UNIVERSITAT POLITÈCNICA DE VALÈNCIA

Escuela Técnica Superior de Ingeniería del Diseño

DESIGN OF A 100 KW ROOF-TOP AIR CONDITIONING UNIT AND USING R32 COOLANT TO MEET CURRENT EFFICIENCY REQUIREMENTS

TRABAJO FINAL DEL

Grado en Ingeniería Mecánica

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RESUMEN

Actualmente y debido al creciente interés por reducir las emisiones de CO2 en la Unión Europea se están regulando y adoptando determinadas legislaciones con objeto de llegar a tener una Europa libre de carbono para el 2050. Esto está afectando de manera crucial al sector de la edificación que representa entre el 30% y el 40% de las emisiones totales y sobre todo al sector de la climatización y aire acondicionado, ya que a su vez este puede representar entre un 70% y 80% de esas emisiones.

Por ello la UE en los pasados años ha venido publicando una serie de normativas donde:

- Se pretende fomentar el uso de sistemas de compresión de vapor para cubrir estas necesidades.
- Se pretende favorecer el uso de fluidos con menos emisiones en este tipo de equipos.
- Se pretende que únicamente se vendan equipos con un mínimo de eficiencia.

Este Trabajo Fin de Grado consiste en realizar un diseño competitivo de una bomba de calor de techo de alta capacidad. El objetivo es que la máquina sea capaz de alcanzar alrededor de los 100 kW de capacidad para el caso más restrictivo. La máquina es reversible, lo que significa que la bomba suministra tanto frío cómo calor. La idea principal es evaluar la posibilidad de diseñar una bomba de calor que funcione con un refrigerante con menor poder de calentamiento global que los que habitualmente se usan en estos equipos. Por tanto, tras haber comparado una serie de refrigerantes, se ha llegado a la conclusión de que se usará el R-32 que, aunque es inflamable puede trabajar con componentes actualmente disponibles en el mercado y el aspecto de su inflamabilidad puede ser regulado.

Una vez seleccionado este refrigerante, se usará el software de modelado IMST-ART para diseñar este equipo de acuerdo con los criterios de eficiencia mínimos requeridos por la normativa.

Por último y con objeto de obtener una estimación aproximada del funcionamiento real de este tipo de sistema, se evaluará el funcionamiento del equipo usando un recuperador de calor, configuración que de acuerdo con la normativa española es necesaria para este tipo de sistema y se compararan los resultados obtenidos con máquinas de R410A actualmente existentes en el mercado.

Palabras claves:

Bomba de calor, roof-top, IMST-ART, refrigerante, R-32, Ecodiseño, Normativa Europea.

ABSTRACT

Due to the growing interest in reducing CO2 emissions in the European Union, legislations are being regulated and adapted to achieve a carbon-free Europe by 2050. The building sector, which accounts for between 30 and 40 % of total emissions, is highly affected. Furthermore, the air-conditioning and climatization sector represent between 70 and 80 % of these emissions. That is why the EU has been publishing a series of regulations in recent years in which:

- Encouraging the use of steam compression systems to meet these needs.
- Encouraging the use of fluids with fewer emissions in this type of equipment.
- Selling only equipment with a minimum of efficiency.

This Final Degree Project consists of performing a competitive design of a high capacity ceiling heat pump. The goal is to reach about 100 kW capacity for the most restrictive case. The machine supplies cold and hot air, which means it is reversible. The main idea is to evaluate the possibility of designing a heat pump that works with a refrigerant with less global warming potential than the common fluids used. Therefore, after comparing several refrigerants, it has been concluded that the R-32 will be used. Although it is flammable, it can work with components currently available on the market and its flammability can be regulated.

Once this refrigerant has been selected, IMST-ART modelling software will be used to design this equipment according to the minimum efficiency criteria required by the regulations.

Finally, to obtain an approximate estimate of the actual performance of this type of system, the execution of the equipment shall be assessed using a heat recovery system, configuration required by the Spanish normative. Furthermore, the obtained results will be compared with a machine that works with R-410A.

Key words:

heat pump, roof-top, IMST-ART, refrigerant, R-32, Eco-design, European Normative.

RESUM

Actualment i a causa del creixent interès per reduir les emissions de CO_2 en la unió europea s'està regulant i adoptant determinada legislació a fi d'arribar a tindre una Europa lliure de carboni per al 2050. Això està afectant de manera crucial al sector de l'edificació que representa entre el 30% i el 40% de les emissions totals, i sobretot al sector de la climatització i aire condicionat ja que al seu torn aquest pot representar entre un 70% i 80% d'aqueixes emissions.

Per aquest motiu, la UE en els passats anys ha vingut publicant una sèrie de normatives on:

- Es pretén fomentar l'ús de sistemes de compressió de vapor per a cobrir aquestes necessitats.
- Es pretén afavorir l'ús de fluids amb menys emissions en aquesta mena d'equips.
- Es pretén que únicament es venguin equips amb un mínim d'eficiència.

Aquest Treball Fi de Grau consisteix a realitzar un disseny competitiu d'una bomba de calor de sostre d'alta capacitat. L'objectiu és que la màquina siga capaç d'aconseguir al voltant dels 100 kW de capacitat per al cas més restrictiu. La màquina és reversible, cosa que significa que la bomba subministra tant fred com calor. La idea principal és avaluar la possibilitat de dissenyar una bomba de calor que funciona amb un refrigerant amb menor poder de calfament global que els que habitualment s'usen en aquests equips. Per tant, després d'haver comparat una sèrie de refrigerants, s'ha arribat a la conclusió que s'usarà el R-32 que, encara que és inflamable pot treballar amb components actualment disponibles en el mercat i l'aspecte de la seua inflamabilitat pot ser regulat.

Una vegada seleccionat aquest refrigerant, s'usarà el programari de modelatge IMST-ART per a dissenyar aquest equip d'acord amb els criteris d'eficiència mínims requerits per la normativa.

Finalment i a fi d'obtindre una estimació aproximada del funcionament real d'aquesta mena de sistema, s'avaluarà el funcionament de l'equip usant un recuperador de calor, configuració que d'acord amb la normativa espanyola és necessària per a aquesta mena de sistema i es compararan els resultats obtinguts amb màquines de R410A actualment existents al mercat.

Paraules clau:

Bomba de calor, roof-top, IMST-ART, refrigerant, R-32, Ecodisseny, Normativa Europea.

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NOMENCLATURE

Symbol	Meaning	Unit
COP	Coefficient of performance	kW/kW
Q_C	Condenser's power	kW
Q_F	Evaporator's power	kW
L	Supplied power	kW
Q	Capacity	W
U	Global coefficient of heat transfer	$W/K \cdot m^2$
A	Area	m^2
SHR	Sensible Heat Ratio	%
SC	Subcooling	ºK
SH	Superheating	ºK
Af	Frontal area	m^2
$\dot{V}$	Volume flow	m^3/h
u	Air velocity	m/s
Sl	Horizontal distance between fins	mm
St	Vertical distance between fins	mm
Dc	Tube diameter	’’
tf	Fins thickness	mm
β	Compactness	m^{-1}
V	Volume of the heat exchanger	m^3
H	Height	m
W	Weight	m
DC	Declared Capacity	kW
EERd	Energy efficiency factor at declared power	kW/kW
Qh	Reference annual heating demand	kWh
Qhe	Annual heating electric consume	kWh
Pdesignc	Thermic charge	kW
Hhe	Number of hours in active mode in heating	Hours
Hto	Hours in thermostat off mode	Hours
Hsb	Hours in sleep mode	Hours
Hck	Hours in crankcase mode	Hours
Hoff	Hours in off mode	Hours
Phe	Consume in thermostat off mode	kW
Pho	Consume in sleep mode	kW
Pck	Consume in crankcase mode	kW
Poff	Consume in off mode	kW
C_{dc}	Cooling Degradation coefficient	-
pl (Tj)	Part load factor	%
Tj	Exterior temperature	ºC
Pc (Tj)	Part cooling load	kW

SUSCRIPTS

Symbol	
Tin,air.evap	Air inlet temperature in the Evaporator
Tout,cond	Air inlet temperature in the Condenser
T.cond	Condenser temperature
T.evap	Evaporator temperature
ΔT,cond	Difference of temperatures at condenser inlet
ΔT,evap	Difference of temperatures at evaporator inlet
T7	Outlet condenser temperature
T6	Inlet condenser temperature
Ta,in	Inlet air temperature
T,a,out	Outlet air temperature
ΔT	Temperature variation
T2	Outlet condenser temperature
T1	Inlet condenser temperature
Nrows	Number of rows
Nt	Number of tubes per row

ACRONYMS AND ABBREVIATIONS

Acronym	Meaning
EU	European Union
GHS	Green House Gasses
AC	Air Conditioning
SC	Space Cooling
RAC	Room Air Conditioning
ODP	Ozone Depletion Potential
GWP	Global Warming Potential
ASHRAE	American Society of Heating, Refrigerating and Air Conditioning Engineers
CFCs	Chlorofluorocarbons or Halocarbons
HCs	Hydrocarbons
HCFCs	Hydrochlorofluorocarbon
HCFs	Hydrofluorocarbons
HVAC	Heating Ventilation Air Conditioning.
VRF	Variable Refrigerant Flow
SHR	Sensible Heat Ratio
CF	Conversion factor
DC	Declared capacity
CR	Capacity ratio

TECHNICAL REPORT

1. INTRODUCTION

This project is based on the design of a roof-top heat pump. The purpose is to introduce all the elements needed to know before the target. That will include descriptions, procedures, explanations, the normative that must be followed and the motivation for doing it.

Firstly, it is introducing some of the definitions of the system that is designing. Those definitions come from the UE 2016/2281 Regulation. This regulation establishes eco-design requirements for the design and putting into service of:

a. Air heating products with a rating heating capacity not exciding 1 MW.
b. Cooling products and high temperature process chillers with rated cooling capacity not exciding 2 MW.
c. Fan coil units.

1.1. Definition

Heat pump means an air heating product: of which the outdoor side heat exchanger (evaporator) extracts heat from ambient air, ventilation exhaust air, water, or ground heat sources; which has a heat generator that uses a vapour compression cycle or a sorption cycle; of which the indoor side heat exchanger (condenser) releases this heat to an air-based heating system; which may be equipped with a supplementary heater; which may operate in reverse in which case it functions as an air conditioner. [1]

1.2. Types of heat pumps

There are many different types of heat pumps; here is briefly commenting some information about the different systems.

1.2.1. By heat source

- **Air source heat pump**: The air is taken from the outside and circulated across its heat exchanger. This air temperature will change depending on the heat pump's function, which can be cooling or heating. [2]
- **Ground source heat pump**: A pipe system is buried in the ground to some extent and the underground's heat is the heat pump's source of energy.[2]

- **Water source heat pump**: the source of energy in this kind of systems is the water, which can be taken from the sea, lakes, rivers or even from ground aquifers. [2]
- **Water Loop heat pump system**: a closed water loop is used as a source of energy to supply. It is different from water source systems.[2]
- **Exhaust air heat pumps**: from the recover heat from exhausted air and used to conditionate a space. [2]

1.2.2. By the source and/or delivery

The first term is the source and the second is the delivered fluid:[2]

- **Air-to-Water** (A-W): both sources are air
- **Air-to-Air** (A-A): the outer source is air and the inner's is water
- **Water-to-Water** (W-W): both sources are water

1.3. Heat pump choice

There are numerous places to install a heat pump system, but this project aims to design a roof-top heat pump driven by an electric compressor, of which the evaporator, compressor and condenser are integrated into a single package.

Therefore, the system that is designing is already defined, so the compression cycle type is the only step to set remaining. In this case, the heat pump type is going to be **air-to-air**. It consists of a heat pump that has a heat generator that uses a vapour compression cycle driven by an electric motor or internal combustion engine and whereby the outdoor side heat exchanger (evaporator) allows heat transfer from ambient air. [1] Hence, the whole system's components are designing following those specifications.

Air-to-Air heat pumps are typically installed into conventional balance ventilation systems incorporating air-to-air heat recovery. The machine provides additional heat recovery for the system, with the evaporator unit of the heat pump inserted into the extract air duct and the condenser element placed in the supply duct. Any additional heat is transferred from the extract to supply air via the heat pumps vapour compression. [3]

The choice of an air-to-air system has been for several reasons. The main one is due to the ease of operation and the low service and installation costs. Thus, due to the low maintenance, the reduced number of components and reliability, it is a potential operation system. These systems possess a significant heat transfer capacity, they do not need to install complex heat distribution systems and are highly efficient. Their rating efficiency is around 3.0-4.0 of SCOP. Taking the ambient air as a source also means that it is environmentally friendly.

Ambient air is by far the most common heat source for heat pump applications worldwide. The reason for this is the unlimited availability that enables an uncomplicated and quick installation. In most European climates, the temperature of ambient air changes significantly depending on the time of year. The fact that the performance of a heat pump reduces as the temperature of the heat source drops, leads to unfavourable characteristics. The execution of an ambient air heat pump will decrease as the heating demand is increasing. At determined points, the temperature difference between the heat source and heat sink will be too high. It will result in difficulties in the heat pump's operation. It can be solved with an auxiliary heating system that will help when the demand raises considerably. [4]

1.4. Heat pump’s main components

These are the heat pumps’ main components:

- Compressor
- Expansion valve
- Evaporator
- Condenser
- Four way reversible valve

Further information about the main components will be explained. Firstly, it is briefly explaining how the working principles. From this cycle, these machines provide heat energy.

1.4.1. Cycle

The most simplistic heat pump configuration consists of an outdoor unit, containing a compressor and heat exchanger working as an evaporator in heating mode and as a condenser when it is summer as shown in Figure 1. The mechanical energy provided by the heat pump during its operation and the exchanged heat with the environment has to be considered as the produced "useful effect". [2]

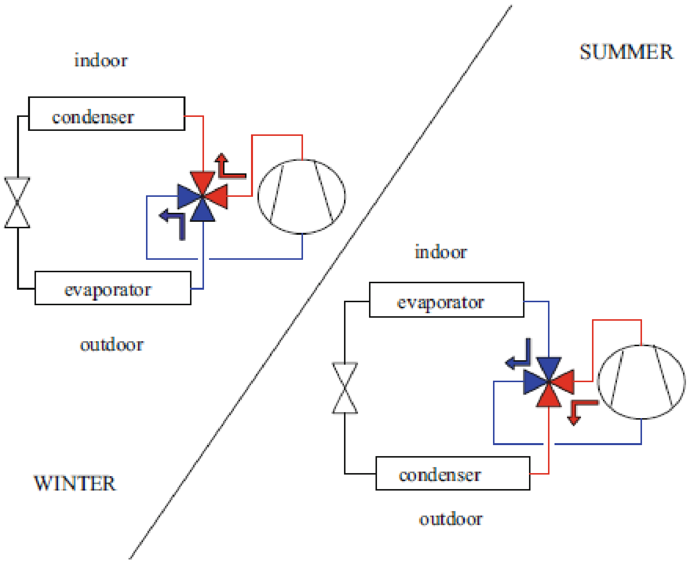


Figure 1-Heat pump configuration depending on the seasons

Coefficient COP is defined to characterize heat pumps performances as the ratio between the energy produced and the energy needed to deliver it. Figure 2 shows a conventional compression scheme that will clarify the formulas. It is the working principle of fluid compression. In addition, it includes an entropy-temperature diagram in which is represented the whole cycle. [2]

→Winter

$$COPw = \frac{Q_C}{L} = \frac{1}{1 - \frac{Q_F}{|Q_C|}}$$

→Summer

$$COPs = \frac{Q_F}{L} = \frac{1}{\frac{Q_C}{|Q_F|} - 1}$$

- Q_C → Condenser's power [kW]
- Q_F → Evaporator's power [kW]
- L → Supplied power [kW]

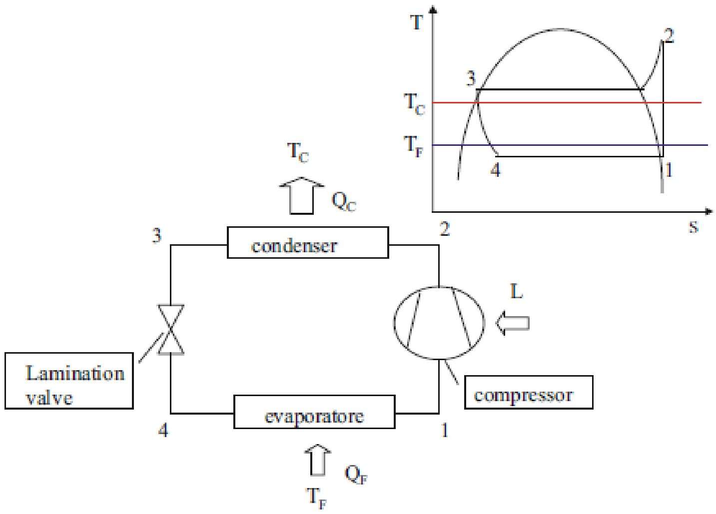


Figure 2-Basic scheme of compression heat pumps

The compression cycle consists of the vapour coming is absorbed by the compressor due to a piston motion. Then, the fluid will be compressed and released to the condenser. The condenser will receive a high-pressurised vapour. It will condensate the fluid, decreasing its temperature but maintaining its pressure. The temperature reduction is due to the heat transmitted to the environment. Thus, from the compressor, the cycle carries a high-pressurised fluid with elevated temperature. The solution will go through the expansion valve, which will decrease the fluid's pressure. It will generate a low pressure and temperature fluid that will arrive at the evaporator. The heat exchanger will evaporate the low-temperature solution, maintaining the pressure. The fluid's temperature will increase because of the system's heat transmission with the conditioned area. Therefore, the fluid has been heated a cooled, delivering or absorbing the heat of the system depending on its configuration.

This cycle is defined in Figure 3 which shows the component's procedure. In addition, Figure 4 provides information about the point before and after every cycle's part and represented in a p-h diagram. [2]

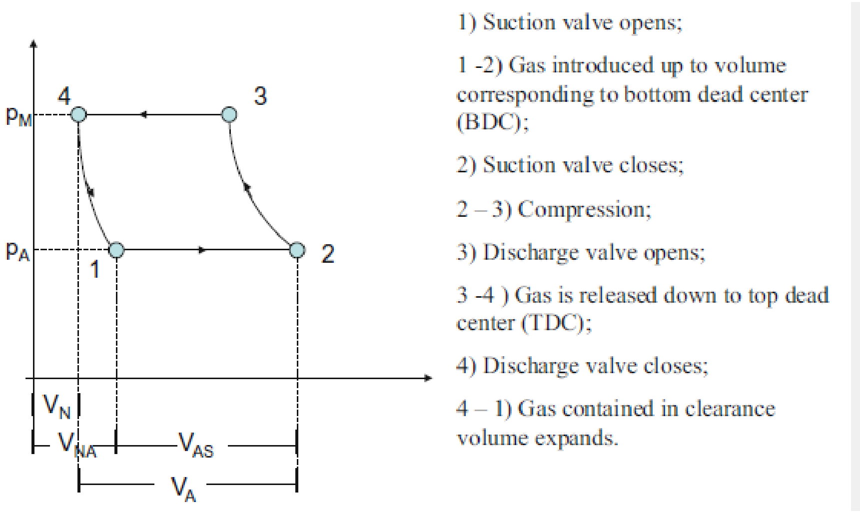


Figure 3-Components' procedure in the compression cycle

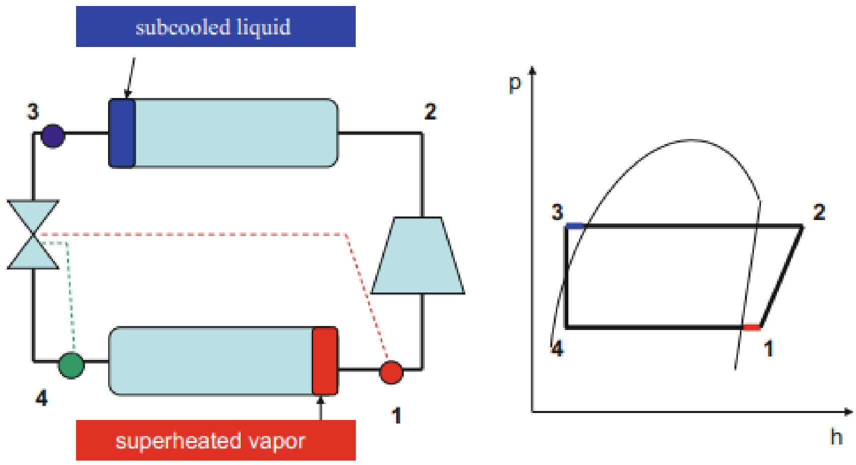


Figure 4-Heat pump cycle

1.4.2. Compressor

Its function consists of absorbing the refrigerant fluid, compress it increasing its temperature due to the pressure increase. Then it delivers a high-pressure fluid to the condenser. There are several compressor types:

1.4.2.1. Reciprocating compressors

A reciprocating compressor to supply gasses under high pressure is used for well-known applications. It usually features a piston, cylinder, crankshaft, connecting rod, spring, suction valve, discharge valve, and bearings. The piston is driven by a rotating crankshaft, which causes its reciprocating motion within the cylinder. Some information is given about this kind of compressors in Figure 5. [5]

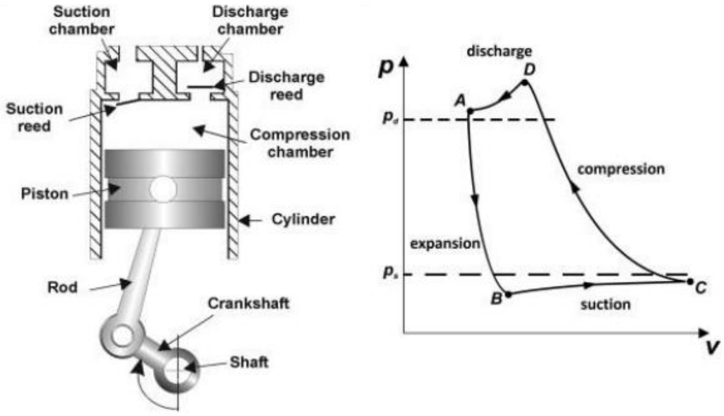


Figure 5-Reciprocating compressors components and working diagram

1.4.2.2. Screw compressors

Thanks to the technological progress in heating and cooling applications, rotary compressors are often employed instead of reciprocating compressors. Among other things, it is due to their smaller size, bit noisy, smoothly running, low vibration and better control and modulation capability. Figure 6 shows this kind of compressor configuration. [2]

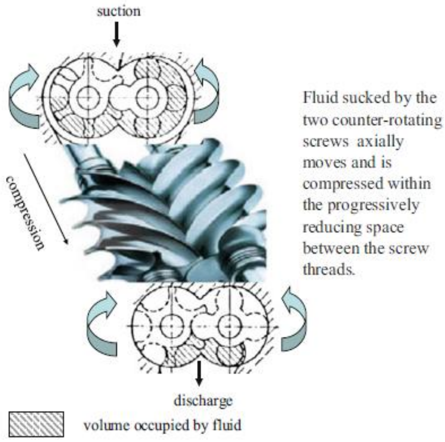


Figure 6-Screw compressors working scheme

1.4.2.3. Scroll compressors

Two scrolls (spirals), a fixed and a movable one constitute scroll compressors. The latter is driven by a shaft that makes it orbit (not rotate) about the shaft axis. Therefore, a chamber is formed, compression starts once the suction port is sealed off, progressively reducing the gas volume between the two scrolls. The usual configuration of a scroll compressor is shown in Figure 7. [2]

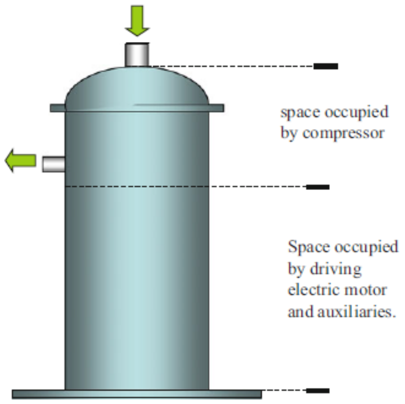


Figure 7-Scroll compressor's typical configuration

1.4.3. Expansion valve

Thermostatic regulation valves function consist of regulating the flow between the heat exchangers, acting as an accelerator between the high and low-pressure refrigeration systems' sides of the cycle. It ensures that at the evaporator's inlet, the fluid's pressure matches the set point to not break the device. This guarantees that the evaporator works fully adequate, and no liquid refrigerant reaches the compressor, which will damage the device. [6]

Thermostatic expansion valves consist of two parts. The first is the body, acting as the regulating system actuator. The second part is the orifice, which contains the regulator and performs the expansion of the fluid. Figure 8 shows an expansion valve model in which can be seen how the fluid passes through it. [6]

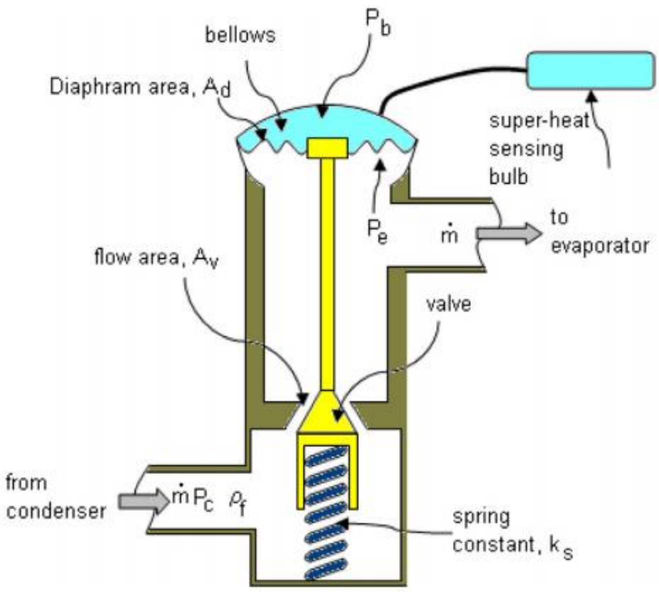


Figure 8-Thermostatic expansion valve configuration

1.4.4. Evaporator and condenser

These devices can exchange heat with several types of indoor and outdoor sources. As it was said, depending on the thermal sources heat pumps work with, can be classified as air/air, air/water and water/water heat pumps. Depending on the heat pump's type, the exchangers are going to have different configurations. [2]

When the heat source is air, the air-cooled heat exchanger consists of a rectangular box. That box is constituted of a specific configuration of finned tubes. One or multiple fans will provide conditioned air. If the refrigerant exchanges heat with water, plate and frame heat exchangers are used. They are high heat transfer efficiency and compact heat exchangers. The plates are generally spaced by rubber sealing gaskets (Gasketed Plate Heat Exchangers GPHE) and are pressed to form troughs at right angles to the main direction of the flow. [4]

There are more heat exchanger types, like "tube in tube" heat exchangers. Constituted by two coaxial tubes with the inner one corrugated in order to increase the heat transfer area constitute these devices. Other examples of heat exchangers are "shell and tube" heat exchangers. [2]

In this project, the system will be an air-to-air type. Thus, heat exchangers' type must correspond with the system configuration. An air-to-air heat exchanger brings two air streams of different temperatures into thermal contact, transferring heat from the exhausted air to the incoming air during the heating season and the opposite process during the cooling season.

