

KoolSys waste heat recovery system for Domestic Hot Water and/or Heating Ventilation and Air Conditioning

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Resumo

Com a crescente preocupação a nível mundial da eficiência energética, as empresas fabricantes de sistemas de refrigeração vêm-se cada vez com a necessidade de se adaptarem e melhoraram os seus sistemas. A RACE não é uma exceção, e nesse sentido, após o desenvolvimento da central de refrigeração KoolSys, que tem como fluido de trabalho CO₂, pretenderam analisar a viabilidade de incorporar um sistema de recuperação de calor dissipado pela mesma e usá-lo para água quente sanitária e /ou aquecimento ventilação e ar condicionado.

Neste sentido, este trabalho teve como objetivo o estudo de integração de um sistema de recuperação de calor, nas centrais da empresa tanto a nível energético como financeiro.

Realizou-se primeiramente uma revisão de literatura em que foram explorados os trabalhos publicados relativos aos sistemas de refrigeração transcritos, com fluido de trabalho o CO₂, os seus diversos componentes, e ainda as possibilidades de recuperação de calor desse tipo de sistemas.

Para avaliar a viabilidade do sistema, foram recolhidos e tratados dados obtidos da central em estudo, como temperaturas e pressões à entrada e saída do gas cooler, bem como entalpias e caudais respetivos, de forma a perceber a quantidade de energia disponível. Os dados foram tratados durante o período de julho de 2022 a janeiro de 2023. Tendo sido, posteriormente comparada a energia disponível com as necessidades de água quente sanitária e aquecimento ventilação e ar condicionado do espaço em análise. Permitindo, por fim, definir um sistema de recuperação de calor tendo em conta os resultados obtidos.

A nível energético foi possível compreender que não haveria qualquer necessidade de armazenamento de energia, tendo em conta que nos meses estudados, havia constantemente energia suficiente para colmatar as necessidades existentes para o mesmo espaço temporal. A nível financeiro conclui-se que a empresa apenas teria retorno do seu investimento num espaço de 4 anos e 8 meses.

Abstract

With the growing worldwide concern for energy efficiency, companies that manufacture refrigeration systems are increasingly faced with the need to adapt and improve their systems. RACE is no exception, and in this sense, after developing the KoolSys refrigeration plant, which has CO₂ as a working fluid, they wanted to analyze the feasibility of incorporating a heat recovery system dissipated by the same and use it for domestic hot water and / or heating ventilation and air conditioning.

In this sense, the objective of this work was to study the integration of a heat recovery system in the company's power plants, both at the energy and financial levels.

The first step was a literature review in which the published works related to the transcribed refrigeration systems were explored, with the working fluid CO₂, its various components, and also the possibilities of heat recovery from this type of systems.

To evaluate the feasibility of the system, data obtained from the plant under study was collected and processed, such as temperatures and pressures at the gas cooler inlet and outlet, as well as enthalpies and flow rates, in order to understand the amount of energy available. The data was processed during the period July 2022 to January 2023. The available energy was then compared with the domestic hot water, heating, ventilation and air conditioning demand of the space under analysis. Allowing, finally, to define a heat recovery system taking into account the results obtained.

On an energy level it was possible to understand that there would be no need for energy storage, taking into account that in the months studied, there was constantly enough energy to meet the existing demand for the same period of time. On a financial level it was concluded that the company would only have a return on its investment within 4 years and 8 months.

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Contents

1	Introduction.....	11
1.1	Context.....	11
1.2	Short description about RACE company	15
1.3	Objetives	16
1.4	Structure	17
2	Literature review about refrigeration systems and waste heat recovery systems	18
2.1	Refrigeration systems.....	18
2.2	Vapour compression refrigeration systems.....	18
2.3	Transcritical refrigeration cycle with CO ₂	20
2.2.1	Compressors	22
2.2.2	Gas cooler	26
2.2.3	Expansion valves	29
2.2.4	Evaporator	33
2.4	Heat released by refrigeration cycles through the environment	35
2.5	Industrial refrigeration	39
3	KoolSys system model and waste heat recovery system.....	41
3.1	Data Assembly	41
3.2	Characterization of KoolSys plant considered in this case study.	42
3.3	Methodology	44
4	Results and discussion	46
4.1	Results.....	46
4.2	Waste heat recovery system.....	48
5	Conclusions.....	52
	References	54
	ANNEX A: CO ₂ flow rate in the system.....	57
	ANNEX B: Results obtained from the data collected	61

Nomenclature

Abbreviations

CO ₂	Carbon dioxide
COP	Coefficient of performance
DHW/AQS	Domestic hot water
EER	Energy efficiency ratio
EEV	Electronic expansion valve
EU	European Union
GWP	Global warming potential
HVAC/AVAC	Heating ventilation and air cooling
IEA	International energy agency
SCOP	Seasonal coefficient of performance
TEV	Thermostatic expansion valve

Symbols

Designation	Definition	Units
Δh_{EVAP}	Enthalpy variation in the evaporator	[kJ/kg]
Δh_{COMP}	Enthalpy variation in the compressor	[kJ/kg]
H	Enthalpy	[kJ/kg]
$\dot{m}$	Mass flow rate	[kg/s]
P	Pressure	[bar]
P _w	Power	[kW]
T	Temperature	[°C]
W _c	Compressor Work	[kW]

