 6.10b**. A grey mineral wool was used in order to absorb the acoustic waves and decrease reverberation inside the enclosure.

(a) WPC enclosure with foam on the back face.

(b) WPC enclosure with foam on the top and back face.

Figure 6.10: Installation of foam on two faces of the enclosure.

6.3 Results and discussion

The sound power levels were determined following the standard ISO 3744 [60] for the five measurements described above. In addition, it was also calculated a tonality penalty for the different measurements. Different operating conditions were tested varying the compressor and fan speeds.

There are two main operating conditions common to all heat pumps that are tested: the night mode and the max day mode. The night mode is defined as the quietest operating mode while the max day as the loudest one, that the heat pump can achieve. Therefore, the night mode normally corresponds to the lowest operating speed of the compressor allowed by the heat pump, without compromising significantly the heating output, whereas the max day is generally defined as the compressor running at the highest possible speed.

The heating conditions are also standardised, which are defined by two temperatures: the ambient air temperature and the output water temperature. The first one corresponds to the temperature of the air which flows through the evaporator in the ODU. The second are the temperature of the water delivered by the heat pump, to the heating system of the house.

For the tests performed, two sets of heating conditions were used: A7/W55 and A7/W35. The number right after the letter "A" corresponds to the ambient air temperature while the number after the letter "W" to the output water temperature.

6.3.1 Sound power

In the **Table 6.1** is possible to observe the values of sound power level, A-weighted (LwA), within the frequency spectrum from 100Hz to 10kHz, for different operating conditions (columns) corresponding to the different sound measurements configurations (rows).

The operating conditions are discriminated in the column headers, where the heating conditions appear as A7W55 or A7W35. The number following the letter "C" corresponds to the frequency (Hertz) modulated by the inverter which determines the compressor speed. The number that follows the letter "F" corresponds to the fan speed percentage, from 0% to 100%.

The column header in grey represents the night mode with a compressor speed of 260Hz and the fan running at 50% of its capacity. This point was measured at A7/W35 heating conditions. The max day mode is presented in the last column (light green), with the compressor running at 360Hz and the fan at 50% with A7/W35.

The first row of values corresponds to the measurements performed to the compressor module with no sound improvements (designated as **BASELINE**) while the second row corresponds to the values of the compressor module with the implemented improvements (designated as **CM 1ST**) described in **Section 6.2.1** of this **Chapter**.

The third row of the table corresponds to the measurements performed with the WPC enclosure without improvements (designated as **WPC**). The fourth row describes the enclosure with sound leakage treatment (designated as **WPC 1ST**) and the fifth row describes the WPC enclosure with sound leakage treatment and the addition of foam (designated as **WPC 2ND**). Accordingly to the improvements described in the **sections 6.2.2 - 6.2.4**, respectively.

The values are presented using a color map where the red tone is associated to higher values of SPL while the green to lower values, with yellow being associated to intermediate values.

Table 6.1: Values of sound power level A-weighted (LwA) 100Hz-10kHz.

	OPERATING CONDITIONS													
CONFIGURATION	A7W55_C90_F50	A7W35_C123_F50	A7W35_C156_F50	A7W35_C180_F50	A7W35_C198_F50	A7W35_C220_F50	A7W35_C240_F50	A7W35_C240_F60	A7W35_C260_F50	A7W35_C280_F60	A7W35_C298_F50	A7W35_C320_F50	A7W35_C340_F50	A7W35_C360_F50
BASELINE	49,1	47,0	49,2	49,3	51,7	51,0	52,8	52,9	52,2	56,6	53,1	53,5	55,1	54,6
CM 1ST	47,8	45,3	47,3	48,1	50,8	49,8	51,9	52,0	50,8	53,1	53,0	54,6	54,7	53,3
WPC	35,8	38,1	37,8	35,6	42,1	39,9	40,3	40,2	43,1	42,1	42,4	45,5	47,2	42,2
WPC 1ST	28,6	30,8	31,0	31,4	36,4	33,3	34,1	34,1	38,3	37,4	37,8	40,4	40,0	37,5
WPC 2ND	28,3	32,8	30,9	31,4	33,6	32,3	31,3	31,4	35,2	37,2	38,2	39,2	38,8	35,8

For a better understanding, a chart representation of the measured sound power values is depicted in **Figure 6.11** using the same color scheme.