These type of batteries does not have moving parts, making them very reliable and long lasting. The only downside that they present is the periodical maintenance procedures due to dust accumulation over the fins. Energy cost savings can be significant using this type of batteries. These devices provide high cooling efficiency and low costs due to the reduced repairing and the system life span.[2]

The type of heat exchangers used for air-to-air configurations is the tube-fins heat exchangers. Tube-fins exchangers can be used for a wide range of tube fluid operating pressures. Finned tube heat exchangers use air to cool or heat fluids and to capture or recover waste heat. The tube surface is enhanced by the addition of fins to increase the surface area of the tube and improve its thermal performance. The profile of the fins has a critical effect. It is essential to ensure a proper connection on the tube surface to provide maximum thermal conductivity [2]. Figure 9 shows an example of tube-fin technology. The larger the fins and the tighter the fin pitch, more thermal conductivity is achieved. The trade-off may be an increase in pressure drop that may, in turn, have adversely affect performance. A balance between the two opposite functions must be made to obtain an effective and optimal thermal performance. [2]

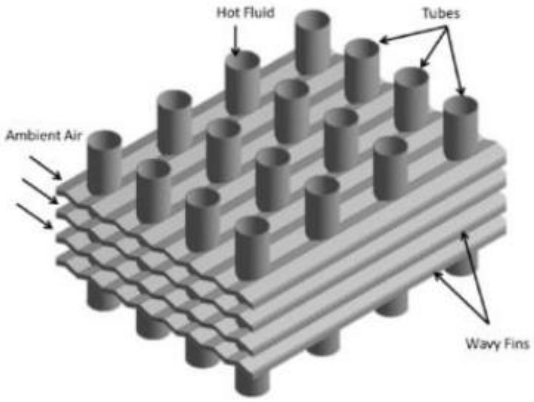


Figure 9-Tube-fin heat exchanger

1.4.5. Four way reversible valve

The 4-way Reversing Valve is the key component to provide heating and cooling from the system to the air-conditioned space by reversing the flow direction of the refrigerant. It is used at room air conditioners, packaged and central air conditioners. The reversing Valves are designed for Heat Pump Systems with a capacity from 3kW to 580kW, suitable for many refrigerants. [7]

The scheme of a reversible cycle is shown in Figure 10 and Figure 11. The cycle inversion is obtained by a slide S that puts into contact the three different ports by duos, moving side to side. That movement is caused by the refrigerant, which flows through the tubes. A valve activated by an electric coil regulates the flow. Depending on the valve position, the heat pump will be in heating or refrigeration mode. [2]

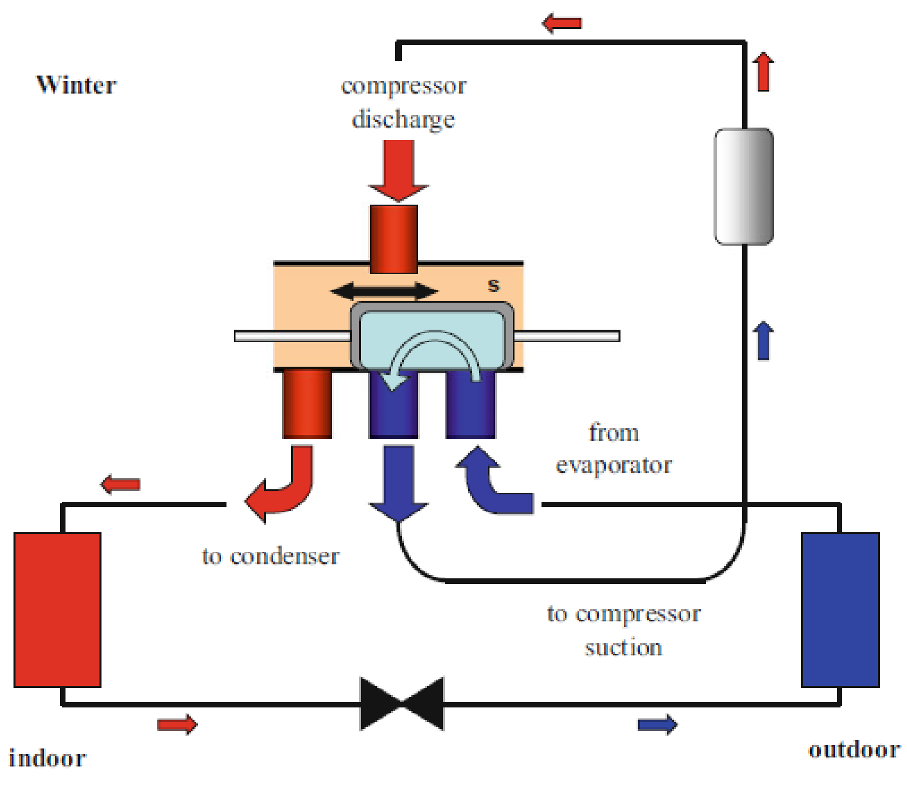


Figure 10-4 way valve heating mode scheme

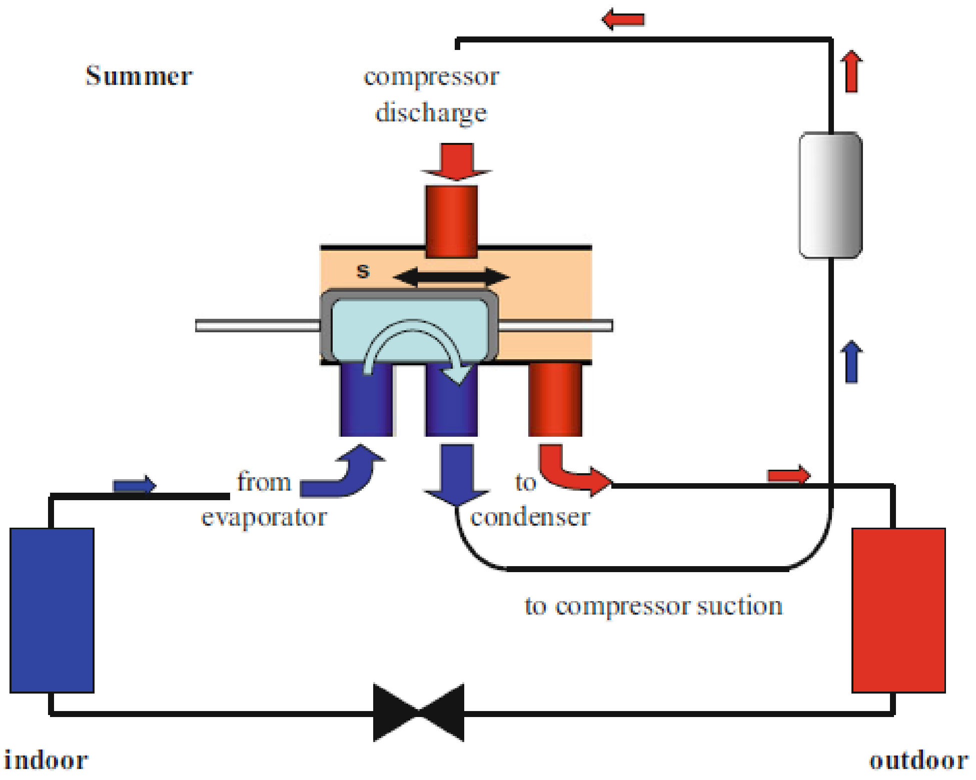


Figure 11-4 way valve cooling mode scheme

1.5. Motivation

In Europe, the ownership and electrical consumption of air-conditioners in buildings have been increasing for decades. Evaluation of the extent to which these trends will continue is fundamental to the development of policies to reduce the environmental and energy security consequences. [8]

The European Union (EU) member states have committed that by 2020-2021, they will decrease greenhouse gas (GHG) emissions by at least 20 % from 1990 levels and increase the energy used produced by renewable sources to 20 %. By 2030, the EU intends to significantly decrease the domestic emissions GHG by 40 %. Furthermore, by 2050 the EU member states have declared the intention of decreasing emissions in Europe by 80-95 %, relative to 1990 levels. The accomplishment of the Paris COP21 agreement will require an even higher reduction of GHG emissions. [8]

Research about the different Air Conditioning (AC) technologies has been made. Classification of AC types has been done, including a breakdown following diverse Space

Cooling (SC) generation (air-to-air or air-to-water) and distribution systems (decentralized or centralized). [9]

Air-conditioning systems: [8]

1. Air-to-air generation
    a. Decentralized systems
        i. Split systems
        ii. Multisplit systems
        iii. Single-duct systems
        iv. Packed units
    b. Centralized systems
        i. Variable refrigerants flow systems
        ii. Rooftops
2. Air-to-water generation
    a. Centralized systems
        i. Air or water distribution systems (chillers)

Each of the categories is more or less appropriate in different climate and building types. How the different AC technologies work in comparable conditions between the most common sectors has been analysed and it can be shown in Figure 12:[8]

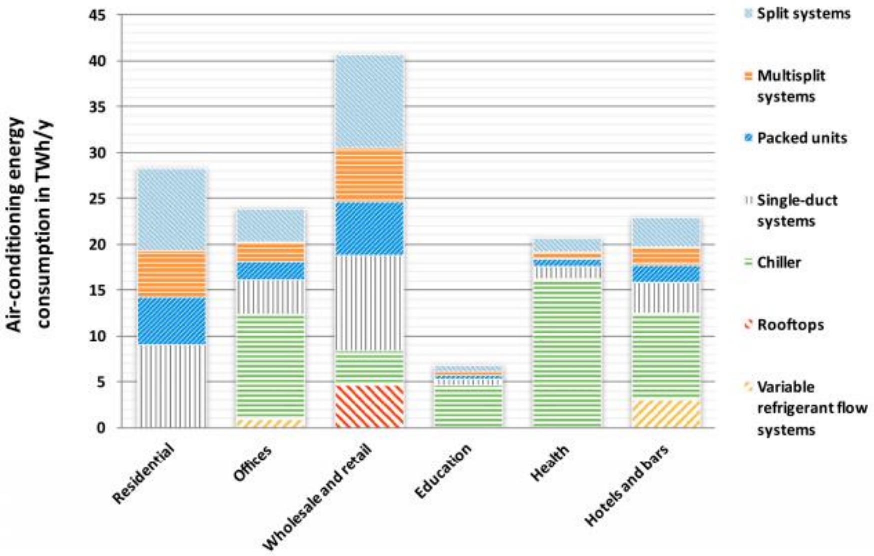


Figure 12-AC energy consumption per type and sector in EU

Besides, Figure 13 gives crucial information about the distribution of installed AC application by sector. [8]

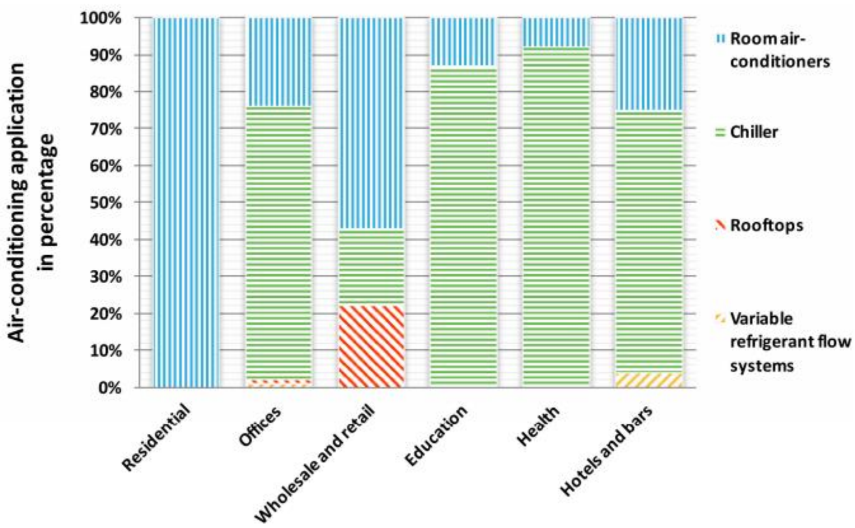


Figure 13-Distribution of installed AC application by sector type

Regarding the working full-load hours in similar conditions, Figure 14 displays estimated values of the different sectors. [8]

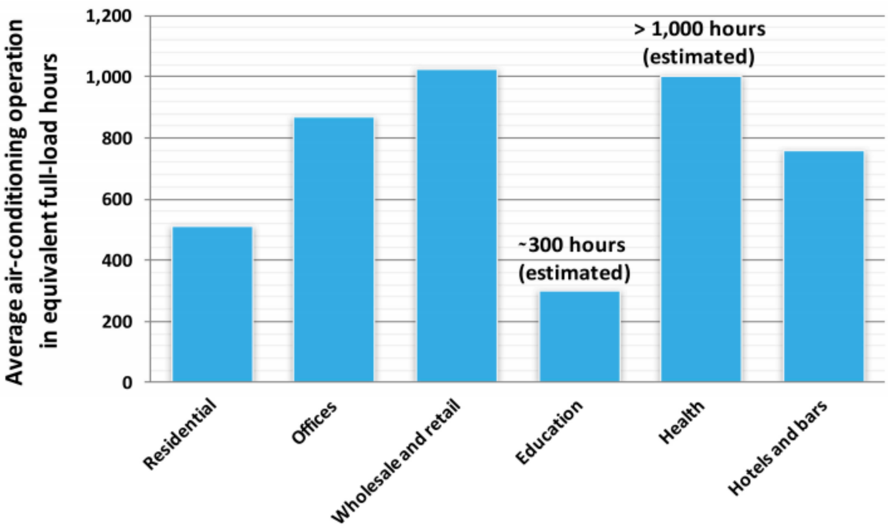


Figure 14-Distribution of average AC equipment full-load hours per sector

Therefore, the obtained data indicate exclusively almost Room Air Conditioning (RAC) dominates the residential sector. A possible reason could be that within dwellings, only certain rooms are air-conditioned.

Figure 15 shows the number of installed units per AC type around Europe. It will provide plenty of information about how distributed the AC equipment is. [8]

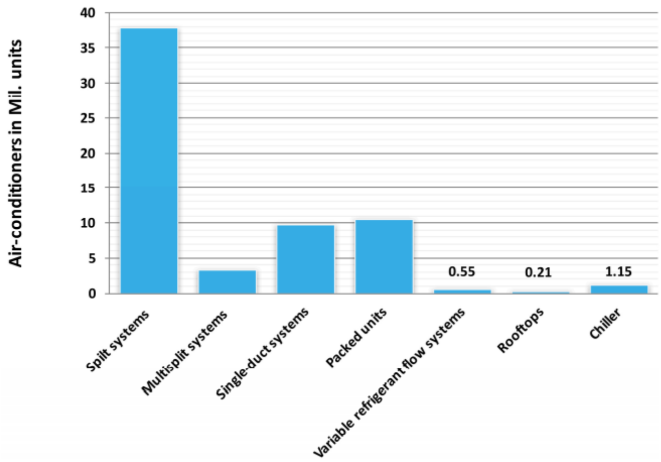


Figure 15-Number of operative units per AC type

The number of split systems is significantly higher than the rest of the AC equipment. It does not mean that it is the most suitable and beneficial configuration for EU purposes. As was commented previously, the aim of reducing the electricity input is primordial.

1.6. Objective

The Eco-design Regulation for Air Conditioning products was published in 2016. This regulation establishes the minimum requirements for AC equipment. Commission Regulation (EU) 2016/2281 of November 2016 implementing 2009/125/EC of the European Parliament and the Council concerning eco-design requirements for AC products. [9]

The seasonal space heating and cooling energy efficiency, and the seasonal energy performance ratio capture the seasonal demand variations by considering the product behaviour at full load and part load conditions established. The regulation defines the minimum levels to achieve for the different types of AC equipment. [8]

This project aims to design a device that complies with Regulation 2281/2016, which would mean it is an equipment that fulfils the achievements required by the EU state members. In addition, it is wanted to get a cost-effective system with remarkable efficiency. Due to the great COP provided by heat pumps, it will be the choice of this project. As it can be seen in Figure 16, the average capacity of roof-top projects is not very high. [8]

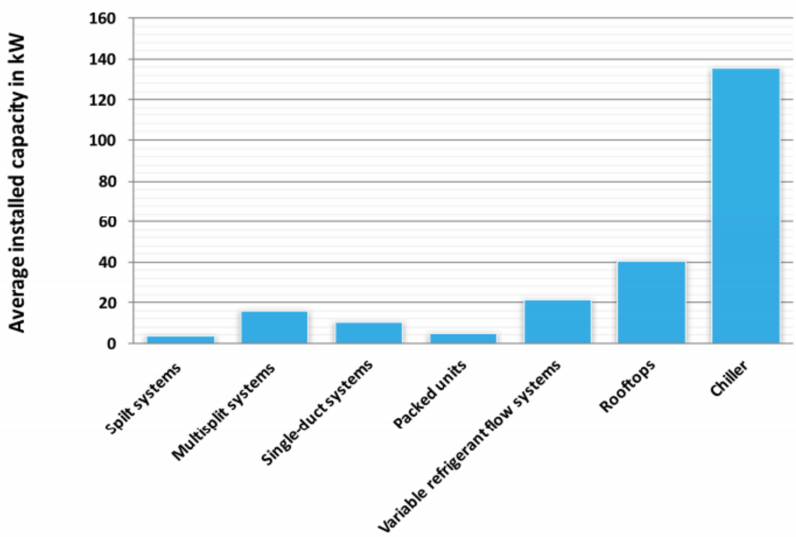


Figure 16-Average installed capacity per AC type

It is wanted to design a high-capacity roof-top to evaluate its performance and compare it against standard equipment. Figure 17 shows that roof-top devices do not present as higher performances as the rest of the AC types. [8]

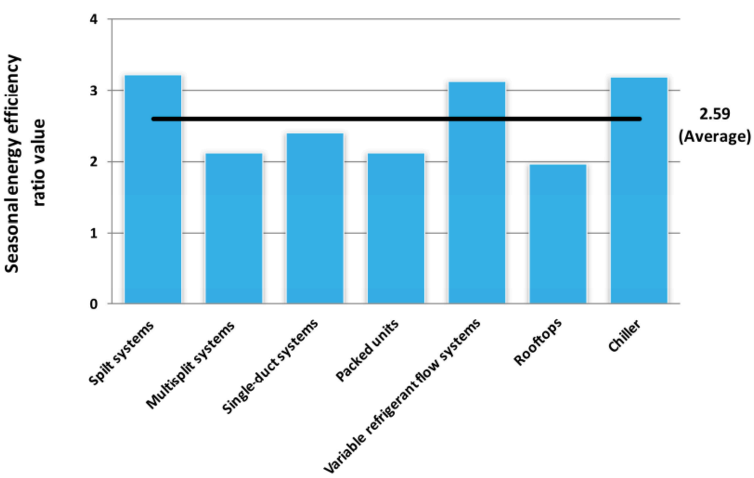


Figure 17-Seasonal energy efficiency ratio (SEER) per AC type

Therefore, the goal is obtaining proper values, which prove that the roof-top design would be a potential option. The number of roof-tops will increase in more sectors and make a transition to more sustainable AC systems. Not only is the performance meaningful, but also the emissions.

Nowadays, all these equipment uses as refrigerant R-410A this refrigerant has a GWP of 2000 and to find a substitute for it is one the major open points in the air conditioning sector because of it is difficult to match its high volumetric capacity, high efficiency and safe operation.

Thus, the structure of the project will consist of selecting the proper refrigerant and the components to build the system. Then, analyse the performance of the system and verify it fulfils with the normative requirements. Finally, it is comparing the design with a similar machine, which works with R-410A. The results will be evaluated, and conclusions will be delivered with the costs of the project and product specifications.

2. REFRIGERANTS

2.1. Definition

A refrigerant is the primary working fluid used for absorbing and transmitting heat in a refrigeration system. Refrigerants absorb heat at a low temperature and low pressure and release heat at a higher temperature and pressure. Most refrigerants undergo phase changes during heat absorption (evaporation) and heat-releasing (condensation). [10]–[12]

The letter R, which means “refrigerant”, followed by numbers that refer to different substances, identifies refrigerants. The standardization of this model is regulated by the ASHRAE (American Society of Heating, Refrigerating and Air Conditioning Engineers). [10]–[12]

Some features refrigerants must have are moderate work pressures and discharge temperatures, low compression rate, low freezing point, safe to manipulate, availability and low cost. [12]

2.2. Classification

General classification of refrigerants by their chemical composition:

2.2.1. By chemical composition

- **CFCs - Chlorofluorocarbons or Halocarbons**

Commonly used in refrigeration and air conditioning as refrigerants. Their use is decreasing due to their environmental impact. They are nontoxic, dangerous to handle and odourless. Some of the most common fluids within this group are the R-11, R-12, R-22 or R-113. They present high ODP reason why they are in elimination phase. [10]–[12]

- **HCs - Hydrocarbons**

Compounds by carbon and hydrogen. They include methane, ethane or butane. HCs are highly flammable, present low toxicity and are considered a potential alternative due to the zero impact to the ozone layer and very low global warming potential (GWP), besides being an affordable choice economically. Some well-known HCs refrigerants are the R-170, R-

290 and R600a. They are ethane, propane and isobutene. Their use requires many precautions. [10]–[12]

- **HCFCs - Hydrochlorofluorocarbons**

Compounds by hydrogen, chlorine, fluorine, and carbon. The component chlorine contributes to ODP, but as previously said, at a very low level. Even though, the purpose is reducing their use. R-22, R-123, R-124. A blend that contains both an HCFC and HFC is considered an HCFC refrigerant. [10]–[12]

- **HFCs - Hydrofluorocarbons**

Compounds by hydrogen, fluorine, and carbon. Principal replacement of CFCs and HCFCs for lack of chlorine composition. Some of them have a higher level of GWP. R-152a, R-32, R-125 and R-143a. A blend that contains different HFCs is considered an HFC refrigerant. [10]–[12]

- **HFOs – Hydrofluoric-olefins**

HFO refers to the chemical composition of the refrigerant. (HFOs are organic compounds composed of hydrogen, fluorine and carbon. HFO refrigerants are categorized as having zero ODP (Ozone Depletion Potential) and low GWP (Global Warming Potential) and offer a more environmentally friendly alternative to CFCs, HCFCs and HFCs. Those type of refrigerants would be the refrigerants used in the future. Now, they are not as developed and controlled as they would be required. The refrigerants are R-1234ze and R-1234yf. [13]

2.2.1.1. Inorganic compounds

- **Ammonia (R-717)**

Colourless gas with a powerful odour that let detect leaks in the system. It is soluble in water and lighter than the air. Despite causing several technical and health problems, it provides a great cooling capability. It is considered a great refrigerant but limited by the system design. [10]–[12]

- **Carbon Dioxide (R-744)**

Colourless, odourless, non-flammable, nontoxic, nonexplosive and is used in cascade refrigeration systems and food freezing applications. Generally used in air conditioning and refrigeration systems. Due to the lightweight, it generates a low COP. [10]–[12]

- **Water (R-718)**

Water is another natural refrigerant with a renewed interest because it is non-toxic, non-flammable, low cost and abundant. Water is widely used as a refrigerant in higher temperature lithium bromide – water (LiBr-H2O) absorption chillers where water is the refrigerant and lithium bromide is used as an absorbent. The challenge for absorption chillers is that even a double-effect absorption cycle only has a COP slightly greater than 1, electric drive centrifugal chillers have a COP greater than 5. [10]–[12]

In addition, refrigerants can be classified by the security regulation.