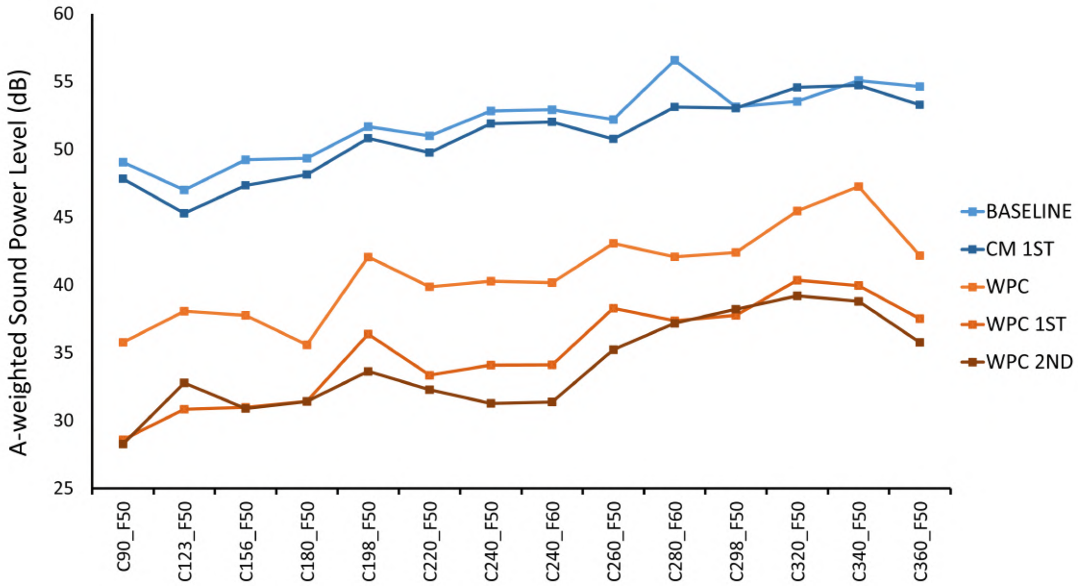


Figure 6.11: Graphical representation of sound power level A-weighted (LwA) 100Hz-10kHz.

Comparing the blue lines it is possible to identify a slight decrease of the sound power level radiated by the compressor module with the implementation of the sound improvements. From a compressor speed of 90Hz up to 260Hz there was a decrease around 1dB and 2dB. At 280Hz occurred the most significant decrease, approximately 3dB, and at higher speeds the decrease was not significant, in fact there was an increase of 1,1dB for 320Hz.

Regarding the usage of the WPC enclosure encapsulating the module, a significant reduction is evident. The lighter orange line of the chart shows an average reduction of 10dB when compared with the compressor module improved. The highest value of sound power using the enclosure occurs for a speed of 340Hz with 47,2dB and the lowest at a speed of 180Hz with 35,6dB. The average value encompassing all operating conditions is 40,9 dB.

With the implementation of the modifications proposed in **Section 6.2.3** the values presented as **WPC 1ST** were obtained. It is perceptible a huge decrease in the sound power radiated when compared with the previous results of the WPC enclosure without any improvement. An average drop of 5,8dB can be noticed and the minimum value is only 28,6dB for a compressor speed of 90Hz while the maximum value is 40,4dB for 320Hz.

The addition of foam between the compressor module and the enclosure, described in **Section 6.2.4** is presented by the brown line. Comparing with the previous measurement, without foam, the decrease is not significant having an average reduction of 1dB. In fact, at some points it occurred an increase of the sound power: for a velocity of 123Hz and 298Hz.

In **Table 6.2**, the lowest, the highest and the average values of sound power of each measurement performed are presented. Moreover, the difference of the average values between two successive tests is shown in the last column.

Table 6.2: Minimum, average, maximum and average difference values of sound power level A-weighted (LwA) 100Hz-10kHz encompassing all the operating conditions.

	OPERATING CONDITIONS			
CONFIGURATION	MINIMUN	AVERAGE	MAXIMUM	AVERAGE DIFFERENCE
BASELINE	47,0	52,0	56,6	
CM 1ST	45,3	50,9	54,7	1,1
WPC	35,6	40,9	47,2	10,0
WPC 1ST	28,6	35,1	40,4	5,8
WPC 2ND	28,3	34,0	39,2	1,0

To conclude, the best result was obtained with the use of the enclosure around the compressor module. In addition, insulating the enclosure properly proved to be important since it reduced the sound power considerably.

On the other hand, the improvements implemented in the compressor module did not showed a significant reduction as well as the usage of foam between the enclosure and the module.

6.3.2 Tonality

Mathematically, tonality occurs when the level of sound pressure of a specific frequency is significantly higher than the sound pressure values of the neighboring frequencies. In terms of human perception, tonality occurs when a frequency stands out from the other frequency and is easily recognised by the human ear.

Therefore, even if a sound has lower sound power but presents tonality, it will be perceived as more annoying than a sound with a higher sound power but without tonality. Consequently, characterising an equipment only in terms of sound power level is not enough to predict the level of annoyance [62].

Tonality is a matter of psychoacoustic study that depends on different factors which not all are quantitative but also qualitative. As an attempt to quantify the tonality impact in noise radiation it is a common practise to calculate a tonality penalty. This penalty consists of the addition of an extra value to the sound power level calculated.

The standard ISO 3744 does not contemplate tonality correction. In addition, the established legislation in Portugal does not consider it in the calculation of the noise emission level of an equipment. However, there are several countries which take it into account. BOSCH company considers this parameter. Therefore, in the sound analysis performed in this study, it was calculated a tonality penalty accordingly to the standard DIN 47315 (German Institute for Standardisation). This standard defines 3 levels of tonality penalty (0dB, 3dB, 6dB) which defines if there is tonality and how much distinguished it is.

The tonality penalties calculated for the different measurements are presented in **Table 6.3**.

Table 6.3: Values of tonality penalty, 100Hz-10kHz.

	OPERATING CONDITIONS													
CONFIGURATION	A7W55_C90_F50	A7W35_C123_F50	A7W35_C156_F50	A7W35_C180_F50	A7W35_C198_F50	A7W35_C220_F50	A7W35_C240_F50	A7W35_C240_F60	A7W35_C260_F50	A7W35_C280_F60	A7W35_C298_F50	A7W35_C320_F50	A7W35_C340_F50	A7W35_C360_F50
BASELINE	0,0	0,0	0,0	0,0	3,0	0,0	3,0	3,0	3,0	6,0	3,0	0,0	0,0	3,0
CM 1ST	0,0	0,0	0,0	3,0	3,0	0,0	3,0	3,0	3,0	3,0	6,0	6,0	3,0	3,0
WPC	0,0	3,0	0,0	3,0	3,0	3,0	6,0	6,0	0,0	3,0	3,0	3,0	3,0	3,0
WPC 1ST	0,0	0,0	0,0	0,0	0,0	0,0	6,0	6,0	0,0	6,0	0,0	0,0	0,0	0,0
WPC 2ND	0,0	3,0	3,0	0,0	0,0	0,0	6,0	6,0	0,0	6,0	3,0	3,0	3,0	3,0

The **BASELINE**, **CM 1ST** and the **WPC** were the configurations that presented more operating conditions with tonality penalties. More than half of the measured points presented tonality, with emphasis to the compressor speed associated to 280Hz in the **BASELINE** configuration which presented 6dB of tonality as well as the 280Hz and 320Hz for the **CM 1ST**. The 240Hz for the **WPC** configuration suffered also an increase of 6dB due to tonality.

On the other hand, the **WPC 1ST** configuration presented less tonality problems, not presenting any 3dB penalty. However for compressor velocities of 240Hz and 280Hz the maximum penalty was applied. With the implementation of the foam between the enclosure and the compressor module (**WPC 2ND**), 3dB tonality penalties appeared for the lower and higher compressor frequencies.

In general, it is possible to conclude that the developed prototype presents some tonalities problems mainly in the compressor module and with the implementation of the WPC enclosure without the implemented improvements.

6.3.3 Models comparison

To compare the developed prototype to the BOSCH monobloc units in terms of sound performance, a comparative analysis was performed.

The prototype was compared with three different models depending on the heating output capacity: a 4kW, a 7kW and a 9kW units. Normally a lower heating output means that the unit is equipped with a smaller compressor. A smaller compressor presents lower values of sound power, therefore it is expected that heat pumps with a lower heating output value also have lower sound power levels.

The models were compared with the sound measurements configurations which presented better results. These are: the compressor module with improvements (**CM 1ST**) and the inclusion of the WPC enclosure with the sound improvements and the addition of foam (**WPC 2ND**).

Table 6.4 presents the sound power levels for the different models, A-weighted within a frequency range of 100Hz to 10kHz. The operating conditions are again presented in the column headers while the model, identification is represented in the rows.

In previous tests performed on the 4kW, 7kW and 9kW units, it was not covered the entire range of operating conditions as it was in the developed prototype. Therefore, only 5 operating points are presented. For instance, for the 4kW and 7kW units, the max day point corresponds to a compressor speed of 320Hz since it is the highest velocity that these units compressor can run.

The prototype was manufactured using components of a 9kW unit, therefore it can be directly compared with that model.

Table 6.4: Values of sound power level A-weighted (LwA) 100Hz-10kHz.

	OPERATING CONDITIONS				
CONFIGURATION	A7W55_C90_F50	A7W35_C156_F50	A7W35_C260_F50	A7W35_C320_F50	A7W35_C360_F50
AirX 90 - 9kW	50,1		57,5		64,0
AirX 70- 7kW		51,2	54,0	53,1	
AirX 40- 4kW		47,9	54,2	52,3	
Compressor Module	47,8	47,3	50,8	54,6	53,3
WPC Enclosure	28,6	31,0	38,3	40,4	37,5

For a better understanding and analysis, a chart representation of the sound power values is presented in **Figure 6.12** with the same color scheme.

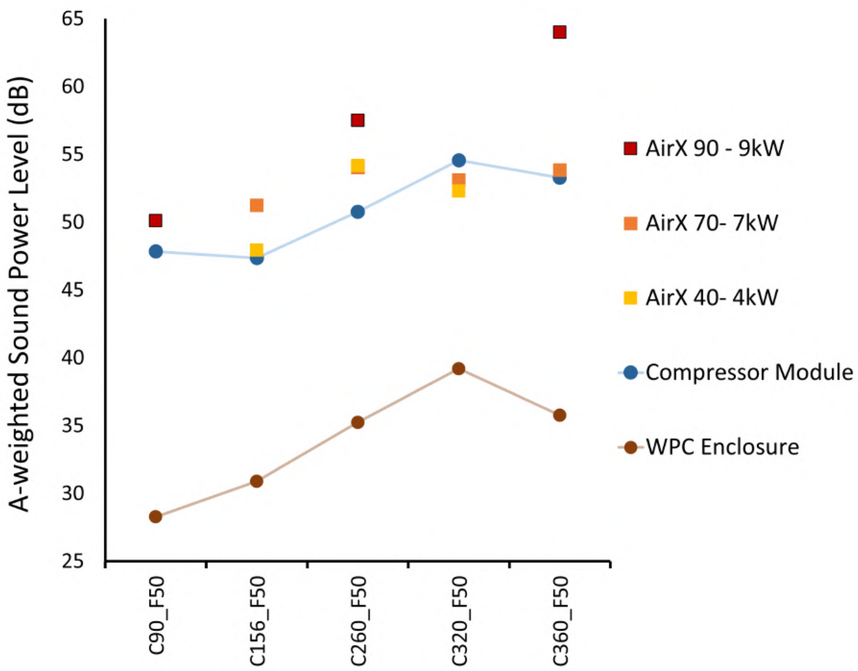


Figure 6.12: Values of sound power level A-weighted (LwA) 100Hz-10kHz.