2.2.2. By the security regulation

2.2.2.1. Security

- **First group (G1)**: if it is not flammable or toxic.
- **Second group (G2)**: toxic or corrosive; flammable or explosive at 3,5 % or more by volume.
- **Third group (G3)**: fuel or explosive at less than a 3.5 %.
  [10]–[12]

2.2.2.2. Toxicity

- **Group A**: low toxicity
- **Group B**: high toxicity

Figure 18 shows an example of refrigerants' classification by security regulation. It comes from ASHRAE Standard 34, which explains the nomenclatures and class of refrigerants. [14]

Flammability		Lower Toxicity	Higher Toxicity
	Higher Flammability	**A3** R-290 Propane R-600a Isobutane	**B3**
	Lower Flammability	**A2** R-152a	**B2**
		A2L* R-32 R-1234yf R-1234ze(E)	**B2L*** R-717 Ammonia
	No Flame Propagation	**A1** R-22 R-134a R-410A R-1233zd(E) R-404A R-407C R-507A R-744 Carbon Dioxide	**B1** R-123
		Toxicity	

*A2L and B2L are lower flammability refrigerants with a minimum burning velocity of ≤10 cm/s.

Figure 18-Safety classification of refrigerants in ASHRAE Standard 34

2.3. Refrigerants' research

In previous years, CFC refrigerants have been the most used option due to the easy handle, acquisition and yield provided. Others must replace those refrigerants with lower GWP and ODP impact as the new normative of ASHRAE stipulates. Besides, cost, availability, efficiency and compatibility with compressor lubricants and equipment are other issues required for CFC replacements.[14]

The halogenated refrigerants used nowadays represents a threat to the environment because of their ODP and GWP. More halogenated such as CFCs, HCFCs and HFCs obtained from hydrocarbons, methane and ethane by replacing chlorine and fluorine in the place of hydrogen. If all hydrogen is replaced in halocarbons, they are fully halogenated, and it can produce massive damage to the environment in case of any system leaks appearances.[14]

Because of this, find environmentally friendly refrigerant replacements is needed, especially after the Kyoto and Montreal protocols, which require the substitution of all the CFCs by less harmful HCFCs by the end of 2020. By the end of 2030, also HCFCs almost must be abolished. [14]

In the last few years, the increased production of HFCs and their blends have brought remarkable benefits to the environment. They are replacements of CFCs and HCFCs, but also can generate a significant concentration of high GWP. [14]

There have been several types of research to replace the halogenated refrigerants in air conditioning and heat pumps to reduce emissions. For example, the R-22 has been considered several replacements such as R-290, R-407C or R-410A, especially in hot climates. [14]

After the Kyoto Protocol, the countries that signed in 1997 compromised to reduce the emissions that damage the environment and the production of fluids with high GWP and ODP. [15]

Some of the refrigerants that were strictly marked were R-134a, R-404A and R-410A [16] (GWP of 1430, 3922 and 2088 respectively) .[17]

The Kyoto Protocol is an international treatment signed in 1997 that took part in the UNFCCC (United Nations Framework Convention on Climate Change). [18] Were fixed the GHG (Greenhouse gas) emission from industrialized countries. [19] Figure 19 shows the deadlines of some refrigerants' prohibition. Besides, Figure 20 gives some extra information about the refrigerants affected by the protocols. [15]

Table 1 – Placing on the market prohibitions by the EU Regulation No 517/2014 (Regulation (EU) No 517/20, 2014).

Products and equipment		Date[a]
Domestic refrigerators and freezers that contain HFCs with GWP ≥ 150.		2015
Refrigerators and freezers for commercial use (hermetically sealed equipment) that contain HFCs with	GWP ≥ 2500.	2020
	GWP ≥ 150.	2022
Stationary refrigeration equipment, that contains, or whose functioning relies upon, HFCs with GWP ≥ 2500 except equipment intended for application designed to cool products to temperatures below −50 °C.		2020
Multipack centralized refrigeration systems for commercial use with a rated capacity ≥ 40 kW that contain, or whose functioning relies upon, fluorinated greenhouse gases with GWP ≥ 150, except in the primary refrigerant circuit of cascade systems where fluorinated greenhouse gases with a GWP < 1500 may be used.		2022
Movable room air-conditioning equipment (hermetically sealed equipment which is movable between rooms by the end user) that contain HFCs with GWP ≥ 150.		2020
Single split air-conditioning systems containing less than 3 kg of fluorinated greenhouse gases, that contain, or whose functioning relies upon, fluorinated greenhouse gases with GWP ≥ 750.		2025

[a] 1 January.

Figure 19-Refrigerants' Regulation

Table 2 — Most commonly refrigerants affected.

Refrigerant (GWP)	Applications	GWP
R134a (1430)	- Domestic refrigerators and freezers	150
	- Refrigerators for commercial use (hermetically sealed equipment)	150
	- Stationary refrigeration equipment (medium evaporation temperature)	2500
	- Multipack centralized refrigeration systems for commercial use (medium evaporation temperature)	150
	- Primary refrigerant circuit of cascade systems	1500
R404A (3922)	- Refrigerators and freezers for commercial use (hermetically sealed equipment)	150
	- Stationary refrigeration equipment (low and medium evaporation temperature)	2500
	- Multipack centralized refrigeration systems for commercial use (medium and low evaporation temperature)	150
R410A (2088)	- Stationary refrigeration equipment (chillers)	2500
	- Movable room air-conditioning equipment	150
	- Single split air-conditioning systems	750

Figure 20-Commonly used refrigerants affected by the Normative.

Besides this, The Montreal Protocol governs the use of refrigerants gases and blowing agents that affect the Ozone layer. This protocol has been adjusted for several years. This project aims to assess the refrigerants currently used in refrigeration systems, especially in rooftop heat pumps, explaining the impact both in the operation and in the environment. [20]

Thus, it is explaining the refrigerants generally used in the industry and they are proposed in the project after a rigorous study.

Therefore, a comparison between different trademarks is made to see which refrigerant they are using in air-to-air reversible rooftop heat pumps. Table 1 displays the most commonly used refrigerants depending on the power of some companies.

Trademarks	<100 kW	>100 kW
LENNOX	R-410A / R-32	R-410A / R-32
CARRIER	R-410A	R-410A
TRANE	R-410A	R-410A
CLIVET	R-410A	-
ETT	R-410A	R-410A
HITECSA	R-410A / R-407C	-
IMBAT	R-410A	R-410A
KEYTER	R-410A	-
YORK	R-410A	-
DAIKIN	R-134a / R-32	R-134a / R-32

Table 1-Refrigerants used by some trademarks

Those are several of the most demanded trademarks nowadays in refrigeration systems. It is comparing the refrigerants used by their rooftop heat pumps. Can be seen that the R-410A is the most frequent refrigerant, but the R-32 is starting to get more opportunities due to the benefits that provides to the system and environment.

It is doing profound research about several refrigerants, exposing their benefits, downsides and systems in which they are used more often. The purpose of the study is to select the most beneficial refrigerant for the system that is pretending to design.

Table 2 shows the results obtained. [21]

REFRIGERANTS	BENEFITS	DOWNSIDES	NORMAL USE
R-717(Ammonia)	-Affordable -Efficient heat transfer -GWP=0 -High refrigeration capacity -Water soluble	-Toxic -Restricted in certain systems -High price of the system -Flammable in certain conditions.	-Cascade systems (NH_3/CO_2) -Thermal storage systems -Small charges and commercial refrigeration
R-744 (CO_2)	-Non flammable -Odourless -High performance -Affordable -High availability -Low energy consumption -Class A1	-Operates at high pressures and temperatures - High price of the system -Not leak detection	- Cascade systems (NH_3/CO_2). -Commercial refrigeration -Packaged systems -Water heat pumps
R-600a (Isobutane) R-290 (Propane) R-1270 (Propylene)	-Nontoxic -Efficient -Affordable - Good compatibility with materials	-High flammability -Additional security costs -Higher cost in the compressor - Required anti-leakage system	-Small charges -Chillers with security systems -Cascade systems -Secondary circuits
R-32	-Low liquid density -High yield -Easy installation -Energy saving	-Operates at higher pressures than R-410A -Flammable in certain conditions -Required new safety equipment -High GWP (677)	-HVAC&R equipment - Residential split-system heat pump -Rooftop heat pumps
R-22	-Class A1 -High performance	-Very high GWP (1760) -High drop of COP with the increase of ambient temperature. -High pressure discharge	- HVAC&R equipment -Split and rooftop heat pumps

R-134a	-Very high performance -Class A1 -ODP=0 -High heat transfer	-Very high GWP (1300) -Absorbs a lot of moisture	-Automotive air-conditioning and chillers. -Thermal storage systems -Heat pumps
R-466	-Easier compatible replacement for R-410A. -Non flammable -Class A1 - Operates at lower pressures than R-410A	-Slightly lower efficiency than R-410A -High liquid density -Low efficiency when it is overcharged. -High GWP (697) -Toxic component	-HVAC&R equipment -Residential split-system heat pump
R-452B	-Similar performance to R-410A -Operates at lower pressures than R-410A -Low liquid density	-Flammable in certain conditions -Required new safety equipment	-HVAC&R equipment - Residential split-system heat pump
R-454B	-Operates at lower pressures than R-410A -Low liquid density	-Flammable in certain conditions -Required new safety equipment	-HVAC&R equipment - Residential split-system heat pump
R-1234yf **R-1234ze(E)**	-GWP <1 -Potential replacement for R-134a -Nontoxic -Similar	-Reduced capacity - Flammable in certain conditions -High cost -Less COP and power than R-134a	-Automotive air-conditioning and chillers.
R-407C	-Similar pressures to R-22. -ODP=0 -Suitable replacement for R-22 -Class A1	-Very high GWP (1774) -Difficulties in compatibility with some equipment.	- HVAC&R equipment -Rooftop heat pumps. -Thermal storage systems
R-410A	-Very high performance -Class A1 -Low discharge pressure -ODP=0	-Very high GWP (1924) -Works at high pressures. -Need for high pressure equipment	-HVAC&R equipment -Residential, unitary and VRF equipment as well as small chillers. -Split and rooftop heat pumps -Thermal storage systems

Table 2-Research about the characteristics of the most used refrigerants

Those are the most used refrigerants nowadays in refrigeration. It is time to choose which one will be more suitable for the system that is designing. First of all, it is needed to dismiss those refrigerants which cannot operate in heat pumps. Then, discard those that do not produce enough COP due to the high power demand.

Dealing with R-454B and R-452B would mean a significant increase in costs. There are some positive points on them, such as energy saving or the high performance. However, the increase of price in the system and the compliance of all of the regulations would be a challenge. R-454B produces lower GWP emissions than R-32, but its working pressures are noticeably below. Even though this refrigerant works with the same principles, it provides higher isentropic efficiency of the compressor and COP than R-410A and R-32, only with a slighter difference, but higher. By contrast, the R-32 has the highest volumetric heat efficiency. It means it needs a lower volumetric flow rate than the other fluids to exchange the same thermal power. Figure 21, Figure 22 and Figure 23 show a comparison between R-32, R-454B and R-410A in which appears the isentropic efficiencies, COP and VHE depending on the user outlet temperature. [22]

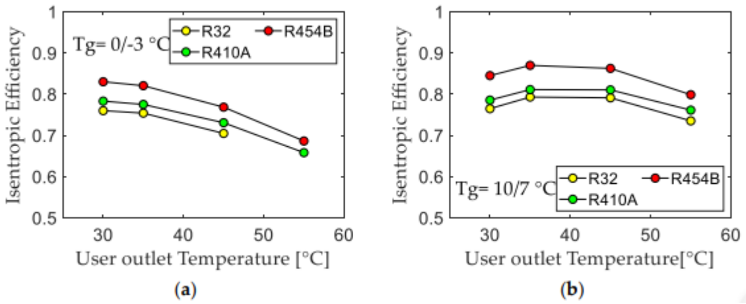


Figure 21-Isentropic efficiency comparison

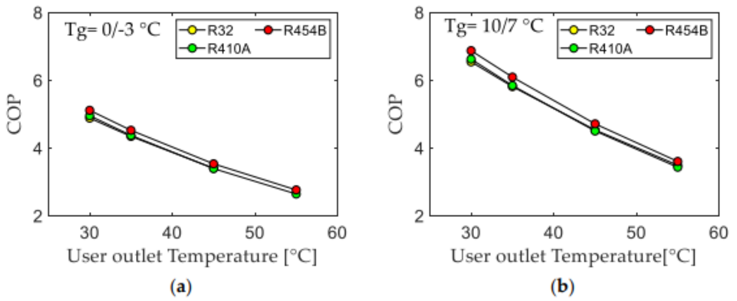


Figure 22-COP comparison

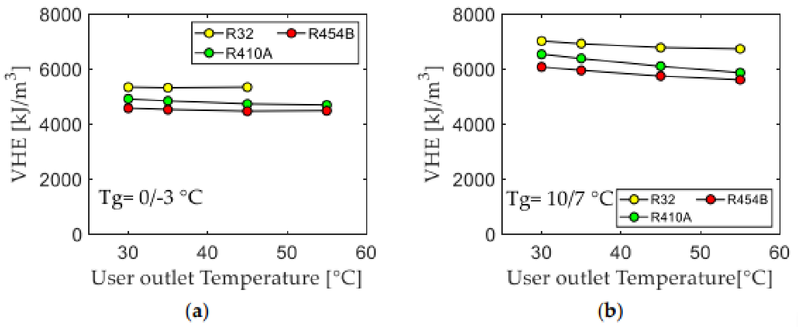


Figure 23-VHE comparison

Thus, the R-454B is a potential refrigerant in terms of replacing the R-410A because of its lower GWP level, but it delivers less capacity. Therefore, in terms of high power and a high charge of fluid, it would not be the option in this project. The R-452B is the closest *drop-in* equivalent to the R-410A in terms of capacity and efficiency. Furthermore, the R-32 would be a better choice in this case due to R-452B, R-454B are patented, and their used is restricted. [23]

Consequently, the remaining refrigerants are required to evaluate certain aspects. The R-22, due to its significant contribution to Global Warming, is dismissing more and more every day, without forgetting that it is one of the most pointed refrigerants in the Montreal Protocol to be erased. In the new systems, that refrigerant is no longer in demand, even though it provides excellent performance results. The R-407C has been the most suitable replacement for the last mentioned refrigerants. It presents undoubtedly great features. Notable COP provided, no chlorine in its composition, which means no damage to the Ozone Layer and by the ASHRAE standard 34 classified it as an A1 class. However, this refrigerant presents some problems combining it with some equipment. An incompatible matching between the fluid and the system would generate huge losses and even a dangerous situation. In addition, it presents an excess of high GWP. Thus, this one would not be a suitable option because other fluids can provide the same performance and not have problems with compatibility.[24]

Refrigerants R-1234yz and R-1234ze are refrigerants with very trustworthy market projections. Its use in the automotive sector has increased considerably due to its high performance in small equipment. They have the disadvantage that their capacity is not very high for big systems and belong to group A2L. Despite the fact they are high-projection refrigerants with very low GWP (around 4 and 6), they will not be used in this design due to their high costs. [15]

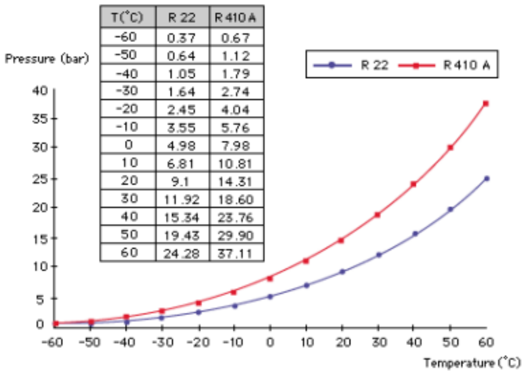
R-134a is a fabulous refrigerant that provides exceptional results. It is easy to manage, belonging to the A1 class. It is not composed of chlorine, so ODP=0. Nevertheless, it produces high levels of GWP and absorbs a large quantity of moisture during its action. Thus, it is finding a more suitable replacement nowadays that fulfil the conditions of this refrigerant. [11]

Finally, there are three potential refrigerants left: R-410A, R-32 and R-466A. R-410A is a mixture of R-125 and R-32. It belongs the group A1. Mixture stable, with a low glide of temperature and low toxicity. Even though it is composed of R-32, it is non-flammable. Now it is used in the new air-conditioning equipment. The R-410A attributes answer perfectly well to the air-conditioning equipment efficiency. It is possibly one of the best alternatives to the R-22 because it works at higher pressures. [25]

R-32 is a refrigerant type A2L, which means it is practically non-flammable. Its working pressures are slightly superior to the R-410A as well as the volumetric capacity. The companies Daikin and Haier are starting to use this refrigerant in their new equipment such as rooftops, chillers, splits, etc. [26], [27]

R-466A is a refrigerant blend developed with thermodynamic properties that potentially make it a compatible design replacement for R-410A. This new blend offers a 65% reduction in GWP over R-410A and is class A1. Refrigerant composed of R32 (49%), R125 (11.5%) and CF3I (39.5%). This fluid can operate at slightly low pressure than R-410A. The great drawback of this refrigerant is the higher liquid density because it contains a notable quantity of CF3I, which is toxic. Hence, it means that the optimum refrigerant charge of R-466A in the system will likely be larger than R-410A, which would increase costs. The efficiency of R-410A and R-466A are nearly the same. [26]

R-410 is the fluid that provides one of the best performances as well as compatibility with the equipment. It is accessible and affordable, from which an excellent profit is taken. This fluid is easy to handle due to it is secure, non-toxic and non-flammable. In Figure 24 can be seen that the pressure in which the R-410A is working higher than R-22 for almost every temperature. [28]



T(°C)	R 22	R410 A
-60	0.37	0.67
-50	0.64	1.12
-40	1.05	1.79
-30	1.64	2.74
-20	2.45	4.04
-10	3.55	5.76
0	4.98	7.98
10	6.81	10.81
20	9.1	14.31
30	11.92	18.60
40	15.34	23.76
50	19.43	29.90
60	24.28	37.11

Figure 24-Working pressures comparison between R-22 and R-410A

By the F-gas: Regulation (EU) 517/2014 by the European Parliament and Council of 16th of April 2014, the aim is reducing totally or at least almost every fluorinated gas with high GWP. Besides, reduction of availability and prohibition of HFCs (with high GWP) recharge in present installations. It means that the fluid R-410A must be reduced as far as possible to comply with the regulation. In addition, the lack of availability is producing an increase in the price of this refrigerant. It means that it would be not cost-effective in the following years. In addition, it is remarkable to consider the taxes of using this refrigerant, which is 0.02·GWP €/kg and more taxes with more GWP level [15]. F-gas regulation does not directly affect the rooftop heat pumps in terms of the refrigerant R-410A, but it bans the use of this refrigerant in scroll chillers by the end of 2024 and heat pumps, splits and window units by the end of 2025. [23]

All the information commented above proves that the use of the R-410A will no longer be considered a potential refrigerant in this research. Refrigerants type "drop-in" let a direct substitution in the existent A/C equipment with lower GWP.

2.4. Refrigerant selection

A possible alternative to it proposed for small systems has been R32, which could be also a good option for larger systems if safety issues are addressed properly as this refrigerant is classified as A2L. Therefore, this refrigerant will be the selected one in order to design the unit.

For using this fluid, it is necessary to comply with certain regulations. They are The Safety Regulations for Refrigeration Installations and the RSIF RD 138/2011 due to this installation has equipment with >= 0.5 kg for refrigerants belonging group L2. Those normative instructions stipulate that those are level 2 installations, for which they must be fixed by a company of this level. In addition, before any service, the documentation required by Art. 21 of the RSIF must be submitted to the appropriate organ of the Autonomous Community. [29] The flammability of this fluid carries with the need of restrictions where systems can be installed or the maximum charge that can be used. It also means components need to be compatible with the refrigerant to ensure they cannot be considered an ignition source. Nevertheless, it is becoming to use more and more to supply large spaces such supermarkets or factories, safer places to use A2L refrigerants. [30]

Nowadays, technology and components suitable for this kind of fluids are well developed and have been available on the market since 2018. As Danfoss Company points out: *R-32 is a popular choice, due to its availability, performance and efficiency*. Daiking also is making efforts to include R-32 machines in the market. The easy manipulation of the fluid is one of the most encouraging characteristics by the maintenance team due to the possibly of dealing with fluid in both gaseous and liquid phase easily.

Thus, the fluid R-32 is going to run this system because it is a single component refrigerant, which means it is much easier to recycle or reuse. It is relatively simple to handle, moderately affordable to produce. Despite the fact it is a bit flammable, is difficult to ignite and even less explode. [31] The components are becoming more affordable due to the demand increase. It is a refrigerant in expansion and there are many companies making significant progresses on their research making the most of this refrigerant even for high power and extreme conditions.

3. DESIGN AND SELECTION OF COMPONENTS

3.1. Compressor

The purpose is to choose an efficient, reliable and comfortable compressor to run the system is designing. It is studying the different compression types to adjust them better to the required power.

Positive displacement compressors are the most suitable type for heat pumps. The main features are achieving a high compression, no restrictions on the operation due to the range of suction pressure and the lack of continuity in transporting the refrigerant. The most positive displacement compressors used are reciprocating and scroll compressors. [32]

Traditionally, rooftop heat pump system used reciprocating compressors. That type of compressors provides a large number of benefits. Especially, the low energy consumption, the low noise and the high evaporating temperatures are some of the advantages of those compressors. They adjust automatically when the plant does not need to run continuously. They are considered remarkably reliable, particularly on smaller machines. Nowadays, the tendency for applications such as rooftops with the performance that is looking for in this project, scroll compressors would be a better choice. Not only is it a significantly reliable device but also fewer costs in maintenance due to the reduced number of moving parts. They present a high service life, low vibration, very resistant and compact. [32]

Therefore, the type of compressor for the rooftop heat pump designed in this project would be a scroll compressor.

Modulating compressor technologies adjust capacity either in stages (65%, 100%) or in a continuous fashion (10-100%) for applications with varying loads or changing ambient conditions. This technology provides whit significantly better results for air conditioning applications. [33]

There are four types of compressor modulation technologies:

1. Multiple Compressors
2. Two-stage
3. Continuous
4. Variable Speed

3.1.1. Type of modulation

Figure 25 shows the customer criteria at the time of initial purchase. It is visible that the preferences about efficiency predominate above the initial investment or superior comfort. It is because there is an inclination to long-term benefits than profitability in a few years of use. [33]

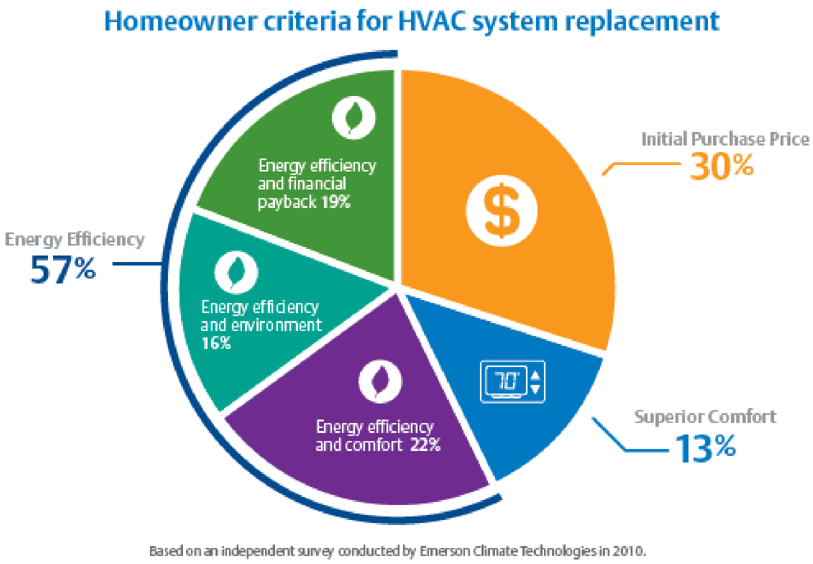


Figure 25-Customer criteria for HVAC initial purchase

Now, it is time to choose the kind of modulation for the project, considering the system is a high power rooftop. Thus, having researched the different types of modulation, their features, how they work and their benefits, finally, the choice is selecting multiple compressors. As it can be seen in the comparison shown in Figure 26, multiple compressors let numerous advantages to the desired system, even arriving at great power. [33]

Modulation Technology	Range	Ideal Applications	Part Load Efficiency	Full Load Efficiency	Comfort	Applied Cost
Two-stage	2-5HP	Light commercial split & package applications	High	High	Medium	Best
Continuous	3-15HP	Light commercial split, package and chiller applications	Low	High	High	Better
Variable Speed	3-10HP	Light commercial rooftop and chiller applications	Highest	Medium	Highest	Good
Multiples	3-120HP	Commercial split, rooftop and chiller applications	High	High	High	Best

Figure 26-Compressors' modulation comparison

Therefore, it is explaining more extensive how works multiple compressors systems due to it are going to take part in the system that is designing.