When analysing the chart depicted in **Figure 6.12** and the correspondent data presented in the **Table 6.4**, a significant decrease is evident between the AirX 90 - 9kW and the compressor module. The most significant operating point is the max day (C360 F50), which suffered a decrease of 10,7dB, followed by the night mode point (C260 F50), which reduced 6,7dB. For a compressor speed of 90Hz the reduction was less significant, 2,3dB.

When comparing this unit (9kW) with the compressor module encapsulated with the WPC enclosure, the reduction is considerably evident. This is justified due to two factors: the reduction induced by the absence of the fan and the sound blocking effect provided by the WPC enclosure.

Another important observation is that the sound levels of the compressor module are close to the values of the 4kW and 7kW units. Furthermore, for all measured operating conditions, the compressor module presented lower values with exception for a compressor speed of 320Hz. This is a positive result, considering that those units have a smaller compressor. For instance, the 4kW has a compressor with approximately half of the power of the one used in the compressor module.

Lastly, when comparing the values of the acoustic enclosure with the 4kW and 7kW units, it is also possible to observe a huge decrease.

Chapter 7

User Experience

As stated in **Section 1.4** it was performed a user experience in order to gather feedback from different people about the proof of concept developed.

In this chapter, the concept of user experience (UX) is briefly introduced along with the objectives to be achieved with it. Afterwards, a description of the participants is presented followed by the structure of the UX sessions. The last section synthesises the outcome provided by the UX.

7.1 User experience and objectives

New products must be customer-oriented in order to be successful for a considerably long period of time. User experience is considered a crucial tool in order to obtain customer-oriented products [64].

User experience is everything that a user perceives when subjected to an experience. The experience can be a complete different subject depending on what is intended to be evaluated. It can be a service, a physical product, a digital product, among others.

A proper user experience has the purpose of transforming the user satisfaction into enthusiasm and, consequently, provide valuable feedback about the potentialities and weakness of the evaluated subject. Submitting the user to experience a particular product offers qualitative and/or quantitative information about the usage behavior such as: reasons to use the product, personal preferences, purchasing desire and knowledge about product [63].

In summary, user experience can be described as a collection of techniques and practices that involves users to test ideas, concepts or products.

In order to design an appropriate user experience it becomes important to define what is intended to know about the product from the participants perspective. With this, the performed UX aimed to research about the following points about the prototype developed:

- Market chances;
- Monetary added value;
- Preferred places to install the modules;
- Positive and negative aspects;
- Improvement suggestions.

7.2 Participants

To perform a good user experience it is important to take into account the profile of the participants. Two groups of participants were considered.

The first was composed by installers, technicians and development engineers with qualifications in the structural, acoustic, heating performance and maintenance areas. All the members of this group were BOSCH personnel, performing a total of eight participants.

The second group of participants were constituted by two university professors from the university of Aveiro with a background on mechanical vibrations, acoustics and heating performance along with three senior BOSCH engineers proficient in heat pump structural project and in the European heat pump sales market.

7.3 Structure

The user experience was divided into two parts accordingly to the participants group. The morning session (part 1), which started at 9:00am and finished at 12:00am, was attended by the first group of participants. The afternoon session (part 2). which started at 13:00pm, and finished at 15:30pm was attended by the second group of participants.

Both sessions had a similar structure which can be divided into three major blocks. The first one included the welcome, some opening remarks and a presentation of the work developed, explaining the development process and the characteristics of the prototype.

The second block was a practical session, where the participants went to the acoustic chamber and had the possibility of observing, appraise and listening to the prototype. Different operating modes were shown varying the compressor and fan speeds with and without the WPC enclosure around the module.

During the final block of the morning session, the participants returned to the UX room where the sound results were presented along with a market overview. In addition, a group discussion was performed and the participants filled in a questionnaire, which is depicted in **Figure A.1**, **Appendix A**.

Figure 7.1 presents the detailed agenda of the morning session presenting the topic, time, subject, location and the responsible person for each moment.

<table>
<tr><th>TOPIC</th><th>TIME</th><th>SUBJECT</th><th>WHERE</th><th>MODERATOR</th><th></th></tr>
<tr><td>Welcome</td><td>09:00
09:10</td><td>Welcome & opening remarks</td><td>UX room</td><td>Idalina & Jürgen</td><td>10min</td></tr>
<tr><td>Introduction</td><td>09:10
09:30</td><td>Presentation I | Modular HP – The foundation
- Scope
- Achievements
- Challenges</td><td>UX room</td><td>Lucas</td><td>20min
10 min brea</td></tr>
<tr><td>Practical session</td><td>09:45
10:45</td><td>Practical session:
- Hands-on & listening of HP</td><td>Acoustic lab</td><td>Participants</td><td>1h00min
10min break</td></tr>
<tr><td>Group discussion</td><td>11:00
11:15
12:00</td><td>Presentation II | Modular HP – The future
- Concept comparison
- Market overview
Group discussion
- Installation effort
- Maintenance & serviceability issues
- Acceptance of concept</td><td>UX room</td><td>Lucas
All</td><td>1h00min</td></tr>
<tr><td>Thank you!</td><td>12:00</td><td>End of part I</td><td></td><td></td><td>Lunch break
☺</td></tr>
</table>

Figure 7.1: User experience morning agenda - part 1.