Multiple compressors system consists of multiple compressors connected in tandems or trios. They are found often in commercial air conditioning applications, such as larger commercial chiller systems, like museums, factories, offices, etc. With multiple compressors, they can operate individually or together, delivering different stages of capacity depending on the demand, maintaining high a high Energy Efficiency Ratings (EER). These systems are especially interesting for rooftops. Figure 27 shows more information about the characteristics of these configurations. [33]

Advantages of multiple compressor modulation

Modulation Capability	Staged capacity modulation for precision load matching
Efficiency	High system efficiency at both full-load and part-load
Flexibility	Versatile compressor combinations
Oil Management	No extra oil management hardware needed
Reliability	Designed and rigorously qualified for long life

Figure 27-Further information about multiple compressors' characteristics

This system provides high system efficiency in both full and part-load phases. Besides, it lets plenty of comfort levels as well as combinations. Nevertheless, most of all, these systems are designed for long life and qualified for extensive periods. [33]

Multiple compressors can be used in packaged rooftop systems, split systems, or chillers to provide stepped modulation for optimal load matching. Figure 28 shows a simple scheme of multiple compressors in tandem configuration. This type of modulation enables original equipment manufacturers to boost system part-load efficiency levels, as well as the ability to meet new energy standards and regulations. [33]

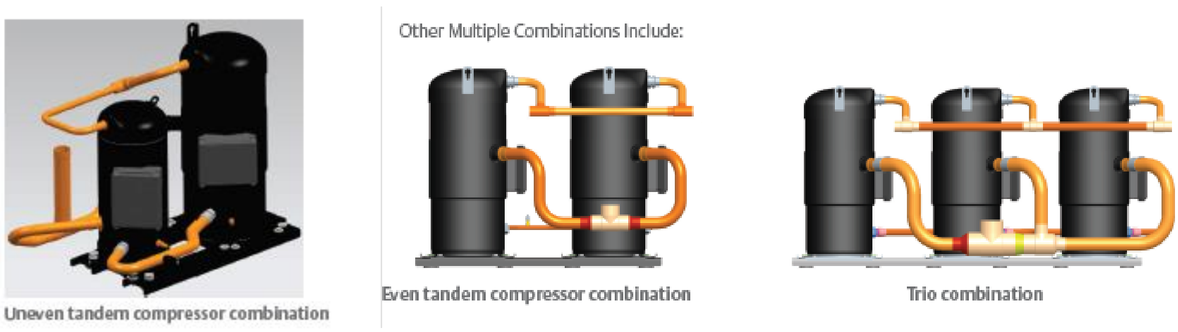


Figure 28-Tandem compressor configuration

3.1.2. Design working temperatures

This point consists of selecting the design temperatures of the system. It is considering that the system would have as the maximum running time per year, 8000 hours. It is needed to choose the appropriate temperatures for acceptable performance for the high power required. The temperatures that are choosing are based on typical working conditions. Table 3 shows the working points of the system:

	Temperatures
Tin,air.evap	27 ºC
Tout,cond	35 ºC
Tcond	55,8 ºC
Tevap	7 ºC
SC	0
SH	10 ºK
ΔT,cond	9 ºK
ΔT,evap	10.5 ºK

Table 3-Initial system values

3.1.3. Compressor selection

Once the temperatures are known, it is time to designate the compressor and comment on its hallmarks. Then, with the selected values, put all the compressors parameters in the IMST-ART software. It will be introduced in the software every parameter that the program asks. Figure 29 shows what kind of values must be introduced to complete the compressor's information.

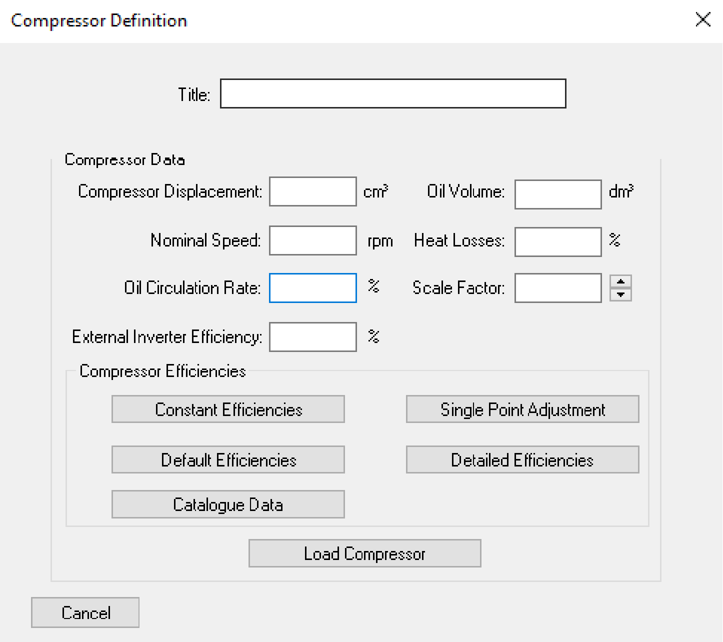


Figure 29-Compressor data to be filled in IMST-ART

As mentioned in the introduction, the most used compressors for these systems are the positive displacement compressors. The characteristics they provide are what this project needs. Regarding the number of possibilities, scroll compressors are going to be the selected type for the system.

Following that, the next step is to introduce the provided values of the compressor in its working cycle per range of temperatures. Inside the "Catalogue Data" tab (visible in Figure 30), there are two different tables, Cooling Capacity (kW) and Compressor Power (kW). These tables must be filled by the values given by the catalogue or given in a simulation run by the compressor's supplier. The tables mentioned previously are the ones in Figure 30.

For the compressor's selection, it is considering a tandem configuration of these two compressors obtained from Blitzer: **GSD60137VL** and **GSD80295VL**. Having all the information about these compressors from the supplier, it is submitting the potential data to the software. The data that must be completed, is indicated in Figure 31 and Figure 32 according to the first compressor, and the Figure 33 and Figure 34 according to the other.

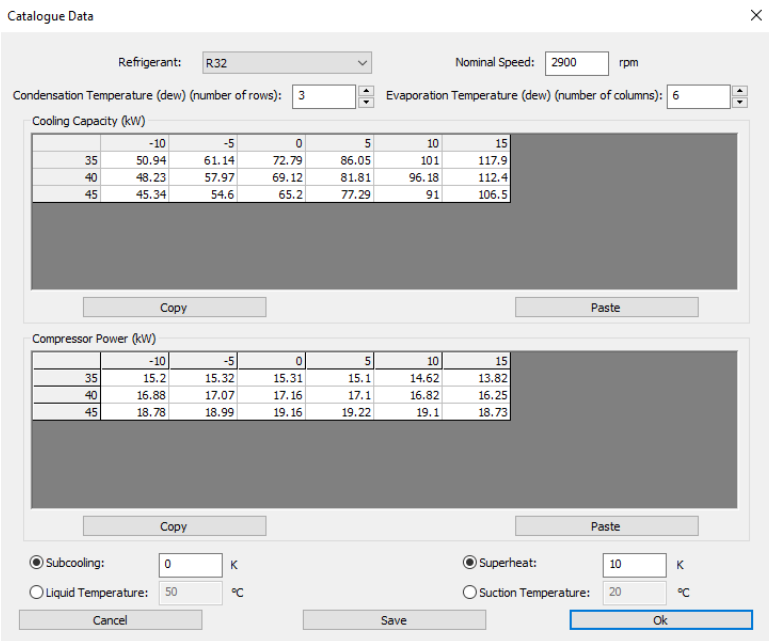


Figure 30-Catalogue data of the compressor

It was decided that in the compressors' catalogue data, in Cooling Capacity and Compressor Power tables, the range of temperatures would be between 30 to 50 ºC. These temperatures are an acceptable work range due to the working time of the desired compressor around those temperatures.

The most important factors to consider are the compressor displacement, the nominal speed and the oil volume. The rest of the information is almost similar in every compressor.

Once both compressors data have been submitted to the software. This method will result in the tandem arrangement, in parallel mode.

- **GSD60137VL**

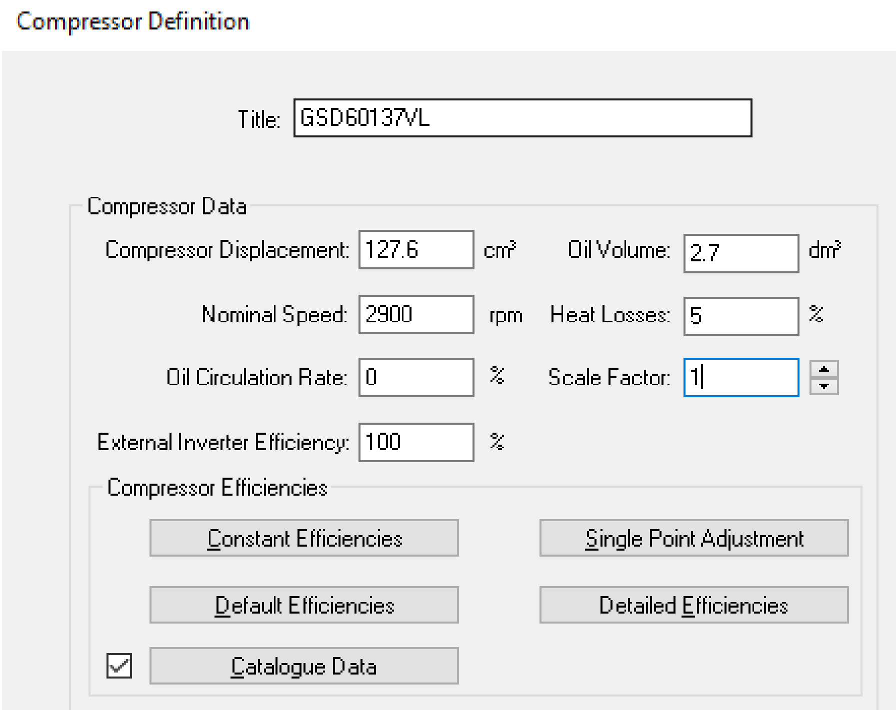
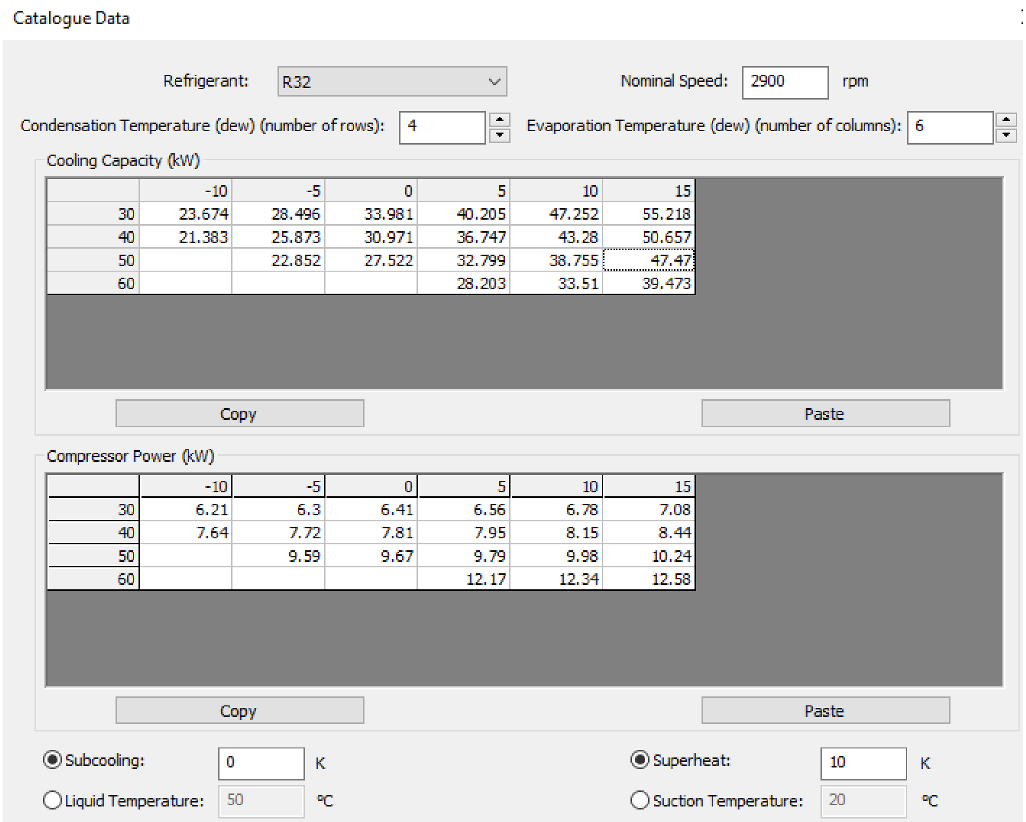


Figure 31-GSD60137VL initial data



Cooling Capacity (kW)

	-10	-5	0	5	10	15
30	23.674	28.496	33.981	40.205	47.252	55.218
40	21.383	25.873	30.971	36.747	43.28	50.657
50		22.852	27.522	32.799	38.755	47.47
60				28.203	33.51	39.473

Compressor Power (kW)

	-10	-5	0	5	10	15
30	6.21	6.3	6.41	6.56	6.78	7.08
40	7.64	7.72	7.81	7.95	8.15	8.44
50		9.59	9.67	9.79	9.98	10.24
60				12.17	12.34	12.58

Figure 32-GSD30137Vl Catalogue Data

- **GSD80295VL**

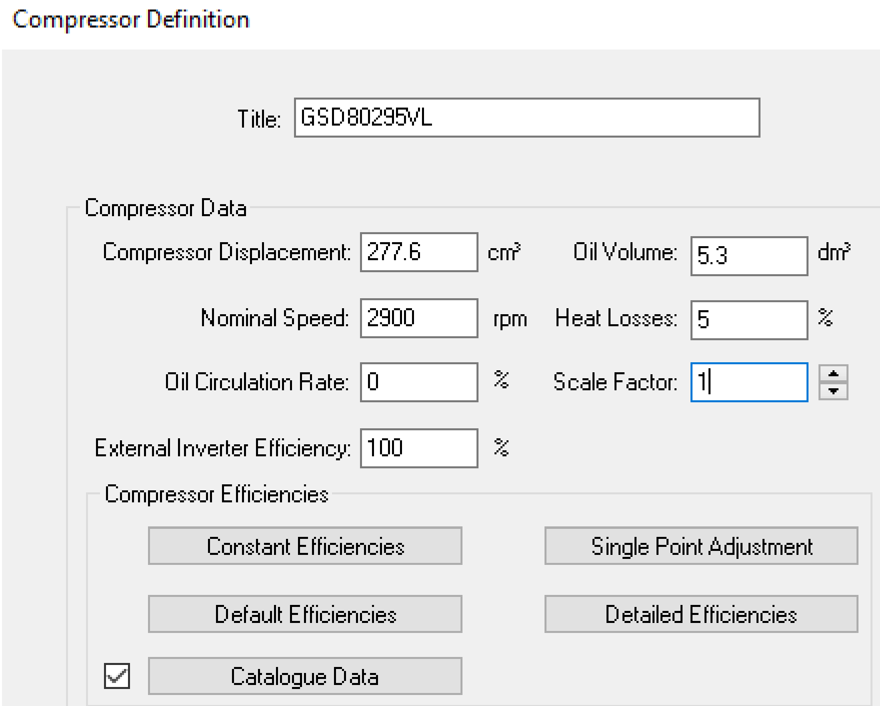
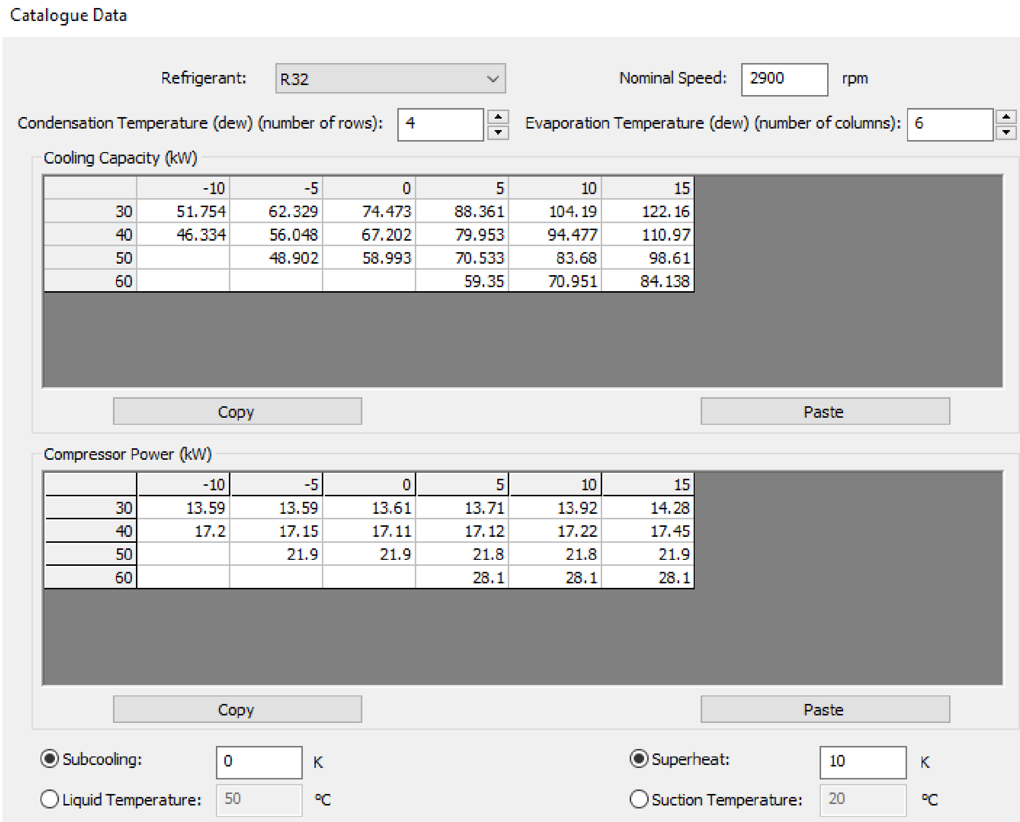


Figure 33-GSD80295VL Initial data



Cooling Capacity (kW)

	-10	-5	0	5	10	15
30	51.754	62.329	74.473	88.361	104.19	122.16
40	46.334	56.048	67.202	79.953	94.477	110.97
50		48.902	58.993	70.533	83.68	98.61
60				59.35	70.951	84.138

Compressor Power (kW)

	-10	-5	0	5	10	15
30	13.59	13.59	13.61	13.71	13.92	14.28
40	17.2	17.15	17.11	17.12	17.22	17.45
50		21.9	21.9	21.8	21.8	21.9
60				28.1	28.1	28.1

Figure 34-GSD80295VL Catalogue Data

Then it will be created a new compressor called "Tandem" will be the sum of the two compressors. Thus, the program will run within this configuration to obtain the proper results.

After that, it will be always talking about the tandem configuration. All the results will refer to the tandem, which parameters are shown in Figure 35 and Figure 36.

- **Tandem**

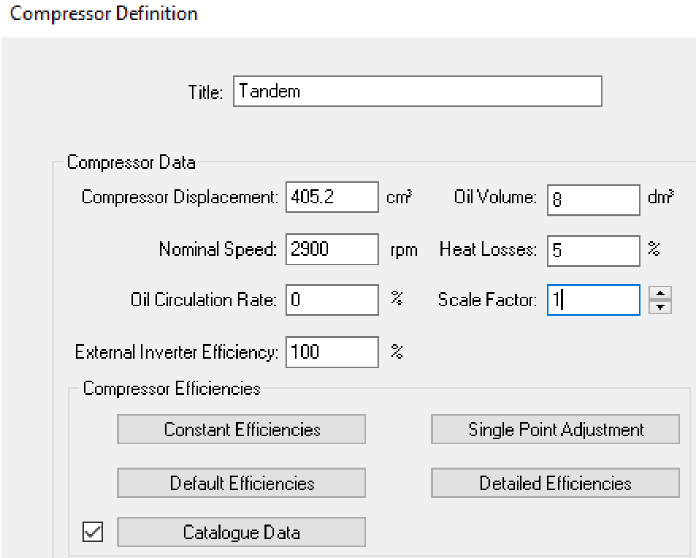
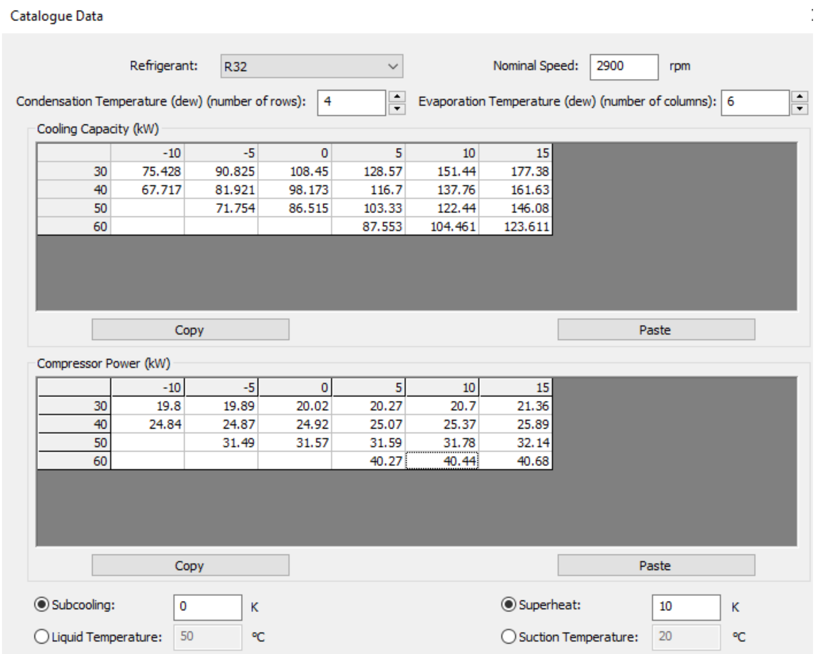


Figure 35-Tandem initial values



Cooling Capacity (kW)

	-10	-5	0	5	10	15
30	75.428	90.825	108.45	128.57	151.44	177.38
40	67.717	81.921	98.173	116.7	137.76	161.63
50		71.754	86.515	103.33	122.44	146.08
60				87.553	104.461	123.611

Compressor Power (kW)

	-10	-5	0	5	10	15
30	19.8	19.89	20.02	20.27	20.7	21.36
40	24.84	24.87	24.92	25.07	25.37	25.89
50		31.49	31.57	31.59	31.78	32.14
60				40.27	40.44	40.68

Figure 36-Tandem Catalogue Data

Therefore, all the information needed about the compressor is known. The following step will be the design of the batteries. The compressor will be the link between the heat exchangers, compressing from low-pressure until high-pressure fluid. The batteries' design is crucial. At the compressor's inlet is major that the fluid will be in the gaseous state. Not only it will increase the device's lifespan but also a higher efficiency of the system.

3.2. Evaporator and condenser design

Hence, introduce heat exchangers information will be the next move. It is required to insert the evaporator and condenser primary values. The information needed is indicated in Figure 37 referring to Condenser, and Figure 38 referring to the Evaporator. The only difference between them is the temperatures, but the gaps are the same.

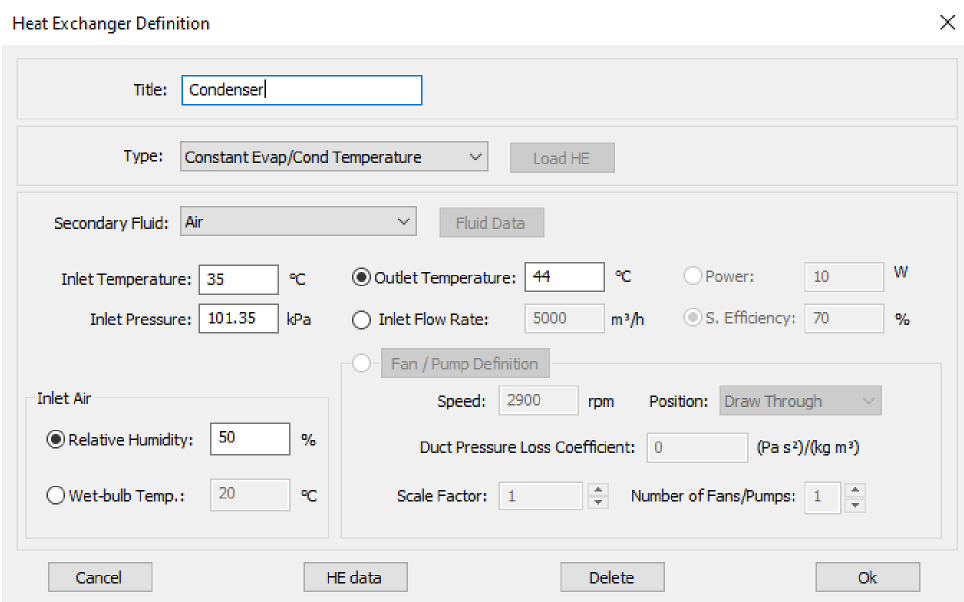


Figure 37-Condenser data

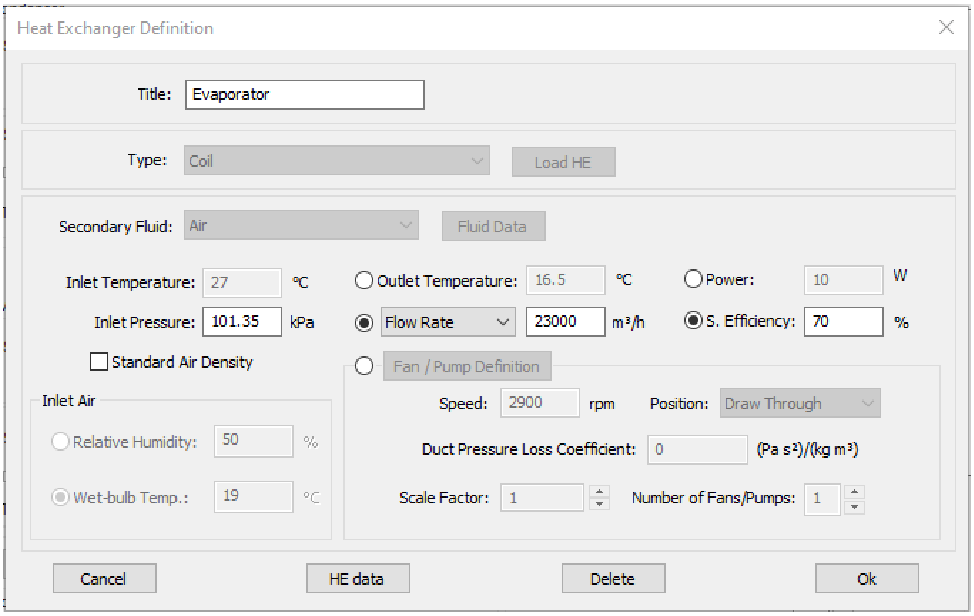


Figure 38-Evaporator data

Then, there is enough data to run the simulation. This software provides plenty of information, which lets to design the evaporator and condenser according to the values given. The most important contents to consider are the capacity and mass flow of each heat exchanger.