For the afternoon session (**Figure 7.2**), the final block was slightly different. Instead of a group discussion, a group interview was performed in pairs where the participants were questioned with the same questions proposed in the morning group discussion. Following the interviews, the questionnaire was also delivered to the participants.

TOPIC	TIME	SUBJECT	WHERE	MODERATOR	
Welcome	**13:00**	**Welcome & opening remarks**	UX room	Idalina & Jürgen	
	13:10				10min
Introduction	**13:10**	**Presentation II | Modular HP – Present & future** - Scope - Achievements - Concept comparison - Market overview	UX room	Lucas	
	13:30				20min
Breakout rooms	**13:45**	R-1 Interaction with modular HP	Acoustic lab	G1	G1 starts
	14:15	R-1 Interaction with modular HP	Acoustic lab	G2	G2 starts
	14:15	R-2 1:1 feedback	UX room	G1	
	14:45	R-2 1:1 feedback	UX room	G2	G1 ends
					G2 ends
	15:15				1h00min
Thank you!	**15:30**	End of part II			Coffee break ☺

Figure 7.2: User experience afternoon agenda - part 2.

A A0 poster was affixed in the room of the UX and in the acoustic chamber which summarised the main characteristics of the prototype, as well as the sound results obtained. The poster is shown in **Figure A.2**, **Appendix A**.

7.4 UX outcome

The participants adopted a very active attitude during the experience. This was crucial to have discerned results from the questionnaire, as well as a fruitful outcome from the group discussion and interviews.

The outcome of the user experience can be divided into two parts: the answers to the questionnaire and the outcome gathered from the group discussion and interviews.

7.4.1 Questionnaire answers

The questionnaire delivered to the participants has objective questions with specific possibilities of answer. It was prepared to be answered from an end-user point of view, therefore it was asked to the participants to interpret the questions as if they were buying the product, as a final client.

In the first question of the questionnaire, which was about classifying the market chances of the developed product from 1 to 5, the most voted answer was the level 4 (Good chances - "It is a good idea") with 6 votes. In addition, the levels 3 and 5 received 3 votes each. Consequently, from the point of view of the participants, it is possible to assume, with a high degree of confidence, that the product would have good chances on the market since there was no negative responses (level 1 and 2) to this question and the most voted was the level 4.

Regarding the second and third questions, about the added value of the modularity feature and the inclusion of the WPC enclosure, the answers presented a lot of scatter, ranging from 0€ to 2000€ for the first question and from 100€ to 1000€ referring to the WPC enclosure. Taking into account the discrepancy of answers given to these questions it is not possible to assume a mean value for the added value of the modularity feature and for the WPC enclosure. Therefore, it is only possible to conclude, that the monetary value for this product depends substantially from person to person.

The fourth and the fifth questions were about the installation location of both modules. For the compressor module, the preferred place is the basement/garage. For the fan module, the most voted place was the backyard. This is valuable information in order to adapt the product to the environment conditions where it will be more often installed.

The last question tries to understand if the participant would buy this product. The most answered possibility was "yes" with a total of 9 while the "no" answer was given by only 2 participants. There was a participant which answered both, "yes" and "no"; this answer was not considered. Herewith, it is possible to assume that the most part of the participants were attracted by the product and consider it useful, reinforcing the good market chances for the product since it is in consonance with the answers given for the first question.

All the answers given by the participants can be found in detail in **Figure A.3**, **Appendix A**.

7.4.2 Group discussion remarks

For the morning session and the afternoon interviews, 4 questions were prepared. As opposed to the questionnaire, these questions were of open answering and more subjective with the intention of instigate and guide the group discussion.

In addition, they were designed to be interpreted from the product development side. The participants were compelled to argue accordingly to their background as engineers, teachers, technicians or salesman.

The proposed questions were the following:

1. What are the strongest qualities do you see in this concept?
2. Which barriers do you see in this concept?
3. Which significant improvements do you suggest for this concept?
4. What other uses do you foresee?

Considering the discussion about the first and second questions it was possible to highlight some positive and negative remarks associated to the developed prototype. These are gathered in **Table 7.1**.

Table 7.1: Positive and negative remarks subsequent from the UX group discussion and interviews.

Positive remarks	Negative remarks
• Significant sound reduction • Customization of the WPC panels • Good for houses with lack of space • Sustainable material (WPC) • Makes maintenance easier • Possibility of replacing one module instead of the entire unit • Easier to transport (smaller and lighter parts)	• Not suitable for propane • Copper pipes extension needed • More parts to install • Refrigerant technician required • Solution suitable only for a market niche • Bigger footprint

When approaching the third and fourth questions, different ideas and opinions emerged regarding improvements to be done to the prototype and different uses foreseen. Therefore the most significant suggestions are summarised in the following list:

- Integrate Solar Panel on the top of the compressor module;
- Replace metal panels of the compressor module with WPC boards;
- Integrated also WPC in the fan module;
- Integrate the WPC feature in a monobloc unit;
- Use the same fan module with different compressor modules (different heating outputs) to extend product portfolio;
- Customisation: create a story around the different accessories of the product;
- Exclude the IDU and replace it by this compressor module including the IDU components in the module.

In addition to the outcome referred above, a few observations emerged that were not part of the UX focus but which are also valuable information.

The first one is about the market to bet on, which by consensus was elected the HPP (High Price Products) market which involves countries in the northern part of Europe that are subject to more extreme weather conditions.

The second observation is about the way to sell the product. It was suggested that the product must be exposed as a high tech product and customisable accordingly to the user expectations. In addition, the exposure of the WPC enclosure to the market must be done with prudence since it is easily liable to replica, considering the way it is developed. Therefore it must be found a way to integrate the enclosure into the heat pump in order to be structurally dependent from it and not as a separated accessory.

The final observation was about the integration of the compressor module into a wall of the building. Even if, in the questionnaire, this option was one of the least voted, it should not be discarded since it could have potential in the Italian market considering that it is a common practice there.

To conclude, the user experience proved to be a success since it met the preconised objectives, providing valuable information about the market chances and the preferred places to install the modules due to the consistent outcome from the questionnaire.

In addition, as a result of the group discussion and the interviews, it was possible to collect valuable and helpful information about the positive and negative aspects of the prototype. The final contribution contemplates the proposals made regarding the future of the developed prototype. Those suggestions contribute to the extent that will sustain the next steps to be taken in the development of the studied matter.

Defining a monetary value about the added value of the modularity feature and the WPC enclosure was the only objective left unfulfilled. This may have occurred sinced there was not a similar product in the market in order to position the solution within the heat pump market.

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Chapter 8

Conclusion

8.1 Summary

The main focus of this master thesis work was to reduce the noise impact of a heat pump. In order to accomplish this and considering that the main noise sources of a heat pump are the compressor and the fan, a modular solution, separating those two components, was proposed and implemented. Therefore, the developed modular heat pump was composed by two modules: the compressor module and the fan module.

There were several engineering challenges to be considered and solved during the development phase of the modular heat pump. Consequently, a CAD design was created as the starting point of the project with the purpose of predicting the best methodology to develop the proof of concept.

Subsequently, the structural part of both modules was developed reusing structural components from the initial monobloc heat pump and prototyping new components obtained through different rapid prototyping technologies, such as, laser cut, hot wire cut and metal bending. In addition a drip tray was installed in the fan module as well as a protection grid beyond the evaporator.

In order to connect the evaporator, placed inside the fan module, with the refrigerant circuit placed inside the compressor module, a pair of flexible hoses was used with an extension of 5m. Consequently, the refrigerant circuit was re-worked and two shut-off valves were installed to allow the manual pump down cycle. The electrical and communication lines between the components of both modules were also remade and extended with the same length as the flexible hose. This determined that the developed modules can be installed at a maximum distance of 5m. As a result, a functional modular heat pump was developed consisting of two modules. Thus, the end user is able to manage the preferred location for both modules installation and consequently the two main noise sources location.

For instance, if there are complaints from the neighborhood, it is possible to place the compressor module inside the building, reducing significantly the noise generated. On the other hand, if the user is annoyed with the produced noise, with this solution, it becomes possible to manage the installation of the modules furthest from the building or even camouflaged in vegetation.

The second part of this master thesis work consisted of formulating a solution to significantly reduce the sound levels of the compressor module. In order to accomplish this, an acoustic enclosure was designed and built from scratch, with the purpose of creating a second sound barrier around the compressor module.

The development of the acoustic enclosure was based in three acoustic principles: high acoustic impedance gradient between materials, sound leakage insulation and absence of structure-borne noise.

The first principle, responsible for ensuring a high sound blocking effect, is strongly affected by the choice of the enclosure material since it is highly dependent on the amount of added mass to the system. In addition, it was decided to maintain an air gap between the walls of the enclosure and the compressor module in order to guarantee a substantial high density gradient between the air and the walls of the enclosure.

Multiple high density materials were considered taking into account different criteria, such as, waterproof, workability, customisation and sound transmission loss. Culminating in the choice of a WPC (Wood plastic composite) material composed of rice husk in a PVC matrix.

In order to guarantee the second principle, it was used sealing foams between the WPC boards, the hoses and the structural profiles forming the external cage. The third principle was partially guaranteed by the air gap, which separated the compressor module from the acoustic enclosure however, the hoses coming out from the module were inevitably in contact with the WPC boards.

Subsequently, multiple sound measurements were performed to the compressor module with and without the WPC enclosure, considering different sound improvements.

From the results analysis, it is possible to conclude that all the sound improvements implemented revealed a decrease onto the sound power levels radiated. In addition, the compressor module proved to emit considerably less noise than the entire heat pump, as a monobloc unit.

Nevertheless, the most relevant inference to be done is about the notable performance that the WPC presented regarding sound reduction. The levels of sound power obtained in the **WPC 2ND** were below 40dB for all operating conditions, with a mean value of 34dB. Moreover, in several operating conditions the sound power level was close to 30dB. To situate these values in terms of human perception, the sound emitted by a human whispering is characterised by a value of 30dB [65].

On the other hand, the developed prototype showed some tonality problems under certain operating conditions, what can comprise the good performance in terms of sound power reduction.

The last part of this master thesis work involved of a user experience analysis. This activity proved to be very fruitful since it provided valuable information about market chances, preferred places to install the modules, positive and negative aspects and improvement suggestions. However, it was not possible to predict a monetary added value of the developed solution.

One important point to highlight from the user experience is about its contribution to define what to be done following this master thesis work.

8.2 Suggestions for future work

There are several topics that can be subject of future work following this master thesis.