3.2.1. Areas of the batteries

Firstly, finding the area of each heat exchanger is required. It starts by considering the Global Coefficient of Heat Transfer. It must be known that it depends on the kind of the heat exchanger and the fluids circulating through it. In this case, the air is the fluid in both ones.

$$U_{evap} = U_{cond} = \mathbf{40} \frac{\boldsymbol{W}}{\boldsymbol{K \cdot m^2}}$$

Then, the LMTD must be calculated. It consists of an operation between the differences in inlet temperatures of the fluids through the exchanger.

3.2.1.1. Condenser LMTD

- T7= 55.8 ºC
- T6= 55.8 ºC
- Ta,in = 35ºC
- T,a,out = 44ºC
- ΔT in the air = 9ºC

It is considering a constant temperature through the condenser.

$$LTMD_{cond} = \frac{(T7 - Ta,in) - (T6 - Ta,out)}{(\ln\left(\frac{(T7 - Ta,in)}{(T6 - Ta,out)}\right)} = 15.87\ K \quad (1)$$

3.2.1.2. Evaporator LMTD

- T2= 7 ºC
- T1 = 7ºC
- Ta,in = 27ºC
- T,a,out = 16.5 ºC
- ΔT in the air = 10.5ºC

It is considering a constant temperature through the evaporator.

$$LTMD_{evap} = \frac{(T2 - Ta,in) - (T1 - Ta,out)}{(\ln\left(\frac{(T2 - Ta,in)}{(T1 - Ta,out)}\right)} = 14.10\ K \quad (2)$$

3.2.1.3. Calculation of the heat exchangers' areas

From the previous values, it is possible to substitute in the following formula and obtain the area of each heat exchanger.

$$Q = U \cdot A \cdot LMTD \quad (3)$$

- Q→ Capacity [W]
- U→ Global coefficient of heat transfer [W/K· m^2]
- LMTD → Logarithmic Mean Temperature Difference [K]
- A → Heat exchange area [m^2]

Before considering any result as valid, it is remarkable to know there will be several modifications in the system to obtain a proper system. These modifications can be in a few places, but especially on the heat exchangers design. The circuits can be modified to increase efficiency, reducing the area with fewer tubes or tube's rows.

Must be considered that the results shall be reasonable, with not a dangerous pressure drop or a high discharge temperature. Thus, the heat exchangers' circuits must be designed accordingly to a sustainable, affordable and efficient solution. Table 4 indicates the area results after substituting (1) and (2) respectively.

	Condenser	**Evaporator**
Q [kW]	147.97	115.36
U [W/K m^2]	40	40
LMTD [K]	15.8771	14.10455
A [m^2]	**232.99**	**204.47**

Table 4-Final areas of the batteries

Therefore, the following step is obtaining the **frontal area** (A_f) of the heat exchanger and from that, the design of them.

3.2.1.3.1. Frontal area of the heat exchangers

$$A_f = \frac{\dot{V}}{u} \qquad (4)$$

- **$\dot{V}$ = volume flow [m^3/h]**

$$\dot{V}\,(\boldsymbol{condenser}) = \frac{Q}{(Ta,out - Ta,in)\cdot\rho(air)}\cdot 3600 \qquad (5)$$

$$= \frac{147.97}{(44-35)\cdot 1.2}\cdot 3600 = \boldsymbol{49323.33\ m^3/h}$$

$$\dot{V}\,(\boldsymbol{evaporator}) = \frac{Q\cdot SHR}{(Ta,in - Ta,out)\cdot\rho(air)}\cdot 3600 \qquad (6)$$

$$= \frac{115.36\cdot 0.69}{(27-16.5)\cdot 1.2}\cdot 3600 = \boldsymbol{22742.4\ m^3/h}$$

0.69 multiplies the evaporator because the Sensible Heat Ratio (SHR) is 69 % accordingly to the IMST-art. It means that the rest of the heat is lost. Then, substituting in (5) and (6) in (4):

- **u = air velocity [m/s]**→Between 2-3

$$\boldsymbol{A_f Condenser} = \frac{49323.33/3600}{2} = \boldsymbol{6.85\ m^2}$$

$$\boldsymbol{A_f Evaporator} = \frac{22742.4\ /3600}{2.3} = \boldsymbol{3.15\ m^2}$$

Table 5 shows the final areas of each exchanger:

Area [m2]	Condenser	Evaporator
Frontal	6.87	3.15

Table 5-Total areas of the batteries

Then, it is time to do the geometric design from the manufacturer's data. Suppliers provide some standards in measures like the fin perforation, fin separation or tube diameter. These dimensions are standard measures, so they can be modified a bit, but not too much.

3.2.1.4. Geometric design of the batteries

The type of heat exchanger that is going to design is coils-type. It is the most recommendable design in these kinds of applications and most of the companies related to rooftop heat pumps use those heat exchangers.

Then, the following data is given:

- Horizontal distance between fins (Sl) = 27.48 mm
- Vertical distance between fins (St) = 31.75 mm
- Tube diameter (Dc) = 0.5’’
- Fins thickness (tf) = 0.1 mm
- Compactness (β) = 717 m^{-1}

From this point, it is possible to estimate the heat exchanger dimensions. The following step is calculating the number of rows and tubes of the condenser and evaporator and their respective dimensions and volumes.

➢ Volume of the heat exchanger (V)

$$V = \frac{A}{\beta} \qquad (7)$$

➢ Number of rows (Nrows)

$$N_{rows} = \frac{V}{Sl \cdot Af} \qquad (8)$$

➢ Height (T) and Width (L)

$$\frac{H}{W} = 0.75 \qquad (9)$$

$$H \cdot W = Af \qquad (10)$$

- Number of tubes per row (Nt)

$$Nt = \frac{H}{St} \qquad (11)$$

Primary dimensions for both heat exchangers are shown in ***¡Error! No se encuentra el origen de la referencia.***. To facilitate the explanation of the arrangement of the exchanger tubes, IMST-ART provides Figure 39, in which the exchanger structure is evident. That allows a more apparent concept of the distribution and distances between the tubes.

	Evaporator	**Condenser**
V [m^3]	0.28	0.32
Nrows	3.77	1.7
H (Height) [m]	1.44	2.23
W (Width) [m]	1.92	2.97
Nt	45.38	70.38

Table 6-Primary dimensions of the batteries

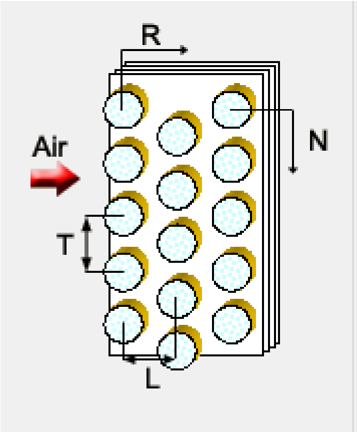


Figure 39-Heat exchanger structure by IMST-ART

3.2.1.5. Circuits design

Thus, it is time of introducing all the data in the IMST-art software and see the results. Then, it is considering making some modifications depending on the results wanted to achieve.

Therefore, it is introducing the flow rate, the spacing between fins, width of the exchanger, the diameter and thickness of the tube. Besides this, the data for the circuit set up: number of rows, number of tubes per row and the number of the circuits. Those circuits are going to be U-pattern type, with counter-current flow arrangement to have a simpler and higher efficient system. Some of the information about the tubes distribution is displayed in Table 7.

Advantages	Disadvantages
The more rows the more efficient is the heat exchanger.	The costs increase as more rows are defined.
With more area, increasing the number of tubes is possible, which means more efficiency.	The increase of the area means a higher occupancy of the heat exchanger as well as the increase of costs.

Table 7-Advantages and disadvantages regarding the tubes distribution

The purpose is to find a well-balanced design between costs and efficiency. A plan that provides excellent outcomes, without taking up a substantial amount of space or increase expenses by materials, form or consumption.

In this case, it is trying to reach a design with a COP that fulfils the regulation requirements, not needed a super high one. Principally, the goal is to make this plan as cost-effective as possible, considering it is a tremendous plan working with an elevated amount of power that supplies a considerable space. Then, Figure 40 and Figure 42 are the following results obtained from IMST-art. These conclusions are provided after certain modifications to make the device as safer as possible regarding the pressure drops and the discharge temperatures. Also, the idea is making it considerably efficient, so the number of circuits and tubes have been changed respecting the minimum number of them. All of the adjustments have been made to obtain the most efficient and safer device reachable with the initial data.
The structure of the frontal area of the condenser and evaporator are represented in the Figure 40, Figure 41 and Figure 42, Figure 43, respectively. Once the space between tubs is known, then the real dimensions of the batterie can be estimated.

- **Condenser**

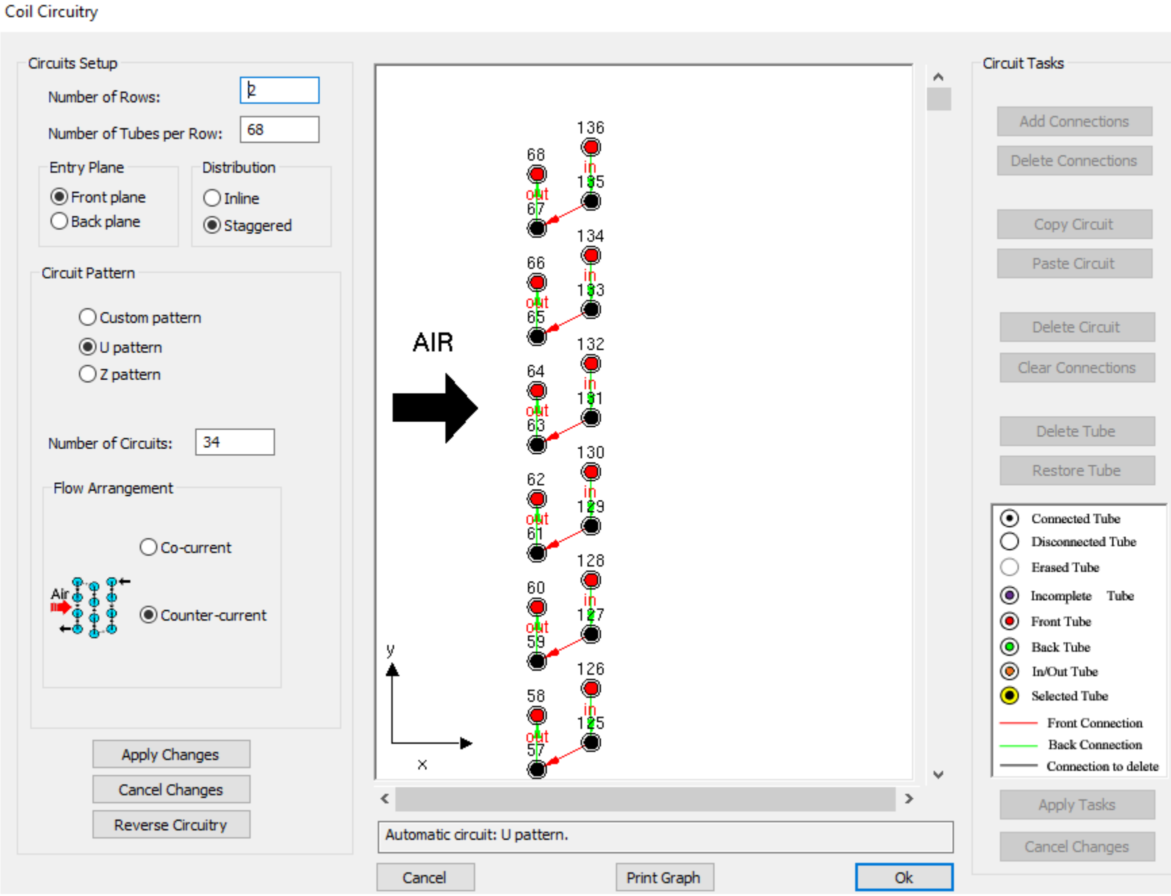


Figure 40-Condenser frontal view

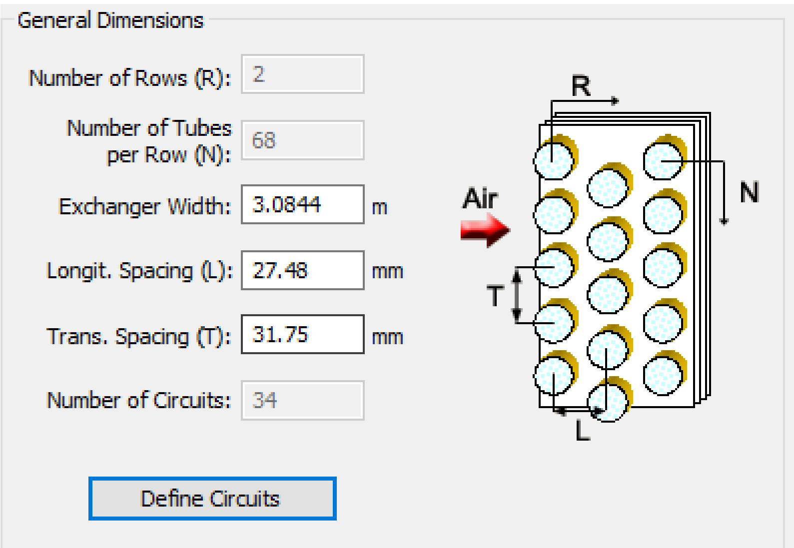


Figure 41-Condenser general dimensions

- **Evaporator**

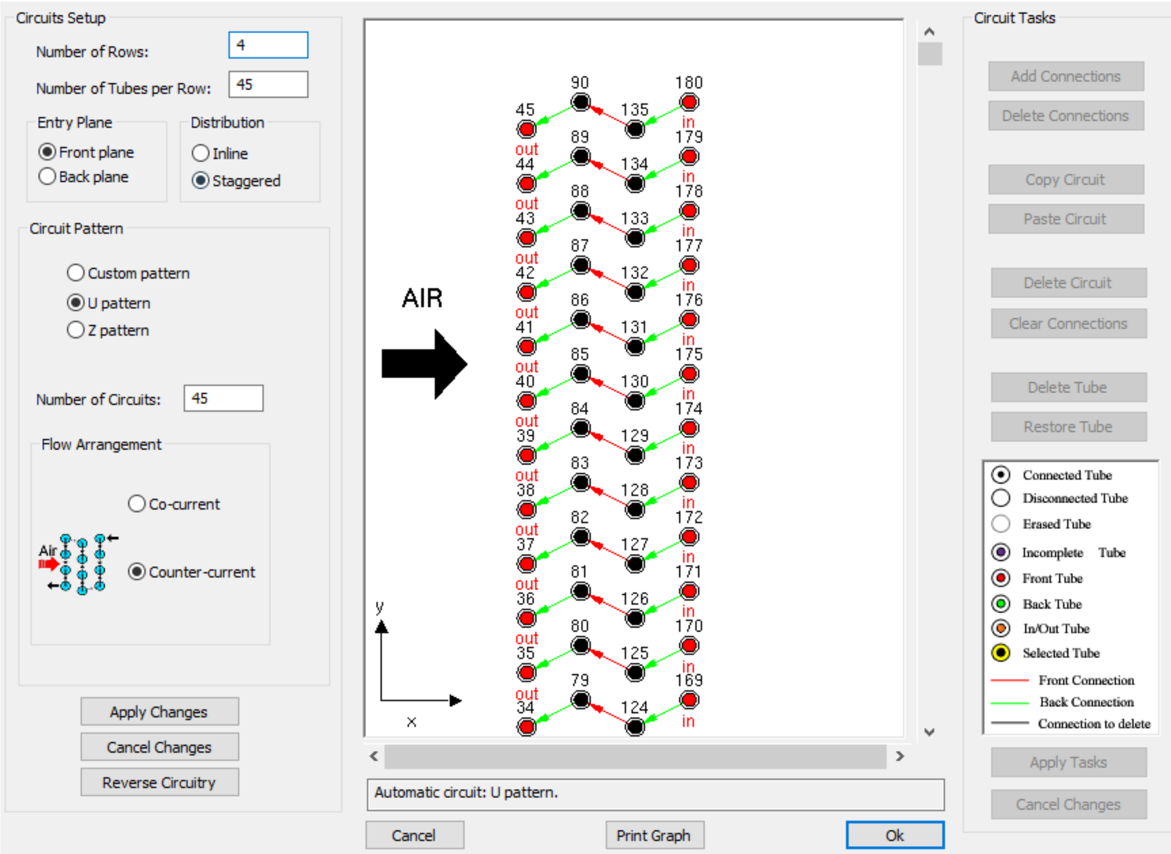


Figure 42-Evaporator frontal view

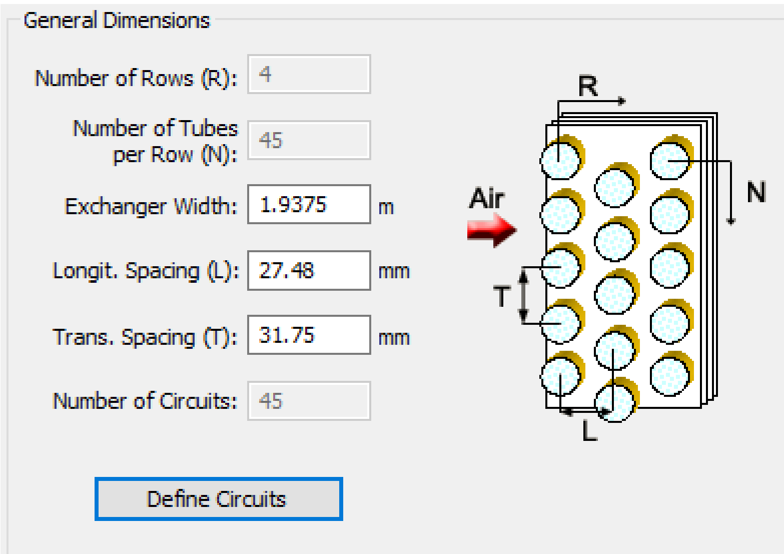


Figure 43-Evaporator general dimensions

Therefore, the results indicate the condenser has 2 rows, including 17 circuits distributed in 68 tubes per row. The number of circuits has been calculated by selecting a divisor of the number of pipes. This design presents significantly adequate results. The reduced number of rows means a lower cost in materials and free space. 2 or 3 rows are the most used numbers of rows in exchangers by the companies due to the provided results.

As for the evaporator, the number of rows increases until 4, which can be considered a high number. 15 circuits are running toward the 45 tubes per row. Even though there are fewer tubes per row than the condenser, which means that the height is shorter, there are still 2 rows more, which will increase considerably the costs.

There are many values to consider regarding batteries. Therefore, Table 8 simplifies and gathers all the relevant parameters related to the condenser and evaporator final design.

	Condenser	**Evaporator**
U [W/K· m^2]	40	40
LMTD [ºK]	15.88	14.10
Heat exchange area (A) [mm^2]	232.99	204.47
Frontal area (A_f) [m^2]	6.85	3.15
Volume flow ($\dot{V}$) [m^3/s]	13.70	6.32
Air velocity (u) [m/s]	2	2.3
Sl [mm]	27.48	27.48
St [mm]	31.75	31.75
Tube diameter (Dc) [“]	0.5	0.5
Tube thickness [mm]	0.813	0.813
Fins thickness (tf) [mm]	0.1	0.1
Compactness (β) [m^{-1}]	717	717
Volume (V) [m^3]	0.28	0.32
Number of rows	2	4
Number of tubes per row	68	45

Table 8-Summary of the important parameters of the batteries

3.3. Selection of system accessories and conduits

3.3.1. Pipeline

On the design of the pipeline, there are three different pipes to consider:

- **Suction line**: the line in which the gas is going through from the evaporator to the compressor. Through this pipe, there is only the refrigerant in gas form coming from the change of phase made in the evaporator. There is some possibility that not all of the coolant has changed to the gaseous phase, leaving traces of liquid through the line. It can be caused due to the reliability or performance of the evaporator or the difficulty of the working temperatures.
- **Discharge line**: the line connects the compressor with the condenser. Through this line, the refrigerant is in the gaseous phase when it passes through the compressor, arriving at the condenser in the same condition. For the security of the condenser, the compressor must compress the refrigerant as much as possible to convert all the fluid to gas. If there is an amount of quantity of liquid at the input, that could damage the condenser.
- **Liquid line**: the line that connects the condenser with the expansion valve and it with the evaporator. The condenser changes the refrigerant into a liquid phase at high pressure. Then the expansion valve only reduces the pressure of the liquid before the entrance into the evaporator.

The factors needed to select are the exterior and the interior diameter of the pipe. For the tubes, it is going to be consulting the Danfoss selector. It is submitting the data seen in Table 9 to get the potential examples of pipes for the system.

Refrigeration capacity [kW]	115.36
Evaporation temperature [ºC]	7
Condensation temperature [ºC]	55.8
Discharge temperature [ºC]	106.9
Subcooling [ºK]	0
Superheating [ºK]	10

Table 9-Required data for the pipeline selection

The discharge temperature is obtained after running the system with the current data. Figure 44 shows the values that would be provided by the system.

-Compressor

	Units	Case 1
Name		Tandem
Number of Compressors		1
Compressor Type		Catalogue data
Displacement	cm³	405.2
Speed	rpm	2900
Ref. Mass Flowrate	kg/s	0.51351
Ref+Oil Mass Flowrate	kg/s	0.51351
Refrigerant Solubility	%	10.891
Inlet Flowrate	m³/h	66.962
Suction Pressure	kPa	1084
Suction Sat. Temp.	ºC	9.2682
Discharge Pressure	kPa	3380.8
Discharge Sat. Temp.	ºC	53.206
Pressure Ratio		3.1189
Suction Temp.	ºC	19.268
Discharge Temp.	ºC	106.9
Inlet Superheat	K	10
Suction Density	kg/m³	27.607
Compressor Power	kW	34.322
Inverter Power	kW	0
Power Input	kW	34.322
Heat Losses	%	5
Isentropic Eff.	%	76.947
Compressor Eff.	%	73.099
Volumetric Eff.	%	94.974

-Condenser

	Units	Case 1
Name		Condenser
Exchanger ´s Type		Coil
Calculation Model		Detailed
Capacity	kW	147.97
Sensible Capacity	kW	147.97
SHR	%	100
Min. Wall Temp.	ºC	47.288
Max. Wall Temp.	ºC	70.088
Refrigerant		R32
Mass Flowrate	kg/s	0.51351
Inlet Temperature	ºC	106.9
Outlet Temperature	ºC	53.077
Inlet Superheat	K	53.69
Outlet Subcooling	K	0
Pressure Drop	kPa	9.8942
Secondary Fluid		Air
Flowrate	m³/h	47946
Face Velocity	m/s	2
Inlet Temperature	ºC	35
Outlet Temperature	ºC	44.604
Pressure Drop	Pa	23.895

-Evaporator

	Units	Case 1
Name		Evaporator
Exchanger ´s Type		Coil
Calculation Model		Detailed
Capacity	kW	115.36
Sensible Capacity	kW	82.776
SHR	%	71.754
Min. Wall Temp.	ºC	11.594
Max. Wall Temp.	ºC	23.032
Refrigerant		R32
Mass Flowrate	kg/s	0.51351
Inlet Temperature	ºC	10.71
Outlet Temperature	ºC	19.268
Inlet Quality		0.28772
Outlet Superheat	K	10
Pressure Drop	kPa	46.713
Secondary Fluid		Air
Flowrate	m³/h	23000
Face Velocity	m/s	2.308
Inlet Temperature	ºC	27
Outlet Temperature	ºC	16.046
Pressure Drop	Pa	77.124

Figure 44-System data after the first simulation by IMST-ART

Then, with that information, and consulting the software offered by Danfoss company, it is comparing and finally selecting the most suitable pipeline for each type as can be seen in Table 10.