To start with, there are numerous improvements which can be implemented into the developed modules. The first to consider is about the EPP lid of the compressor module due to the fact that, during the sound measurements, the top face of the module presented the second highest values of sound pressure. Therefore, it would be worthwhile to re-design the lid of the module, considering changing the material, its thickness or adding different layers of materials.

In addition, there are two weaknesses associated to the refrigerant valves installed in the compressor module. The highest values of sound pressure were detected on the side of were the valves were installed. Actually, it was possible to distinguish the noise radiated by the circulation of the refrigerant through the valves from the overall noise. Therefore it would be important to encapsulate or integrate the valves inside the module in order to attenuate the noise.

Furthermore, the brackets that support those valves proved to be structurally weak since, when the refrigerant hoses were tightened to the valves, a brazed connection in the refrigerant circuit broke due to the excessive torsional force transmitted. Consequently, the support of the valves to the module should be redesigned.

Regarding the developed enclosure, there are some improvements that can be done in future works. To start with, establish the ideal number of boards to use on each side of the enclosure taking into account the maximum width of the WPC boards possible to produce ensuring quality.

Secondly, it would be beneficial to extrude the structural profiles of the enclosure also in WPC and perform new sound measurement along with an analysis on the structural stability for this solution.

The developed solution presented some tonality problems which compromise the efficiency of the enclosure therefore this problem deserves to be a subject of study in future works. Thus, it would be worthwhile to evaluate what is the best size of the air gap between the WPC walls and the compressor module and tuning it for the operating conditions that presented worst sound power values. In addition, it could also be interesting to investigate what is the best geometrical profile and thickness of the WPC boards in order to obtain an interior finishing of the enclosure that would decrease reverberation.

These future studies could be sustained by numerical simulation and validated by experimental measurements.

Lastly, during the development of this master thesis work, there was also the intention of testing the prototype with the compressor module hung on the enclosure structure instead of placed on the floor. With the purpose of suppressing the structure-borne noise transmitted through the floor, since when walking around the enclosure it was possible to feel the vibration of the floor. This concept was not tested, therefore it is also proposed as a future work, evaluating which is the best hanging solution in terms of hanging points and materials to be used.

Considering a more general approach, not focusing in improving the developed prototype but rather proposing new projects based on conclusions withdrawn from the work developed, there several promising proposals, strengthened by the UX, which can be defined as future works.

To start with, since the WPC presented good results, it would be worthwhile to integrate these material in a monobloc unit maintaining the air gap feature as well as sealing and avoiding structure-borne noise around the compressor and refrigerant parts.

This could be interesting to study considering that the expected refrigerant to be used in the future is propane. This refrigerant is highly inflammable, consequently it is predictable that heat pumps will tend to be as compact as possible avoiding the need to perform refrigerant circuit installations.

Another proposal would be to attach the compressor module to the IDU, developing a new concept of indoor units which would include all the components present in the developed compressor module. This solution could be relevant since it would reduce the number of parts to be installed and the concept of separating the two main noise sources would remain the same.

The last suggestion for future work would be the replacement of the metal panels of the modules with WPC boards. Consequently, the WPC feature would be integrated into the structure of the heat pump and the metal panels would be replaced by a sustainable material.

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Appendices

Appendix A

User Experience

MODULAR HEAT PUMP QUESTIONNAIRE

QUESTIONNAIRE

1. In your opinion what are the market chances for this concept from 1 to 5?
   - ☐ 1 (No chances at all – Ridiculous idea)
   - ☐ 2 (Poor chances – It will not work)
   - ☐ 3 (Uncertain chances - "Nim")
   - ☐ 4 (Good chances - It is a good idea)
   - ☐ 5 (Excellent chances - Brilliant Idea!)

2. In your opinion, what is the added value for the modular solution?
   a. ________€

3. In your opinion, what is the added value for the modular solution with the WPC feature?
   a. ________€

4. Rate the possible place to install the compressor module between 1 to 5 (1 less preferred – 5 favourite).
   - _ Basement/Garage
   - _ Wall integrated
   - _ Buried
   - _ Backyard
   - _ Front yard
   - _ Rooftop
   - _ Rooftop of the garage

5. Rate the possible place to install the fan module between 1 to 5 (1 less preferred – 5 favourite).
   - _ Basement/Garage
   - _ Backyard
   - _ Front yard
   - _ Rooftop
   - _ Rooftop of the garage

6. Would you buy it?
   - ☐ Yes.
     Why?
   - ☐ No.
     Why not?

THANK YOU!

Figure A.1: Questionnaire given to the participants of the user experience.