Pipe	Type	Diameter [mm]	Inlet velocity [m/s]	Outlet velocity [m/s]	Drop pressure [bar]
Suction line	ANSI 2 1/8	53.98	10.36	10.40	0.033
Discharge line	ANSI 1 5/8	41.28	6.37	6.37	0.046
Liquid line	ANSI 7/8	22.23	1.96	1.96	0.117

Table 10-Pipeline final selection

The tubes selected, have been chosen regarding the safety recommendations in tubes design about the velocities going through them depending on the fluid phase. The instructions are that through a pipe where the fluid is in the liquid phase (Liquid line), the velocity must be between 0.7 and 2 m/s to make the system safe. About the tubes in which there is circulating fluid in the gaseous phase, the recommendation is that the velocity should be around 10 m/s and 5 m/s.

3.3.2. Liquid visor

The liquid visor consists of a device installed most of the times in the liquid line and is used to control the refrigerant status. This device is also selected using the Danfoss selector. Applying the same data of the pipes. Hence, Table 11 shows the selection.

Type	Diameter [mm]	Inlet velocity [m/s]	Drop pressure [bar]
SGS 7/8	22.23	1.97	0.003

Table 11-Liquid visor data

3.3.3. Differential pressure switch

This device function consists of adjusting the pressure range within the electric circuit. It regulates the circuit opening and closing it by electric drive. For this tool, the Danfoss's software has also been used. The selected model is the **MP55E** seen in Figure 45, available for the refrigerant R-32.

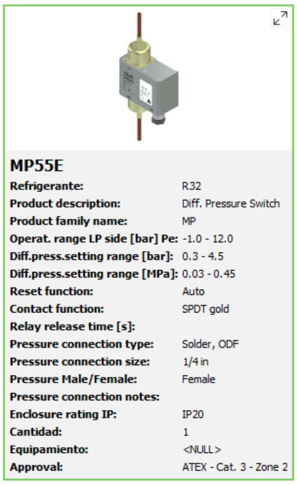


Figure 45-Differential pressure switch

3.3.4. Liquid receiver

These devices are used to control the change of temperatures that occur in the fluid circulating through the system in refrigeration and air conditioning facilities. Thus, they are devices that have the function of compensating for variations in refrigerant pressure and volume. There is no difference in their operation as to whether it is a vertical or horizontal tank. They are used depending on the available space.

It is by standardising a load of 50 % more of the volume than calculated to ensure a security range of possible capacity. It is measured by seeing the quantity of refrigerant is consumed during the cooling mode minus the used by the heat mode. The refrigerant density will divide it and the volume required will be given. Then, at the time of finding a device in catalogues, an increase of 50% of the given volume will be necessary.

Therefore, the quantity of refrigerant needed in the different working cycles is 10.319 kg in Heat Mode and 14.136 kg in Cooling Mode. The difference in mass between them is 3.817 kg. This quantity means the amount of refrigerant left over when the system is in Heat Mode. This fluid must be stored in the liquid deposit that is designing. The density of the refrigerant used in this system is 820.15 kg/m3. Then the volume needed for the liquid deposit will be:

$$V = \frac{m}{\rho} + 50\% \qquad (12)$$

$$V = \frac{m}{\rho} + 50\% = \frac{3.817}{820.15} \cdot 1000 + 50\% = 6.3819 + 50\% = 6.98\, l$$

Thus, the minimum volume required for a liquid deposit for the system will be 6.98 litres. Having checked the Tecnac’s catalogue of liquid deposits, in this case, horizontal by recommendation of the company because of the exchangers' design, the receiver selected is explained in Table 12:

Model	Code	Volume [l]	Inlet valve	Outlet valve
RH-10-GB	86.001	10	7/8”	1” x 5/8” ODS

Table 12-Liquid receiver data

Dimensions [mm]:

- Ø1= 159
- L= 580
- A= 128
- B= -
- C= 400
- T= 80

These dimensions can be compared with the sketch of the device shown in Figure 46:

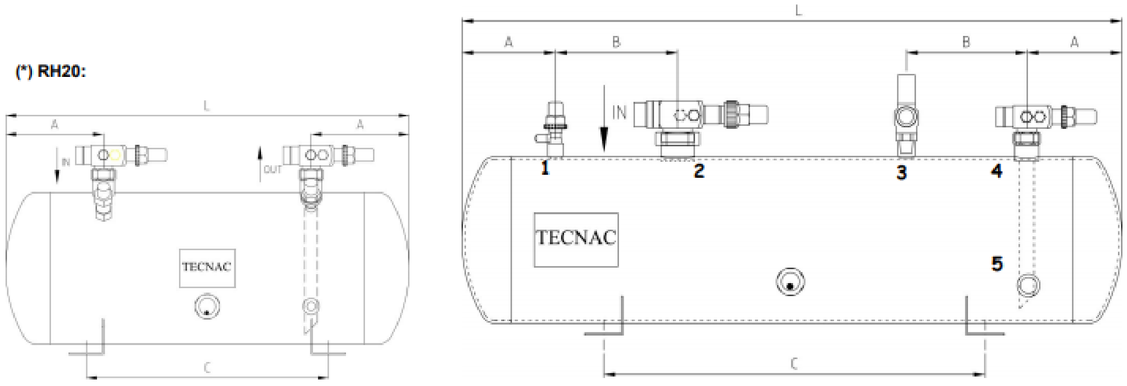


Figure 46-Tecnac's liquid receiver scheme

3.3.5. Liquid exchanger

Devices with the function of preventing and excrete as much as possible the refrigerant and oil returning from the compressor absorption. These machines are commonly used in installations under great temperature variations, cycle inversions and large tubes designs. The information about the selected device is recorded in *Table 13*.

Type	Volume [l]	Welding fittings ODF	Welding fittings for exchangers	Changer capacity [gas kg at 30 ºC]
LCYE 1517 S/MMS	15.2	2” 1/8”	7/8”	10.5

Table 13-Liquid exchanger data

Dimensions [mm]:

- Ø1 = 219.1
- Ø2 = 224
- Ø3 = 8 x Ø10.2
- Ø4 = 190
- L= 638
- E1 = 114
- E2 = 141

These dimensions can be compared with the sketch of the device shown in Figure 47:

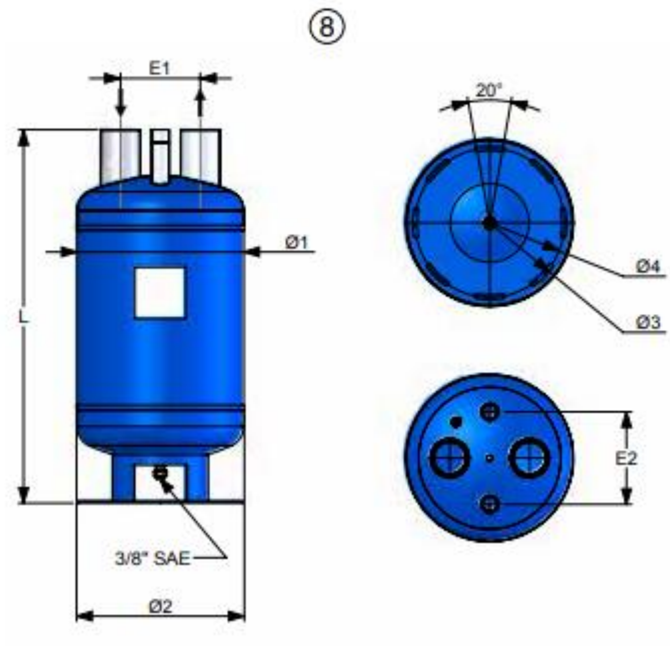


Figure 47-Liquid exchanger data

3.3.6. Expansion devices selection

The valves choice has been made consulting the Danfoss selector software. It has been a magnificent tool due to the possibility of comparing a large number of types that could be potentially adequate to this system. The selection must be considering the dimensions of the pipeline.

There are 3 valves required for the system design. Those are the expansion valve, the four-way valve and the retention valve.

- **Four-way valve**

These types of valves are used in cooling applications where the cycle is reversible. It connects the suction and discharge line with the rest of the system.

- **Expansion valve**

A device that removes pressure from the high-pressure fluid coming from the condenser, which has previously condensed the refrigerant. The result is a low-pressure fluid at the evaporator's inlet. This device is positioned in the liquid pipe to fulfil its function. [6]

- **Retention valve**

A valve that only allows the fluid to flow through it in an only one-way direction depending on the preferences of the system designer. It forbids the return of fluid provoking damages in the exchangers and the rest of the system. [34]

Then, introducing the mean data into the Danfoss software, the selected devices are defined in Table 14*:*

Valve	Type	Diameter [mm]	Charge [%]	Inlet velocity [m/s]	Drop Pressure [bar]
Retention	NRVH 22 E	22	43	2.16	0.281
Expansion	ETS Colibri 12C-22	22	64	1.96	24.14

Table 14-Selected valves data

The **4-ways valve** is different. It has been selected on the Danfoss website. The model is **061L1287 and** has a discharge conn size of 1 5/8 inches and an evaporation/condensation conn size of 2 1/8 inches. Thus, the diameters of the valve match with the corresponding tubes of the system.

3.4. Cooling cycle inversion: heat pump mode

Now, the system is going to be designed in heat pump mode to check if the capacity and the efficiency are appropriate for the system. Test requirements for heat pump design are according to the standard UNE-EN 14511-2:2019. The conditions are shown in Table 15:

Exterior heat exchanger		Interior heat exchanger	
Dry bulb temperature at inlet	Wet bulb temperature at inlet	Dry bulb temperature at inlet	Wet bulb temperature at inlet
7ºC	6ºC	20ºC	15ºC

Table 15-Standard heat pump conditions

Therefore, the initial values for the system in heat mode are the shown in the Table 16, values chosen particularly for this project.

	Temperature
Tin,air.evap	7 ºC
Tout,cond	29 ºC
Tcond	53 ºC
Tevap	10 ºC
SC	0
SH	10 ºK
ΔT,cond	24 ºK
ΔT,evap	3 ºK

Table 16-Heat pump initial data

Once the initial data is defined, it is introduced in IMST-art. Besides, the condenser and evaporator conditions must be changed to the ones indicated by the normative. The number of tubes and rows remains the same because the thing is that the system is reversible, so the exchangers remain in the same place with other initial conditions. Now, the previous condenser has the function of an evaporator due to the change of the circuit direction.

The exchangers have different conditions; they can be seen in Figure 48 in the case of the indoor exchanger and in Figure 49 in the case of the outdoor exchanger.

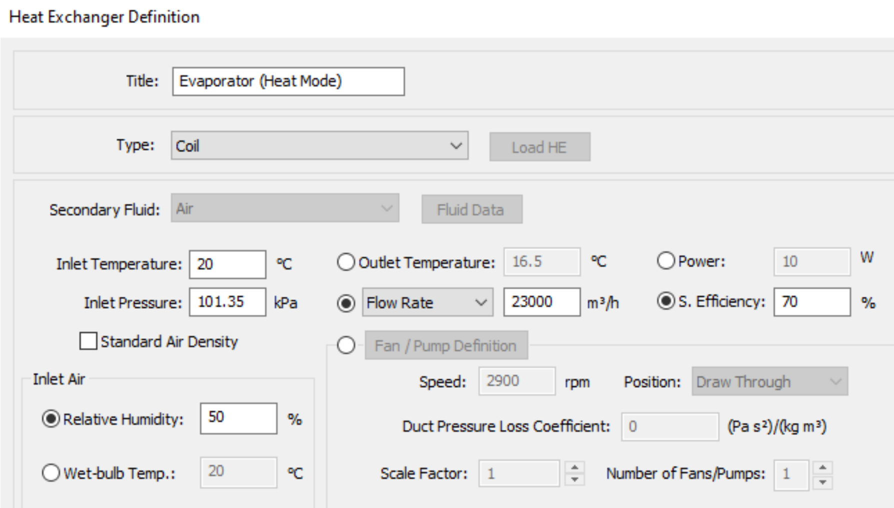


Figure 48-Indoor heat exchanger in Heat Mode

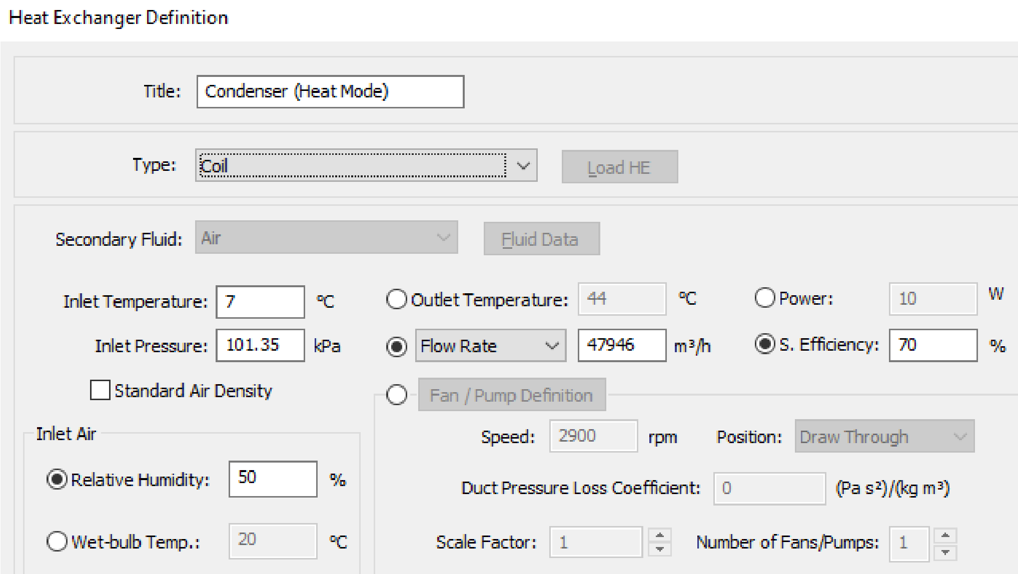


Figure 49-Outdoor heat exchanger in Heat Mode

It has been called "Evaporator – (Heat Mode)" at the exchanger that is doing the same thing as the condenser in Cooling Mode. The same way with the "Condenser – (Heat Mode)", it is fulfilling the function of the evaporator in Cooling Mode. Therefore, all the system in Heat Mode is defined and ready to be run by the software. It provides data that appears in Figure 50. The compressor also remains the same, with two compressors in tandem.

-Compressor -Condenser -Evaporator

	Units	Case 1
Name		Tandem
Number of Compressors		1
Compressor Type		Catalogue data
Displacement	cm³	405.2
Speed	rpm	2900
Ref. Mass Flowrate	kg/s	0.32614
Ref+Oil Mass Flowrate	kg/s	0.32614
Refrigerant Solubility	%	7.2434
Inlet Flowrate	m³/h	66.673
Suction Pressure	kPa	689.1
Suction Sat. Temp.	ºC	-5.0895
Discharge Pressure	kPa	2395.8
Discharge Sat. Temp.	ºC	38.612
Pressure Ratio		3.4768
Suction Temp.	ºC	4.9105
Discharge Temp.	ºC	98.404
Inlet Superheat	K	10
Suction Density	kg/m³	17.61
Compressor Power	kW	24.252
Inverter Power	kW	0
Power Input	kW	24.252
Heat Losses	%	5
Isentropic Eff.	%	76.779
Compressor Eff.	%	72.94
Volumetric Eff.	%	94.565

	Units	Case 1
Name		Evaporator (Heat Mode)
Exchanger ´s Type		Coil
Calculation Model		Detailed
Capacity	kW	105.62
Sensible Capacity	kW	105.62
SHR	%	100
Min. Wall Temp.	ºC	30.829
Max. Wall Temp.	ºC	55.066
Refrigerant		R32
Mass Flowrate	kg/s	0.32614
Inlet Temperature	ºC	98.404
Outlet Temperature	ºC	38.467
Inlet Superheat	K	59.792
Outlet Subcooling	K	0.00023217
Pressure Drop	kPa	8.3434
Secondary Fluid		Air
Flowrate	m³/h	23000
Face Velocity	m/s	2.308
Inlet Temperature	ºC	20
Outlet Temperature	ºC	33.619
Pressure Drop	Pa	58.302

	Units	Case 1
Name		Condenser (Heat Mode)
Exchanger ´s Type		Coil
Calculation Model		Detailed
Capacity	kW	82.581
Sensible Capacity	kW	81.261
SHR	%	98.402
Min. Wall Temp.	ºC	-3.0636
Max. Wall Temp.	ºC	6.1619
Refrigerant		R32
Mass Flowrate	kg/s	0.32614
Inlet Temperature	ºC	-3.9807
Outlet Temperature	ºC	4.9102
Inlet Quality		0.24683
Outlet Superheat	K	9.9997
Pressure Drop	kPa	26.133
Secondary Fluid		Air
Flowrate	m³/h	47946
Face Velocity	m/s	2
Inlet Temperature	ºC	7
Outlet Temperature	ºC	2.1929
Pressure Drop	Pa	31.155

Figure 50-System final data in Heat Mode

4. EFFICIENCY ANALYSIS

4.1. Calculation for the determination of seasonal space cooling energy efficiency (η_{sc})

Supported by IMST-art software, it has been obtained the heating and refrigeration power that the system provides. The final step is calculating the seasonal efficiency to verify the system fulfils the minimum requirements by the European Normative. Hence, it is following the EN-14825 and EN 2016/2281 to obtain and evaluate the values. Firstly, it is introducing several definitions to make it more explicit. It starts defining the different operating modes from which the energy efficiency calculation is to be applied.

- **Active mode**: means the mode corresponding to the hours with a cooling or heating load of the building and whereby the cooling or heating function of the unit is activated. This condition may involve on/off-cycling of the unit in order to reach or maintain a required indoor air temperature. [1]
- **Standby mode**: means a condition where the warm air heater, comfort chiller, air conditioner or heat pump is connected to the mains power source, depends on energy input from the mains power source to work as intended and provides only the following functions, which may persist for an indefinite time: reactivation function, or reactivation function and only an indication of enabled reactivation function, and/or information or status display. [1]
- **Thermostat-off mode**: means the condition corresponding to the hours with no cooling or heating load, whereby the cooling or heating function is switched on, but the unit is not operational; cycling in active mode is not considered as thermostat-off mode. [1]
- **Crankcase heater mode**: means the condition in which the unit has activated a heating device to avoid the refrigerant migrating to the compressor to limit the refrigerant concentration in oil at compressor start. [1]

- **Off mode**: means a condition in which the comfort chiller, air conditioner or heat pump is connected to the mains power source and is not providing any function. Also considered as 'off mode' are conditions providing only an indication of 'off mode' condition, as well as conditions providing only functionalities intended to ensure electromagnetic compatibility pursuant to Directive 2004/108/EC of the European Parliament and of the Council.[1]

4.1.1. SEER calculation

Starting with the Cooling Mode, this unit is air/air type and the conditions to determine the required DC and EERd for cooling are given in Figure 51:

Rating point	Outdoor temperature	Part load ratio	Outdoor side heat exchanger	Indoor side heat exchanger
Air-to-air air conditioners				
	T_j (°C)		Outdoor air dry bulb temperatures (°C)	Indoor air dry bulb (wet bulb) temperatures (°C)
A	35	100 %	35	27 (19)
B	30	74 %	30	27 (19)
C	25	47 %	25	27 (19)
D	20	21 %	20	27 (19)

Figure 51-Part load conditions for air conditioners, comfort chillers and heat pumps

That Table is in UE 2016/2281 Regulation, Annex III, Table 21. It presents the different work points depending on the system's charge. For every single case, there are distinct exterior temperatures to maintain the interior's conditions. To every ambient temperature corresponds a diverse load factor, from which the refrigeration demand is calculated. [1]

This table means that the system must do a parametric study for loads of 100, 74, 47 and 21% of the system total power. It sets different temperature conditions for the bulbs. Therefore, a parametric study is doing in IMST-art software from which is getting the system information working at these different powers [1]. The analysis made is shown in Figure 52:

	Units	Case 1	Case 2	Case 3	Case 4
Cond. Inlet Temperature	°C	35	30	25	20
Evap. Inlet Temperature	°C	27	27	27	27

Figure 52-Parametric study for cooling

The manufacturer must provide two coefficients,

- **C_{dc}** (Cooling Degradation coefficient): the measure of efficiency loss due to cycling of the product; if it is not determined by measurement, then the default degradation coefficient shall be 0,25 for an air conditioner or heat pump, or 0,9 for a comfort or high temperature process chiller. [1]
- **CR** (Capacity Ratio): it is the part load for cooling, divided by the declared cooling capacity (DC). [1]

Two possible options are depending on DC's value:

- If the unit's DC corresponds with the required refrigeration load, it means that CR is equal to one. Then, it is using the corresponding declared energy efficiency EERd, depending on the temperature point's period. [1]
- On the other hand, if the unit's DC is superior to the required refrigeration load, the CR will be higher than one. It means that the machine has to conduct on/off cycles. That usually happens in units with fixed power, staged or variable. In such cases, C_{dc} must be used to calculate the corresponding EERbin value. In such case the EERbin will be calculated with the following formula: [1]

$$EER_{bin} = EER_d \times [1 - C_{dc} \times (1 - CR)] \qquad (13)$$

The UE 2016/2281 Regulation, Table 27, which here is presented as Figure 53, many exterior temperatures, and the number of hours at which the average cooling time is at that temperature. Besides, the last column indicates how to calculate EER. On the other hand, EN-14825 Normative gives one formula to determine the refrigeration demand (Pc (Tj)) from a partial load factor, which depends on the outside temperature. [1]

$$pl(T_j) = \frac{T_j - 16}{Tdesign - 16} \quad (14)$$

- $pl(T_j)$ → Part load factor [%]
- T_j → Exterior temperature [ºC]

Then, after substituting in (14):

$$Pc(T_j) = pl(T_j) \times Pdesignc \quad (15)$$

- $Pc(T_j)$ → Partial cooling load [kW]
- $Pdesignc$ → Design power for cooling [kW]

European cooling season for comfort chillers and air conditioners

Bins	Outdoor temperature (dry bulb)	'Average cooling season' bin hours	EER calculation
j	T_j	h_j	
#	°C	h/annum	
1	17	205	*EER(D)*
2	18	227	*EER(D)*
3	19	225	*EER(D)*
4	20	225	D — Measured value
5	21	216	Linear interpolation
6	22	215	Linear interpolation
7	23	218	Linear interpolation
8	24	197	Linear interpolation
9	25	178	C — Measured value
10	26	158	Linear interpolation
11	27	137	Linear interpolation
12	28	109	Linear interpolation
13	29	88	Linear interpolation
14	30	63	B — Measured value
15	31	39	Linear interpolation
16	32	31	Linear interpolation
17	33	24	Linear interpolation
18	34	17	Linear interpolation
19	35	13	A — Measured value
20	36	9	*EER(A)*
21	37	4	*EER(A)*
22	38	3	*EER(A)*
23	39	1	*EER(A)*
24	40	0	*EER(A)*

Figure 53-European cooling season for comfort chillers and air conditioners

Then, all the information now is known, so it is possible to obtain the Capacity Ratio and from there, calculate the rest of the required factors:

$$pl(T_j) = \frac{T_j - 16}{35 - 16} \qquad (14)$$

$$CR = pl(T_j) \times \frac{Pdesignc}{DC} \qquad (16)$$

$$EER_{bin} = EER_d \times [1 - C_{dc} \times (1 - CR)] \qquad (13)$$

Thus, the annual demand results by multiplying the power by the number of hours at which European cooling season for comfort chillers and air conditioners. If its EERbin (13) divides it, the electric consumption will be known.

$$\boldsymbol{Annual\ cooling\ demand}\ [kWh] = bin\ hour\ (h_j) \times Pc(T_j) \qquad (17)$$

$$\boldsymbol{Annual\ consumption}\ [kWh] = \frac{Annual\ coolin\ demand}{EERbin} \qquad (18)$$

$$SEER = \frac{Q_C}{Q_{CE}} \qquad (19)$$

- $\boldsymbol{Q_C}$ **(Reference annual cooling demand)**→ means the reference cooling demand to be used as a basis for calculation of SEER and calculated as the product of the design cooling load (Pdesign,c) and the equivalent active mode hours for cooling (HCE), expressed in kWh. [1]

$$Q_C = Pdesignc \times H_{CE} \qquad (20)$$

  - H_{CE} → active mode hours for cooling [h]
  - $Pdesignc$ →Design cooling load [kW]

- Q_{CE} **(Annual energy consumption for cooling) →** means the energy consumption required to meet the 'reference annual cooling demand' and is calculated as the 'reference annual cooling demand' divided by the 'active mode seasonal energy efficiency ratio' (SEERon) and the electricity consumption of the unit for thermostat-off, standby, off and crankcase heater mode during the cooling season, expressed in kWh. [1]

$$Q_{CE} = \frac{Q_C}{SEERon} + H_{TO} \times P_{TO} + H_{SB} \times P_{SB} + H_{CK} \times P_{CK} + H_{OFF} \times P_{OFF} \qquad (21)$$

The operational hours are represented in Table 17. Those values have been chosen following the UE 2016/2281 Regulation "Operational hours per functional mode for comfort chillers, air conditioner and heat pumps", Table 29, Average season's row. [1]

OPERATIONAL HOURS [h]				
Active mode H_{CE}	**Thermostat Off mode H_{TO}**	**Crankcase heater mode H_{CK}**	**Off mode H_{OFF}**	**Standby mode H_{SB}**
600	659	2036	0	1377

Table 17-Operational hours per functional mode for comfort chillers, air conditioners and heat pumps in the average season

The power values are estimated from manufacturers' catalogues because they require tests to be determined. Table 18 has been estimated the values that are going to be considered in this system. [1]

POWER CONSUMPTION [kW]			
Thermostat Off mode P_{TO}	**Crankcase heater mode P_{CK}**	**Off mode P_{OFF}**	**Standby mode P_{SB}**
0.128	0.06	0	0.015

Table 18-Estimated power for every mode

From IMST-art, it is simulating a parametric study with the information given in Figure 52, which results are displayed in Figure 54. In that figure, it is included the Cdc factor and the resultant CR and EERbin.

Compressor		Higher compressor		Lower compressor				
Declared cooling capacity DC kW	EERd	Declared cooling capacity DC kW	EERd	Declared cooling capacity DC kW	EERd	Cdc	CR	EERbin
109,68	3,00	85,25	3,70	48,35	4,84	0,25	1,00	2,75
114,54	3,46	89,13	4,28	50,16	5,44	0,25	1,00	3,97
119,23	3,99	92,90	4,93	51,88	6,05	0,25	0,99	5,27
123,82	4,58	95,57	5,65	53,55	6,76	0,25	0,43	5,37

Figure 54-Parametrict study's result in Cooling Mode

Hence, once all the information is clear, it is time to substitute the values in formulas. Due to the great amount of information and values, it has been reduced in Figure 55, table that shows all the required information to obtain the SEER.

Rating point	Bins j	Outdoor air dry bulb temperature Tj ºC	bin hours hj h	Part load ratio %	Partial cooling load Pc(Tj) kW	EERbin	Annual cooling demand kWh	Annual power input kWh
	1	17	205	5%	5,77	5,37	1183,39	220,17
	2	18	227	11%	11,55	5,37	2620,77	487,61
	3	19	225	16%	17,32	5,37	3896,53	724,97
D	4	20	225	21%	23,09	**5,37**	5195,37	966,62
	5	21	216	26%	28,86	5,35	6234,44	1164,64
	6	22	215	32%	34,64	5,33	7446,69	1396,74
	7	23	218	37%	40,41	5,31	8809,04	1659,00
	8	24	197	42%	46,18	5,29	9097,67	1720,38
C	9	25	178	47%	51,95	**5,27**	9247,76	1755,95
	10	26	158	53%	57,73	5,01	9120,76	1821,36
	11	27	137	58%	63,50	4,75	8699,36	1831,92
	12	28	109	63%	69,27	4,49	7550,60	1681,69
	13	29	88	68%	75,04	4,23	6603,89	1560,84
B	14	30	63	74%	80,82	**3,97**	5091,46	1281,80
	15	31	39	79%	86,59	3,73	3376,99	905,98
	16	32	31	84%	92,36	3,48	2863,23	822,11
	17	33	24	89%	98,13	3,24	2355,23	727,34
	18	34	17	95%	103,91	2,99	1766,43	590,09
A	19	35	13	100%	109,68	**2,75**	1425,84	518,71
	20	36	9	105%	115,45	2,75	1039,07	378,01
	21	37	4	111%	121,23	2,75	484,90	176,40
	22	38	3	116%	127,00	2,75	380,99	138,60
	23	39	1	121%	132,77	2,75	132,77	48,30
	24	40	0	126%	138,54	2,75	0,00	0,00

Figure 55-SEERon calculation in Cooling Mode

$$\boldsymbol{SEERon} = \frac{Annual\ cooling\ demand}{Annual\ power\ input} \qquad (22)$$

$$\boldsymbol{SEERon} = \frac{Annual\ cooling\ demand}{Annual\ power\ input} = \frac{104623.17}{22579.24} = 4.63$$

$$Q_C = Pdesignc \times H_{CE} \quad (20)$$

$$Q_C = Pdesignc \times H_{CE} = 113.46 \cdot 600 = 65808\, kWh$$

$$Q_{CE} = 14674.72\, kWh$$

$$\boldsymbol{SEER} = \frac{Q_C}{Q_{CE}} \quad (19)$$

$$\boldsymbol{SEER} = \frac{Q_C}{Q_{CE}} = \frac{65808}{14674.72} = 4.48$$

Regulations (EU) No 811/2013 and (EU) 813/2013 call for seasonal space heating energy efficiency based on the VGC (higher calorific value) of fossil fuel. [1]

$$\eta_{sc} = \frac{1}{CF} \times SEER - \Sigma F(i) \quad (23)$$

- CF =2.5
- Σ F (i) = F (1) + F (2)

F (1) = Correction to consider the negative contribution to the seasonal energetic efficiency in heating and cooling due to the adjusted contribution of the temperature controls, equal 3%.[1]

F (2) = Correction to consider the negative contribution to the seasonal energetic efficiency in heating and cooling of brine and water heat pumps. This factor is only applicable to water-brine/water or water/brine-water and is equal to 5%. [1]

$$\boldsymbol{\eta_{sc}} = \frac{1}{CF} \times SEER - \Sigma F(i) = \frac{1}{2.5} \times 4.48 - 0.03 = 176\ \%$$

From 1 January 2021, the seasonal space cooling energy efficiency of cooling products shall not fall below the values that Table 4 shows in UE 2016/2281 Regulation. That table designates that for rooftop heat pumps, the minimum seasonal cooling energy efficiency is 138 %. [1]

$$\boldsymbol{\eta_{sc} = 176\% > \eta_{sc}(minumum) = 138\%}$$

Therefore, for the Cooling mode, this heat pump design notably fulfils the requirements expressed in EN-14825 and UE 2016/2281 Regulation.

4.2. Calculation for the determination of seasonal space heating energy efficiency (η_{SH})

Firstly, this process is quite similar to the cooling process. In this procedure, there is the calculation of the different seasons: average, colder and warmer. The efficiency must fulfil the normative for heating mode.

To calculate the SCOP/ SCOPon/ SCOPnet of the application explained in EN 14285 chapter 7, partial charge factors must be based on the partial charge factors' formulas and not by numbers given for each climate. The SCOP, SCOPon and SCOPnet will be defined by three reference design conditions that are summarised in Figure 56:

- Temperature design "average" (A): Dry bulb temperature conditions corresponding to an external temperature of -10ºC and an interior temperature of 20ºC. [1]
- Temperature design "colder" (C): Dry bulb temperature conditions corresponding to an external temperature of -22ºC and an interior temperature of 20ºC. [1]
- Temperature design "warmer" (W): Dry bulb temperature conditions corresponding to an external temperature of +2ºC and an interior temperature of 20ºC. [1]

Average	*Warmer*	*Colder*
-10˚C	2˚C	-22˚C

Figure 56-Design temperatures in heating

There are some factors not introduced previously, but in this process, they have to be considered, those are the Bivalent Temperature (Tbivalent) and the Operation Limit temperature (T_{OL}), they are defined as:

- Bivalent Temperature (Tbiv) [ºC] → means the outdoor temperature (Tj) declared by the manufacturer at which the declared heating capacity equals the part load for heating and below which the declared heating capacity has to be supplemented with electric back-up heater capacity in order to meet the part load for heating, expressed in degrees Celsius. [1]

- For the medium heating station, the dry bulb bivalent temperature is +2ºC or less. [1]
- For the coldest heating station, the dry bulb bivalent temperature is -7 ºC or less. [1]
- For the warmest heating station, the dry bulb bivalent temperature is +7ºC or less. [1]

For higher or equal dry bulb temperatures of -10ºC, wet bulb temperature is the same as the dry bulb's temperature minus 1ºK. For dry bulb temperatures inferior to -10ºC, wet bulb temperature does not define.

- Operation Limit Temperature (T_{OL}) [ºC]→ means the outdoor temperature declared by the manufacturer for heating, below which the heat pump will not be able to deliver any heating capacity and the declared heating capacity is equal to zero, expressed in degrees Celsius. [1]

If T_{OL} is inferior to the design temperature of the climate considered, then the exterior dry bulb temperature is equal to the design temperature for the partial charge condition E given in Figure 57. This table corresponds with Table 21 (Annex III) of the UE 2016/2281 Regulation. [1]

In case of the coldest climates and the T_{OL} is less than -20ºC, must be applicate an additional partial charge G at -15ºC.

Air-to-air heat pumps

Rating point	T_j (°C)	Part load ratio	Outdoor air dry bulb (wet bulb) temperatures (°C)	Indoor air dry bulb temperature (°C)
A	− 7	88 %	− 7(− 8)	20
B	+ 2	54 %	+ 2(+ 1)	20
C	+ 7	35 %	+ 7(+ 6)	20
D	+ 12	15 %	+ 12(+ 11)	20
E	T_{ol}	depends on T_{ol}	$T_j = T_{ol}$	20
F	T_{biv}	depends on T_{biv}	$T_j = T_{biv}$	20

Figure 57-Partial charge conditions for air-air units

There is one factor that is necessary to consider before the calculations.

- $COPbin(T_j)$ (bin-specific coefficient of performance) → means the coefficient of performance of the heat pump for every bin_j with outdoor temperature (Tj) in a season, derived from the part load, declared capacity and declared coefficient of performance (COPd (Tj) and calculated for other bins through inter/extrapolation, when necessary corrected by the applicable degradation coefficient. [1]

To obtain the declared loads in each point and the correspondent COPbin is necessary to use the Normative EN-14825, which identifies the closest power jump to reach the required heating load within a range of not more than 10%. If there would be not enough capacity to cover the required load, the normative allows determining the heating power at the temperatures that are above or under the previously mentioned load and interpolate. [1]

This study is made by introducing all the settings in the software and simulating it in Heating mode. Figure 58 displays the parametric study made.

	Units	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6
Cond. Inlet Temperature	°C	20	20	20	20	20	20
Cond. Wet-bulb Temp	°C	15	15	15	15	15	15
Evap. Inlet Temperature	°C	-7	2	7	12	-10	-8
Evap. Wet-bulb Temp	°C	-8	1	6	11	-11	-9

Figure 58-Parametric study in Heating Mode

The process to follow is almost the same as the cooling mode. There is the difference that there are more calculations due to the three different seasons. Following the normative, the only consideration that is strictly necessary is the Average season. Therefore, after running the system, declared powers and COPd are obtained. An informative table has been made to contemplate the data easily. Figure 59 shows the results of the system after the simulation.

			Two compressors		Higer compressor		Lower compressor	
Rating point	Outdoor air dry bulb (wet bulb) temperature Tj ºC	Indoor air dry bulb (wet buld) temperature ºC	Declared heating capacity DC kW	COPd	Declared heating capacity DC kW	COPd	Declared heating capacity DC kW	COPd
A	-7 (-8)	20(15 max)	72,53	3,14	50,62	3,37	23,78	3,70
B	2 (1)	20(15 max)	94,20	3,57	66,36	4,10	31,64	4,62
C	7 (6)	20(15 max)	107,74	3,81	76,29	4,42	36,73	5,16
D	12 (11)	20(15 max)	122,19	4,06	87,33	4,83	42,47	5,71
E	-10	20(15 max)	66,13	2,99	46,05	3,17	21,52	3,43
F	-8	20(15 max)	70,43	3,09	49,11	3,30	23,04	3,62

Figure 59-Heating capacity and COP obtained after the IMST-art simulation

Defrost losses is a factor to consider. It is supposed to be delivered by the manufacturer. It is going to be estimated comparing with other similar machines. The losses that are going to be reflected in this project are displayed in Figure 60:

Rating point	Defrost capacity losses %	Defrost COP losses %
A	4%	4%
B	3%	3%
C	1%	1%
D	0%	0%
E	5%	5%
F	4%	4%

Figure 60-Defrost losses

Thus, following the same procedure as the cooling mode, Figure 61 shows the obtained values for the Average heating season. It is the only season that is going to show its process, the other two have the same process but with other values. It is showing the different rating points, partial loads and their resultant partial heating load. Therefore, with that information, it is calculating the defrosting losses and the required factor to obtain the efficiency.

Heating season	Average (A)							
Rating point	Outdoor air dry bulb (wet bulb) temperature Tj ºC	Indoor air dry bulb (wet buld) temperature ºC	Part load ratio %	Partial heating load Ph(Tj) kW	Declared heating capacity DC (with defrost) kW	Cdh	CR	COPbin
A	-7	20	88	88,00	69,62784	0,25	1,00	3,01
B	2	20	54	54,00	54	0,25	1,00	4,13
C	7	20	35	35,00	36,36	0,25	0,96	5,06
D	12	20	15	15,00	42,47	0,25	0,35	4,78
E	-10	20	100	100,00	62,82	0,25	1,00	2,84
F	-8	20	92	92,31	67,61	0,25	1,00	2,96

Figure 61-Average heating season obtained values

The operational hours for heating are available also in Table 29 from UE 2016/2281 Regulation. In this case, Table 19 displays the information for an Average season in a reversible system to calculate the SCOP. [1]

OPERATIONAL HOURS [h]				
Active mode H_{CE}	**Thermostat Off mode H_{TO}**	**Crankcase heater mode H_{CK}**	**Off mode H_{OFF}**	**Standby mode H_{SB}**
1400	179	179	0	0

Table 19-Operational hours in Heating Mode

Then, the required data is available to calculate the annual power input and the annual heating demand regarding a range of temperatures. Following the formulas previously commented, Table 20 shows all the needed values to calculate the annual demand and power input in Heating Mode.

Rating point	Bins j	Outdoor air dry bulb temperature Tj	bin hours hj	Part load ratio	Partial heating load Ph(Tj)	COPbin	Declared heating capacity	Annual heating demand	Annual power input
		ºC	h	%	kW		kW	kWh	kWh
	1 to 20	-30 to -11	0						
	21	-10	1	100%	100,00	2,87	63,57	100,00	58,60
	22	-9	25	96%	96,15	2,92	65,59	2403,85	824,47
F	23	-8	23	92%	92,31	2,96	67,61	2123,08	716,43
A	24	-7	24	88%	88,46	3,01	69,63	2123,08	705,05
	25	-6	27	85%	84,62	3,14	67,89	2284,62	728,52
	26	-5	68	81%	80,77	3,26	66,15	5492,31	1684,37
	27	-4	91	77%	76,92	3,39	64,42	7000,00	2067,64
	28	-3	89	73%	73,08	3,51	62,68	6503,85	1852,81
	29	-2	165	69%	69,23	3,64	60,95	11423,08	3142,52
	30	-1	173	65%	65,38	3,76	59,21	11311,54	3008,58
	31	0	240	62%	61,54	3,88	57,47	14769,23	4526,63
	32	1	280	58%	57,69	4,01	55,74	16153,85	4440,17
B	33	2	320	54%	53,85	4,13	54,00	17230,77	4168,04
	34	3	357	50%	50,00	4,32	50,47	17850,00	4132,54
	35	4	356	46%	46,15	4,50	46,94	16430,77	3647,45
	36	5	303	42%	42,31	4,69	43,42	12819,23	2733,27
	37	6	330	38%	38,46	4,88	39,89	12692,31	2603,32
C	38	7	326	35%	34,62	5,06	36,36	11284,62	2229,82
	39	8	348	31%	30,77	5,01	37,58	10707,69	2139,35
	40	9	335	27%	26,92	4,95	38,80	9019,23	1822,26
	41	10	315	23%	23,08	4,89	40,02	7269,23	1485,39
	42	11	215	19%	19,23	4,84	41,25	4134,62	854,58
D	43	12	169	15%	15,38	4,78	42,47	2600,00	543,65

Table 20-Annual demand and power input calculation for Heating Mode

$$\boldsymbol{SCOPon} = \frac{Annual\ heating\ demand}{Annual\ power\ input} \qquad (22)$$

$$\boldsymbol{SCOPon} = \frac{206562}{50718.65} = \mathbf{4.30}$$

$$Q_H = Pdesignc \times H_{HE} \qquad (20)$$

$$Q_H = 100 \cdot 1400 = 140000\ kWh$$

$$Q_{HE} = 34425.04\ kWh$$

$$\boldsymbol{SCOP} = \frac{Q_C}{Q_{CE}} \qquad (19)$$

$$\boldsymbol{SCOP} = \frac{140000}{34425.04} = \mathbf{4.07}$$

And the same way as in Cooling Mode:

$$\boldsymbol{\eta_{sh}} = \frac{1}{CF} \times SCOP - \Sigma F(i) \qquad (23)$$

$$\boldsymbol{\eta_{sh}} = \frac{1}{2.5} \times 4.30 - 0.03 = \mathbf{160}\ \% > \mathbf{125}\ \%$$

The obtained efficiency significantly exceeds the minimum required by the normative. As a comparison, Table 30 from UE 2016/2281 Regulation provides benchmarks for seasonal efficiency for both modes. [1]

Therefore, either Cooling Mode (η_{sc}=176 %) and Heating Mode (η_{sh}=160 %) comply with benchmarks products. [1]

4.3. Comparison between the designed system and a machine of similar capacity that uses R-410A

A machine with similar capacity has been provided to make a comparison in this project. The device works with the refrigerant fluid R-410A. Due to its efficiency, easy handle and because it is non-toxic nor flammable, it is the most commonly used by companies. However, this fluid presents a high GWP. Then it is searching for new refrigerants that can achieve the same performance as R-410A but within the acceptable ranges of pollution. The R-410A was introduced during the Montreal Protocol to reduce the refrigerant gases that attack the ozone layer, as well as R22, which has the quality of not being toxic and, therefore, was the most used in commercial and domestic air conditioning.

Nowadays, there are potential refrigerants such as the R-32. It is a suitable option to replace R-410A in commercial air conditioning. Thus, this project aims to compare both machines and get conclusions on the cost-effectiveness and efficiency of the designed heat pump.

4.3.1. Efficiency comparison

Previously, the efficiency and the SCOP was calculated for the R-32 design:

- $\boldsymbol{\eta_{sc} = 176\%}$
- $\boldsymbol{SEER = 4.48}$
- $\boldsymbol{SCOP = 4.07}$
- $\boldsymbol{\eta_{sh} = 160\ \%}$

Those are considered by the Normative EN 2016/2281 a significantly high performance. They are above the benchmark seasonal efficiencies.

The machine provided was an IMST-ART file, then it is doing with the results obtained the same process as it was done with the R-32 design. The method of the Regulation has been followed equally. Therefore, the obtained values are shown in Figure 62 for the summer season and Figure 63 displays the winter season results:

	Compressor		Higher compressor		Lower compressor	
Rating point	Declared cooling capacity DC kW	EERd	Declared cooling capacity DC kW	EERd	Declared cooling capacity DC kW	EERd
A	149,74	2,74	97,76	3,99	97,76	3,99
B	157,91	3,14	103,17	4,64	103,17	4,64
C	165,68	3,59	108,32	5,42	108,32	5,42
D	173,11	4,11	113,16	6,35	113,16	6,35

Figure 62-Cooling mode results of the R-410A machine

	Two compressors		Higer compressor		Lower compressor	
Rating point	Declared heating capacity DC kW	COPd	Declared heating capacity DC kW	COPd	Declared heating capacity DC kW	COPd
A	112,54	2,99	60,29	3,50	60,29	3,50
B	142,07	3,38	78,70	4,23	78,70	4,23
C	156,76	3,58	90,30	4,67	90,30	4,67
D	179,51	3,77	103,05	5,13	103,05	5,13
E	104,13	2,84	54,97	3,26	54,97	3,26
F	109,67	2,93	58,40	3,42	58,40	3,42

Figure 63-Heating mode results of the R-410A machine

Then, after obtaining the declared capacity of each modulation, it is doing the same process done before in order to obtain the performance of the R-410A machine. Table 21 shows the results:

	Machine with R-32	**Machine with R-410A**
η_{SC} [%]	176	178
Q_C [kWh]	65808	89844
Q_{CE} [kWh]	14675	19862
SEER	4.48	4.52
η_{SH} [%]	160	145
Q_H [kWh]	140000	140000
Q_{HE} [kWh]	34425	37812
SCOP	4.07	3.70

Table 21-Final energy efficiency values of the R-410A design

The results reflect the high efficiency of both systems. However, it should be noted that the result obtained in terms of efficiency shifts towards the design with R-32. In the summer season, the calculated efficiencies are almost identical. Even so, the R-410A machine has a much higher annual demand, which will lead to much higher consumption as can be observed. That means during this season the system with R-32 will not only have high efficiency but will also not consume as much as the other machine does. From the values obtained, during the cooling mode, the designed machine works by making much better use of the energy it receives. It results in the eminent refrigeration capacity of the refrigerant.

As for the winter season, it is clear that the efficiency of the R-32 machine is superior to the R-410A. It has a considerably higher performance, plus a lower consumption. It definitively proves that the energy efficiency of the R-32 is greater than the R-410.

4.3.2. Cost effective

The costs of the heat pump are linked to the weight of the exchangers designed. Therefore, the heavier batteries, the more expensive the project is going to end. Hence, it is proving some weight and configuration values obtained by IMST-ART software after the design process. Table 22 is showing the evaporator configuration and Table 23 the condenser.

	Machine with R-32	**Machine with R-410A**
Number of rows	4	4
Number of circuits	45	63
Fin mass [kg]	39.70	37.64
Tube mass [kg]	94.34	105.61
Total mass [kg]	135.86	144.77

Table 22-Indoor batteries geometry information

	Machine with R-32	Machine with R-410A
Number of rows	2	2
Number of circuits	34	65
Fin mass [kg]	47.75	48.47
Tube mass [kg]	113.42	136.01
Total mass [kg]	162.60	184.99

Table 23-Outdoor batteries geometry information

In addition, it will also be checked that for a similar power, the refrigerant charge will be lower in the design with R-32 than in the R-410A. The comparison between heating and cooling mode between the two machines is indicated in Table 24 and Table 25 respectively.

- Heating

Rating point	A	B	C	D	E	F
Machine with R-32	8.502	9.424	10.004	10.639	8.206	8.405
Machine with R-410	9.568	10.630	11.299	12.017	9.245	9.463

Table 24-Refrigerant charge during winter season

- Cooling

Rating point	A	B	C	D
Machine with R-32	14.362	14.252	14.138	14.088
Machine with R-410	15.874	15.788	15.788	15.842

Table 25-Refrigerant charge during summer season

As mentioned above, the price of heat exchangers is often related to their weight. Table 22 and Table 23 show the difference in weight between batteries. The indoor battery of both systems is quite similar. About 7 kg difference being the lighter R-32. This difference increases more with the battery located on the outside. Those 23 kgs of difference are already considerable.

Thus, concerning the weight of the exchangers, the result is favourable for the R-32. Due to their efficiency, their interchanges are more compact and lightweight. As the R-32 characteristics mentioned before, a lower refrigerant load is required to acquire the same power as the R-410A. That means that the price will be lower, in addition to being this refrigerant cheaper than the R-410A.

5. HEAT RECOVERY

The Real Ordinance 178/2021, 23rd of March, which modifies Royal Ordinance 1027/2007, approves Regulation of Thermal Installations in Buildings (RITE in Spain). That regulation stipulates the climatic systems of buildings in which the air supply is higher than 0.28 m3/s or 1008 m3/h, according to the eco-design normative, heat recovery would be mandatory.[35]

The remarkable part about this process is allowing the system to reuse the potential of the airflow output to reduce machine consumption, in this case, roof-tops.

The RITE regulation states that all the airflow must be recirculated from the conditioned area to analyse the machine efficiency. That means that there is no fresh air renewal for such analysis. This project also aims to design an even more efficient system. In this way, a heat recovery device will be installed in our system to reduce consumption. [35]

5.1. Cooling mode

Eco-design normative indicates the inlet temperatures to the condenser and evaporator. For example, in cooling:

- $T_{in\ evap}$= 27ºC
- $T_{in\ evap}$= 35ºC

The RITE normative indicates that an acceptable work operation would be that part of the conditioned area's air recirculates and mixes with the ambient air. [35]

The operation mode of an AC system can be high-occupation or low occupation. That consist of the supplied air requirements. High-occupation allows recirculating a large amount of supply air, whereas low-occupation mode does not require large quantities. High-occupation mode results in more design inconvenience. The main drawback of the high-occupation design is the change in the temperature jump in the evaporator, since less flow is recirculated, which modifies the working rate of the evaporator, and the rest of the battery design parameters are affected.

Therefore, the designed system is operating in low-occupation mode, which will provide high performance and simpler operations. Then, an acceptable quantity of recirculated air would be 70 % for this kind of operation. Hence, to maintain the airflow, the rest of air must be supplemented with 30% of the air coming from the ambient, as is shown in Figure 64.