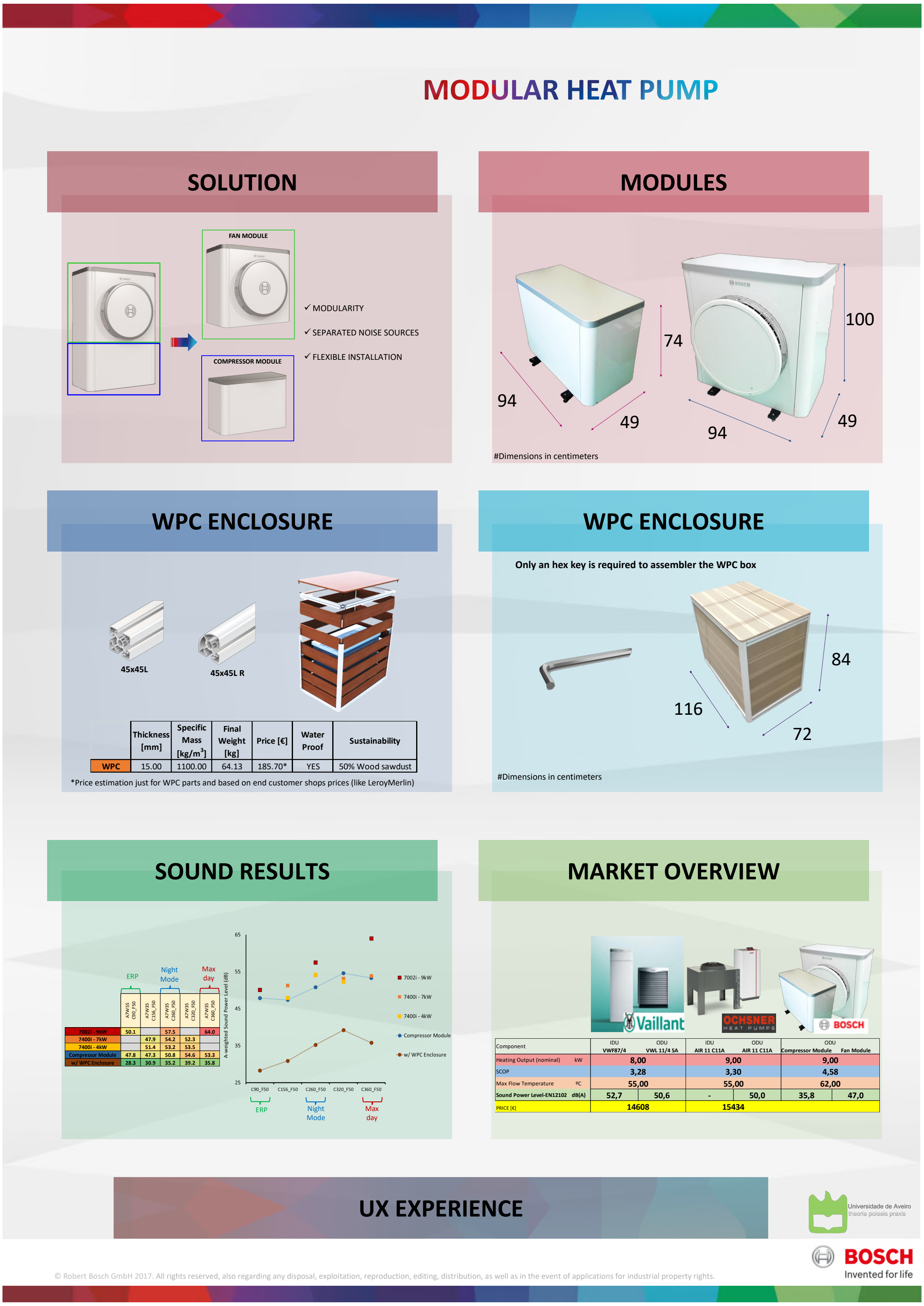

Figure A.2: Affixed poster in the UX room and in the acoustic chamber.

1. In your opinion what are the market chances for this concept from 1 to 5?

	1 (No chances at all – Ridiculous idea)
	2 (Poor chances – It will not work)
3	3 (Uncertain chances - “Nim”)
6	4 (Good chances - It is a good idea)
3	5 (Excellent chances - Brilliant Idea!)

Reference: HP 5000€

2. In your opinion, what is the added value for the modular solution?

0.00 €	0.00 €	3.00%	699 €	500 € (5-10%)	10,000 €	50 €	- €	2,000 €	50 €	400 €	- €
									For installation of system		

3. In your opinion, what is the added value for the modular solution with the WPC feature?

+20 - 30%	300	5.00%	899 €	1,000 € (10-20%)	11,500 €	150 €	500 €	3,000 €	100 €	1,000 €	600 €
									For installation of system		

4. Rate the possible place to install the compressor module between 1 to 5 (1 less preferred – 5 favourite).

													Mean
Basement/Garage	4	5	3	5	5	4	5	2	3	5	1	5	3.92
Wall integrated	0	5	1	5	5	1	1	4	4	2	1	1	2.50
Buried	5	5	5	3	4	1	1	4	2	2	2	1	2.92
Backyard	3	3	2	3	4	5	4	3	3	3	4	3	3.33
Front yard	0	3	1	2	3	5	4	1	2	2	3	2	2.33
Rooftop	1	1	3	5	1	5	4	5	1	1	1	5	2.75
Rooftop of the gar.	2	1	4	4	2	5	4	5	5	4	3	4	3.58

5. Rate the possible place to install the fan module between 1 to 5 (1 less preferred – 5 favourite).

													Mean
Basement/Garage	1	2	5	1	3	1	1	2	1	1	1	X	1.73
Backyard	3	5	5	1	4	5	4	3	5	5	4	4	4.00
Front yard	2	2	3	1	2	5	4	1	2	3	3	2	2.50
Rooftop	5	1	2	5	5	5	1	5	2	2	3	5	3.42
Rooftop of the gar.	4	1	2	4	5	5	1	5	3	4	3	5	3.50

6. Would you buy it?

Yes.	10
No.	3

Figure A.3: Answers obtained from the proposed questionnaire.