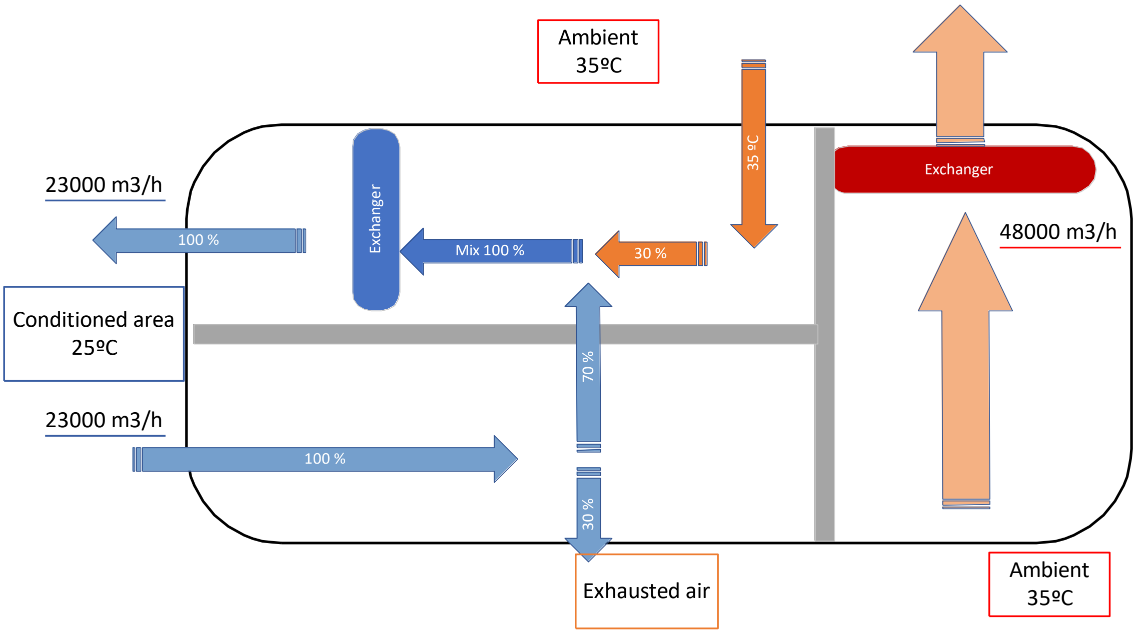


Figure 64-Scheme of air recirculation in the system

It is evaluating the heat balance of the ambient temperatures indicated by the Normative. Those exterior temperatures were 35, 30, 25 and 20 ºC and it is considering a constant interior temperature of 25 ºC.

Then, it is studying the situation of an ambient temperature of 35ºC, and the conditioned area is wanted to maintain the temperature around 25 ºC. The temperature balance is represented in equation (24).

$$T_{evaporator\ inlet} = \frac{0.7 \cdot m1 \cdot T_{int} + 0.3 \cdot m1 \cdot T_{ext}}{m1} \qquad (24)$$

$$T_{evaporator\ inlet} = \frac{0.7 \cdot 23000 \cdot 25 + 0.3 \cdot 23000 \cdot 35}{23000} = 28ºC$$

Now, with recirculation, the inlet temperature of the evaporator has decreased from the initials 35 ºC. It means that the evaporator will need less energy to reduce the temperature to the set point.

It is showing the calculation of the first temperature to clarify the process. The rest of the results are shown in Table 26.

Exterior temperature [ºC]	Interior temperature [ºC]	Evaporator inlet temperature [ºC]
35	25	28
30	25	26.5
25	25	25
20	25	23.5

Table 26-Final temperatures after recirculating part of the air and mix it with ambient air during the summer season

With these calculations, it is observed that the temperatures that would reach the evaporator after having been mixed with a 70 % flow of air from the interior and with a 30 % flow of air of the total 23000 m3/h.

Therefore, the option of installing a heat recovery system and checking the impact it would have on the efficiency of the system is now proposed. Then, the type of recuperator that is going to use is the sensible recuperator. Work is not exchanged, and the potential and kinematic energy maintain constant. The efficiency of the recuperator is going to be defined to analyse how the system performs.

EFI=80%

The main thing is obtaining the temperature that would be the heat exchanger's inlet temperature after the heat recovery process. The efficiency is the ratio between the power given by the system and the maximum power that the system can provide.

Points 1 and 3 are the inlets of the heat exchangers. The minimum would be considered in the formula.

Then, this temperature will mix with the recirculated air and will decrease more the inlet evaporator temperature. The scheme in Figure 65 shows how the system will work.

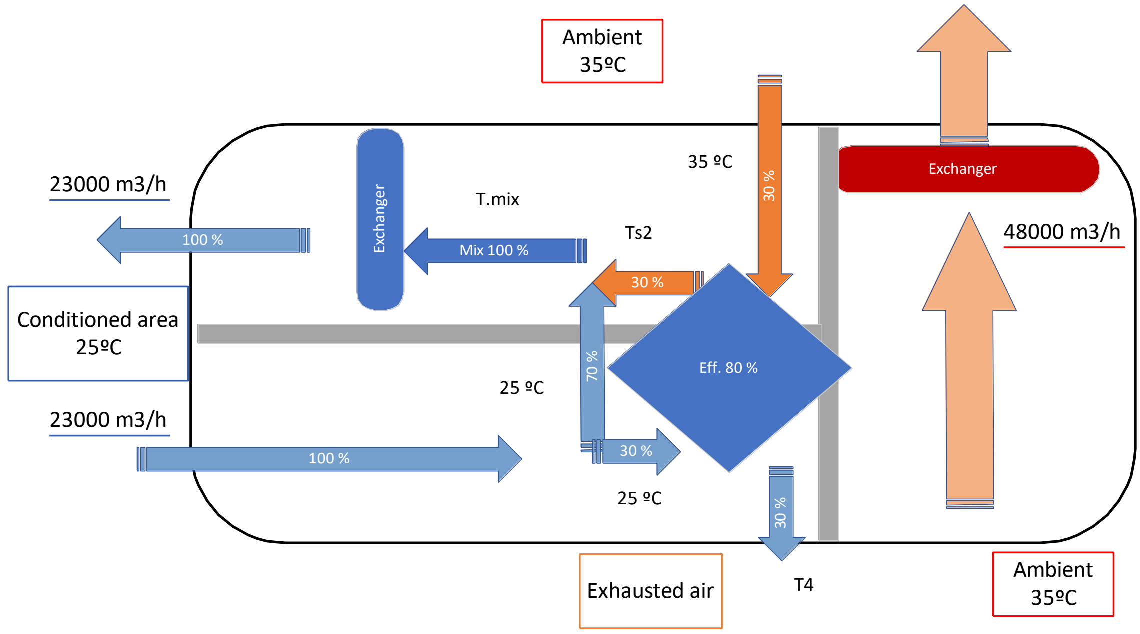


Figure 65-Heat recovery system work operation

$$T_{s2} = T_{s1} + EFI \cdot (T_{s3} - T_{s1}) \qquad (25)$$

$$T_{s2} = T_{s1} + EFI \cdot (T_{s3} - T_{s1}) = 35 + 0.8 \cdot (25 - 35) = 27ºC$$

Then, doing the mix temperature equation again (24), but this time with lower temperature after the heat recovery system shown in (25):

$$T_{evaporator\ inlet} = \frac{0.7 \cdot m1 \cdot T_{int} + 0.3 \cdot m1 \cdot T_{ext}}{m1} = \frac{0.7 \cdot 23000 \cdot 25 + 0.3 \cdot 23000 \cdot 27}{23000}$$
$$= 25.6ºC$$

The evaporator inlet temperatures after the heat recovery are shown in Table 27:

Exterior temperature [ºC]	T_{S2} [ºC]	Interior temperature [ºC]	Evaporator inlet temperature [ºC]
35	27	25	25.6
30	26	25	25.3
25	25	25	25
20	24	25	24.7

Table 27-Temperatures that will arrive to the evaporator after the heat recovery process for summer season

It is introducing these values in IMST-ART. Thanks to the heat recuperator, the temperatures of the evaporator inlet have been reduced. That means some energy the compressor does not need to consume to get to the proposed setpoint. The saved amount of capacity is calculated with equation (26):

$$P_s = \frac{Q \cdot 0.3 \cdot \rho_{air}}{3600} \cdot Cp_{air} \cdot (\Delta T) \quad (26)$$

- Q [m^3/h]→ Airflow =23000 m^3/h
- ρ_{air} [kg/m^3]→ Air density = 1.2 kg/m^3
- Cp [J/kg·K] → Air specific heat = 1 J/kg·K
- ΔT [K] → Temperature difference between ambient temperature and temperature after heat recovery. = (Ambient temperature – T_{S2})

Then, in order to obtain the new ERRd, it is calculated as is shown in equation (27)

$$EERd = \frac{P_s + P}{C} \quad (27)$$

- Ps [kW] → Saved power because of the heat recovery
- P [kW] → Power obtained by the parametric study resulted from the heat recovery
- C [kW] → Compressor capacity

This energy will be reflected with an increase in the EERd. Then, a study from the obtained temperatures is made, it is shown in Figure 66:

	Units	Case 1	Case 2	Case 3	Case 4
Cond. Inlet Temperature	°C	35	30	25	20
Evap. Inlet Temperature	°C	25.6	25.3	25	24.7

Figure 66-Parametric study after heat recovery during summer season

Therefore, the obtained values after the parametric study and substituting in equations (25) and (26) are shown in Figure 67:

Rating point	Compressor					Higher compressor					Lower compressor				
	Declared cooling capacity DC	Capaciticy saved	Compressor power	EERd without counting the saved epower	EERd	Declared cooling capacity DC	Capaciticy saved	Compressor power	EERd without counting the saved epower	EERd	Declared cooling capacity DC	Capaciticy saved	Compressor power	EERd without counting the saved epower	EERd
	kW	kW	kW			kW	kW	kW			kW	kW	kW		
A	109,13	18,40	35,38	2,99	3,60	84,72	18,40	21,89	3,68	4,71	55,21	18,40	8,93	4,70	8,24
B	113,87	9,20	31,92	3,44	3,86	88,48	9,20	19,64	4,26	4,97	56,23	9,20	8,15	5,26	8,03
C	118,49	0,00	28,71	3,97	4,13	92,13	0,00	17,68	4,90	5,21	57,35	0,00	7,50	5,84	7,64
D	123,00	-9,20	25,79	4,57	4,41	95,68	-9,20	15,91	5,61	5,44	58,43	-9,20	6,89	6,49	7,15

Figure 67-Obtained capacity and EERd results after heat recovery process

Then, with the improved EERd, the final seasonal efficiency will be enhanced. It is comparing the results in Table 28:

	With Heat recovery	Without Heat recovery
SEER	4.49	4.49
Q_C	65478 kWh	68076 kWh
Q_{CE}	14603 kWh	15164 kWh
η_{SC}	176.7 %	176.4 %

Table 28-Final comparison with the heat recovery system installed and without it in cooling mode

Frequently, recovery is not done in cooling mode due to minimal improvement. For it to be a substantial improvement, previous processes such as adiabatic cooling should be applied.

However, for this mode, the performance obtained from the machine is notably high-grade. Although installing a heat recovery system does not affect this mode, the system will be highly rewarded in heat mode.

5.2. Heating mode

For the winter season, the procedure to follow is the same as for the cooling mode. The indoor batterie will do the condenser function and the evaporation process will be done by the outdoor exchanger, just the opposite of the cooling mode. The airflow remains the same, just the ambient and the conditioned area change. It is analysing how a heat recovery would affect the performance of the system during the mentioned season.

Firstly, it is calculating the temperatures obtained after installing a heating system. That will give us some temperatures after joining recirculated air with air coming from the outside in the heat recovery system. In addition, this resulting temperature shall be associated with 70 per cent of the recirculated airflow of the conditioned area. The system works as Figure 65 shows. Thus, it will make the temperature that reaches the inner battery higher. This process will make the system much more efficient.

The heat recovery system is supposed to have the same efficiency as the one commented previously, an 80 %.

In the case of heating, it consists of increasing the temperature that arrives at the condenser to reduce the power input that would be needed to reach the desired temperature.

The exterior temperatures to consider will be what the normative indicates. In the first point, an ambient temperature of -7 ºC will be considered. The objective is to maintain an interior temperature of 20 ºC. Hence, equation (24) is calculating the new temperature after obtaining the temperature with heat recovery with equation (25).

- T_{s3}→Temperature in the conditioned area=20 ºC
- T_{s1} → Ambient temperature= -7 ºC

$$T_{s4} = T_{s3} - EFI \cdot (T_{s3} - T_{s1}) = 20 - 0.8 \cdot \left(20 - (-7)\right) = -1.6ºC$$

$$T_{condenser\ inlet} = \frac{0.7 \cdot m1 \cdot T_{int} + 0.3 \cdot m1 \cdot T_{ext}}{m1}$$
$$= \frac{0.7 \cdot 23000 \cdot 20 + 0.3 \cdot 23000 \cdot -1.6}{23000} = 13.52ºC$$

In Table 29, a summary of the temperatures obtained with the previously mentioned processes has been made. It has been made for each of the temperatures indicated in the regulations.

Exterior temperature [°C]	T_{S2} [°C]	Interior temperature [°C]	Condenser inlet temperature [°C]
-7	-1.6	20	13.52
2	5.6	20	15.88
7	9.6	20	16.88
12	13.6	20	18.08
-10	-4	20	12.8
-8	-2.4	20	13.28

Table 29-Temperatures that will arrive to the evaporator after the heat recovery process for winter season

Then, it is running the parametric study shown in Figure 68:

	Units	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6
Cond. Inlet Temperature	°C	13.52	15.88	16.88	18.08	12.8	13.28
Cond. Wet-bulb Temp	°C	8.52	10.88	11.88	13.08	7.8	8.28
Evap. Inlet Temperature	°C	-7	2	7	12	-10	-8
Evap. Wet-bulb Temp	°C	-8	1	6	11	-11	-9

Figure 68-Parametric study after heat recovery during summer season

Therefore, once the software provides the values from the parametric study made, it is substituting in equation (26) to obtain the power saved and (27) to get the improved COPd. Figure 69 shows the results:

Rating point	Compressor					Higher compressor					Lower compressor				
	Declared heating capacity DH	CapacitICY saved	Compressor power	COPd without counting the saved epower	COPd	Declared heating capacity DH	Capaciticy saved	Compressor power	COPd without counting the saved epower	COPd	Declared heating capacity HC	Capaciticy saved	Compressor power	COPd without counting the saved epower	COPd
	kW	kW	kW			kW	kW	kW			kW	kW	kW		
A	73,48	12,42	19,29	3,60	4,45	51,39	12,42	12,05	3,91	5,29	24,26	12,42	4,62	4,27	7,94
B	95,16	8,28	23,19	3,91	4,46	67,11	8,28	13,91	4,47	5,42	32,08	8,28	5,27	5,07	7,66
C	108,59	5,98	25,38	4,09	4,51	77,06	5,98	14,95	4,80	5,56	37,16	5,98	5,66	5,53	7,62
D	122,83	3,68	27,79	4,24	4,55	87,96	3,68	16,20	5,08	5,66	42,81	3,68	6,14	5,94	7,57
E	67,00	13,80	18,13	3,48	4,46	46,76	13,80	11,47	3,77	5,28	22,00	13,80	4,43	4,02	8,08
F	71,35	12,88	18,92	3,56	4,45	49,86	12,88	11,88	3,84	5,28	23,51	12,88	4,53	4,18	8,03

Figure 69-Obtained capacity and COPd results after heat recovery process

Then, with the improved COPd, the final seasonal efficiency will be enhanced. It is comparing the results in Table 30 :

	With Heat recovery	**Without Heat recovery**
SCOP	5.77	4.07
Q_H	140000 kWh	140000 kWh
Q_{HE}	24247 kWh	34425 kWh
η_{SH}	228 %	160 %

Table 30-Final comparison with the heat recovery system installed and without it in heating mode

The results indicate the numerous advantages of installing a heat recovery system. Its function during the winter season is significantly higher than for the summer season. The reason the recuperator has more effect in the heating mode is the fact that the temperatures reaching the indoor battery have increased considerably comparing it with the ambient. That means the machine will not need as much power to drive air into the conditioned area to the established setpoint.

Thus, although the heat recovery system is an additional investment, it represents a superior increase in system efficiency. That will offset by savings in electricity consumption. In a short period, this device will be amortized due to its great functionality and the benefits obtained.

6. CONCLUSION

The result of this project is to design an efficient roof-top heat pump as dictated by RITE. In addition, this project proposes to use a refrigerant much more environmentally friendly than those normally used for this type of systems, such as the R-410 A. This refrigerant has a GWP around of 2000 and, even though for the time being the regulations do not prevent its use in the high capacity pump, its price will be increasing due to the high pollution rate. The result of this project is to design an efficient roof-top heat pump as dictated by RITE.

In this study, many refrigerants have been analysed. Many would be potentially valid, but due to their price, flammability or lack of development, they have not been considered adequate for this project. Thus, the refrigerant selected for the design of this system was the R-32. Although it is A2L, little by little has been recognized for its high efficiency and easy use. This refrigerant has numerous advantages such as its efficiency, low GWP (650) and low refrigerant load. More and more companies are betting on this fluid, especially for smaller heat pumps.

Designed the batteries and selected the compressors and other accessories for the system operation with this refrigerant, its performance has been evaluated according to the Eco-design regulations. The designed machine far exceeds the minimum efficiencies described in the standard. Therefore, in terms of efficiency, the device with R-32 can be considered suitable. However, it has been decided to compare it with a machine using R-410A and evaluate the results.

The comparison favours the design with R-32. The efficiency in cooling mode is very similar, but in heating mode, the machine with R-32 has much more efficiency and hence, less consumption. The weight of the batteries is also very positive for the R-32. Batteries designed for this coolant are lighter than for the R-410A. Going the weight of these batteries related to the price means that the machine costs are more economical. Therefore, it could be said that a roof-top of these characteristics that works with R-32 would be a valid alternative for study. In addition to this, consideration has been given to introducing a heat recovery system to improve efficiency by reducing consumption. This system favourably affects machine performance, especially heating mode.

There is currently no conclusive alternative to replace the R-410A. Many companies are investing heavily in researching different possibilities. This project, based on efficiency, has considered the R32 as a possible option. The results indicate that it is a potentially reliable choice. Daikin and Danfoss are large companies within the field that are also betting on their high capacity heat pumps for this refrigerant.

The comparison made in the project reflects the excellent performance of the system with this refrigerant. The disadvantage of the R-32 is its low flammability. That makes it necessary to have expensive components because of the safety required in this type of facility. Despite everything, more and more people are betting on this refrigerant and the elements are becoming more and more accessible. For large capacity areas, it is relatively easy to deal with the flammability of the R-32. That is why it is proposing in many fields to use this refrigerant for systems in supermarkets and factories.

Therefore, this project concludes that a system of these capabilities using R-32 is a valid alternative. The low price of the refrigerant and its high efficiency compared to the R-410A will make this system cost-effective and highly efficient. Not only that, but it is also a much greener system because of its low GWP.

7. BUDGET

The budget is the document in which the costs incurred during the research project will be reflected.

It is a research project with the aim of proving the machine designed with R-32 could be a potential option for the future of rooftops and AC systems. The fact is that nowadays, there is not a definitive refrigerant that substitutes the R-410A. Many companies and engineers are studying many possibilities and variables to get cost-effective and efficient results.

Hence, the scope of this budget cannot be the costs of building the system. That task would need a high level of expertise in the field. Some of the requirements would be contacting suppliers and experts to gather enough information. It is because this kind of budgets depends on a significant number of conditions. Some examples are the installation, the steel, the cover, isolations and many more duties. Besides, proper benchmarking would be needed to acquire the most affordable and high-quality components of the system. It demands plenty of time and resources, which this project cannot afford.

However, from the obtained results and the knowledge gained, a general conclusion can be added without considering the selected components from different companies' software. Aspects about the prices that could be achieved by heat exchangers and the refrigerant charge will be discussed.

Most of the times, the price of the batteries is related to the weight. The comparison made previously shows the cost of the exchangers designed would be more affordable than the regular batteries working with R-410A. Furthermore, even though the R-410A is not banned from the Montreal Protocol, the costs related to its use have increased. The F-gas Regulation (EU) 517/2014 has defined taxes for using refrigerants with elevated GP. Therefore, despite the R-32 is an A2L refrigerant and requires safety components by normative, it is starting to be affordable. R-410A will no longer be a cost-effective refrigerant due to the high costs of its acquirement and the taxes expected. In addition, R-32 is growing in popularity due to its advantages, and more companies are considering it. Thus, access to materials and components will become easier due to high demand, which will reduce costs.

To conclude, now there is not a final substitute for refrigerant R-410A. Nevertheless, the study made indicates the system designed working with R-32 could be a potential option not only because of its benefits but also because it has a similar price or below of the standard R410A system.

In addition to this, information is to be included on the costs that would result from commissioning such a project from an engineer.

Because this project is a research project, most of the costs will be derived from human resources. The other potential quantity is originated from computer resources.

Because this project is a research project, most of the costs will result from human resources. Computer resources arise the other potential quantity.

Table 1 reflects the IT costs. This table represents the costs given by the purchase of a PC in which the project is developed. During the project, the software used to design and simulate the machine has been the IMST-art software. A license to get this software is what is represented in the table. Besides, the document has been created in a Microsoft Word document. That means that the license is a cost that has to be included in Table 31.

Concept	Prize	Date of purchase	Amortization period (years)	Time of use	Amortized period (months)	Amortization Cost
PC ASUS F556U-TJ Intel Core i5	550 €	06/04/2020	3	06/05/2021	13	198,61 €
Software IMST-art	3000 €	06/04/2020	5	06/05/2021	13	650,00 €
Microsoft Office	200 €	06/04/2020	2	06/05/2021	13	108,33 €

Table 31-Depreciation of the computer part

Costs related to human resources include the time spent by the engineer in the design process. Regarding the engineer, due to the lack of experience, he is considered a junior engineer. The approximated time spent has been 300 hours designing with a prize of 40€ per hour worked, considering working hours spent by a junior engineer.

Some senior engineer working hours will be included. The senior engineer helped the junior engineer with the following activities:

- -Heat exchangers design with IMST-art. 8 sessions of an hour.
- -Follow-up tutorials, ten meetings of approximately 1-2 hours each approximately.

All of the human resources costs are represented in Table 32:

Concept	Sessions	Cost per hour [€]	Total cost
IMST-art training course	8	75,00 €	600,00 €
Personal tutorials	10	75,00 €	1125,00 €

Table 32-Costs arising from human resources

To all the comments above, added costs of materials and resources needed to develop the project. This project has been carried out from a personal office. It is considered that the room already belongs to the engineer doing the project, so it will not involve an additional cost. The costs that are considering are office equipment and internet connexion. About office equipment are counting the desk and the chair with which the engineer is going to work. The internet connexion is only considering the months of work, not counting holidays. The costs of resources are explained in Table 33.

Concept	Supplier	Cost per month	Total cost
Desk	IKEA	-	300,00 €
Chair	IKEA	-	250,00 €
Internet connection	Vodafone	25,00 €	225,00 €

Table 33-Costs of resources and materials

Table 34 will show a final table that will summarise the whole costs generated by this research. On the table will appear the costs of IT amortization, human resources and material. These would be the most general costs considered in this project.

Concept	Costs
PC ASUS F556U-TJ Intel Core i5	198,61 €
Software IMST-art	650,00 €
Microsoft Office	108,33 €
IT amortization	**956,94 €**
Senior engineer	1725,00 €
Junior engineer	12000,00 €
Human resources	**13725,00 €**
Desk	300,00 €
Chair	250,00 €
Internet connection	225,00 €
Office equipment	**775,00 €**
Total budget	**15456,94 €**
IVA (21%)	3245,96 €
Total budget by contract	**18702,90 €**

Table 34-Final budget

Therefore, that would be an approximate estimate of research work done by a junior engineer. Such projects are complex to budget because of the hours the engineer has been working on the project. In this case, the hours in which the engineer has been investigating on his own about the heat pumps and their operation have not been counted.

8. CONTRACT DOCUMENT

The contract document consists of determining the minimum required conditions to implement the project. This document is not the most significant regarding research projects. It is notably more prominent in projects that are thought to be manufactured.

The project aims to present a non-usual roof-top heat pump with a not common refrigerant used.

Therefore, the objective of the project's contract document is to explain the purpose of its realization.

The current roof-top heat pumps' market does not offer a wide variety of new generation refrigerants. The most common refrigerants seen in the market are R-134a and R-410A. Those are refrigerants that have great features. Both refrigerants are very easy to handle because they are non-flammable. Besides, they provide excellent efficiency.

Nowadays, halogenated refrigerants represent a significant menace to the environment due to their high GWP. Thus, after the Kyoto, Montreal and Paris protocol, those refrigerants must be reduced and not far in time, entirely eradicated from modern machines.

Now, there are not many roof-top heat pumps working with the new generation's refrigerants. Low-power heat pumps are starting to grow in residential applications. They have shown their tremendous potential and will have more impact in the next few years.

Thus, this project has the purpose of explaining how to design a roof-top heat pump since the beginning. It is comparing different refrigerants and batteries configurations in order to design a machine that complies with the Normative.

In this project, IMST-ART software has been used to obtain all the values provided in the technical memory. This software is particularly suitable to simulate the machine's performance. It is an advanced performance simulation, computer-aided engineering design system. IMST-ART combines accurate and fast algorithms, an easy-to-use graphical interface and powerful analysis capabilities into a single software package suitable for modelling any vapour-compression refrigeration system operating with almost any refrigerant and secondary fluids.

Through quick and easy simulations, a much more optimal configuration of the equipment can be obtained. That will considerably reduce costs due to possible modifications due to poor performance or design failures.

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