Figure Index

Figure 1: Industrial heat consumption in the EU classified by industrial section (Eurostat, 2015 data) [6].....	12
Figure 2: Annual energy consumption in industry in EU countries (left axis) and totals for the EU (red line – right axis) (Eurostat 2005 – 2015 data) [6].....	13
Figure 3: Global air-conditioning stock in millions of units until from 1990 - 2016 and projections until 2050 [9]	14
Figure 4: CO2 emissions from, and emissions intensity of, air conditioning in the Net Zero Scenario, 2000-2030 [9]	15
Figure 5: Scheme of a vapour compression cycle	19
Figure 6: T-s and P-h diagrams of a vapour compression cycle [13].....	19
Figure 7: Transcritical CO2 refrigeration cycle process with main components [14]	20
Figure 8: Phase diagram of R744 (CO2) [14]	21
Figure 9: Subcritical (left) and transcritical (right) refrigeration cycle process [14]	21
Figure 10: A typical hermetic reciprocating compressor [13].....	22
Figure 11: Semi hermetic reciprocating compressor [13]	23
Figure 12: Open type reciprocating compressor [13].....	23
Figure 13: Compressor types (Heap, 1979) [13]	24
Figure 14: Influence of the gas cooler temperature and pressure on cycle process [17].....	27
Figure 15: Effect of gas cooler pressure on refrigeration capacity and COP [14]	28
Figure 16: Principle of gas cooler to minimize the refrigerant exit temperature [14].....	28
Figure 17: Parts of a thermal expansion valve [17].....	30
Figure 18: Capillary tube [17]	31
Figure 19: Electronic Expansion Valve [13]	32
Figure 20: Room type and Large-scale industrial type air coolers [13]	34
Figure 21: Shell-tube type evaporator [13]	35
Figure 22: Waste Heat potential per industrial sector and temperature level for EU industry in 2015 [6]	36
Figure 23: Waste heat potential in each EU country per temperature level in food industry [6]	36
Figure 24: KoolSys system and componentes	42
Figure 25: Average energy released by a day in each month [kWh/day]	47
Figure 26: Energy requirements by hour for heating space in kWh.....	48
Figure 27: Waste heat recovery system [AutoCad].....	49

Table Index

Table 1: Temperature range of waste heat recovery and their application [7]	38
Table 2: Climate data of a county in the centre of Portugal [20]	41
Table 3: Quantities and prices of the components needed to the wasted heat recovery system.	51
Tabel A 1: Regression coefficients coefficients of a compressor at 25Hz in transcritical mode	58
Tabel A 2: Regression flow rate coefficients of a compressor at 70Hz in transcritical mode .	59
Tabel A 3: Regression power coefficients of a compressor at 25Hz in transcritical mode.....	59
Tabel A 4: Regression power coefficients of a compressor at 70Hz in transcritical mode.....	60
Table B 1: Data collected by RACE from the Koolsys system.....	61
Table B 2: Data calculated from the KoolSys collected data	61
Table B 3: Released hourly energy for an average day of each month in kWh.	62
Table B 4: Released average energy for each day of each month [kWh]	63
Table B 5: Heating loads requirement by day of each month in kWh (Simulated)	64
Table B 6: Heating loads requirement by day of each day of each month in kWh (Simulated)	65
Table B 7: Heating Loads requirements for HVAC by month in kWh (Simulated)	66
Table B 8: Hourly available energy after retrieving the demand for heating loads [kWh]	66
Table B 9: Hourly consumption profile of hot water during an average day of each month in litres (Simulated)	67
Table B 10: Heating water requirements by month for AQS system in kWh (simulated).....	67
Table B 11: Seasonal coefficient of performance (SCOP) of the HVAC system	67

1 Introduction

1.1 Context

Nowadays, one of the major concerns in Europe and the entire world, is the energy efficiency of equipment and processes, especially in HVAC and refrigeration systems.

Due to the breakout of the war in Ukraine in 2022, it became more urgent that the energy consumption should be more efficient and greener than it was, making extremely important that the world dependency of fossil fuels and natural gas should be replaced by renewable energy sources.

In March of 2007 the political leaders of the European Union agreed that 20% of the energy consumption should be supplied by renewable energies [1]. This goal was in fact exceeded, since the end of 2020, 25,8% of the primary consumption of energy consumption (demand of total energy by a country) was, provided by renewable energies [2].

The European Union is also committed to develop an ambitious climate policy for 2050. The goal is that Europe became the first continent to remove the same amount of CO₂ as it produces. The intermediate objective is to reduce the CO₂ emissions in 55% until 2030, which influence in almost all the sectors, transports emissions, agriculture, among others [3].

According to the International Energy Agency (IEA), 40% of the total energy is consumed by the industry sector. As the global economy and population grow, so does the demand for materials and goods, which increase the importance of understanding which technologies and strategies can support the sustainable manufacture [4].

Due to the globalization, supermarkets had an exponential growth rate. Supermarkets, in developed countries are one of the biggest energy consumers, due to refrigeration, HVAC and illumination. The refrigeration systems in supermarkets are responsible for 50% of their total consumed energy [5].

Figure 1 represents the heat consumption, from EU industries in 2015. As expected, industries like iron and steel, chemicals, food, drink, and tobacco are the biggest heat consumers in the EU, in contrast to those like textiles and leather, wood products, and construction. Therefore, as the Koolsys is a refrigeration system that operates with food and drinks, theoretically, it has a large amount of heat consumption.

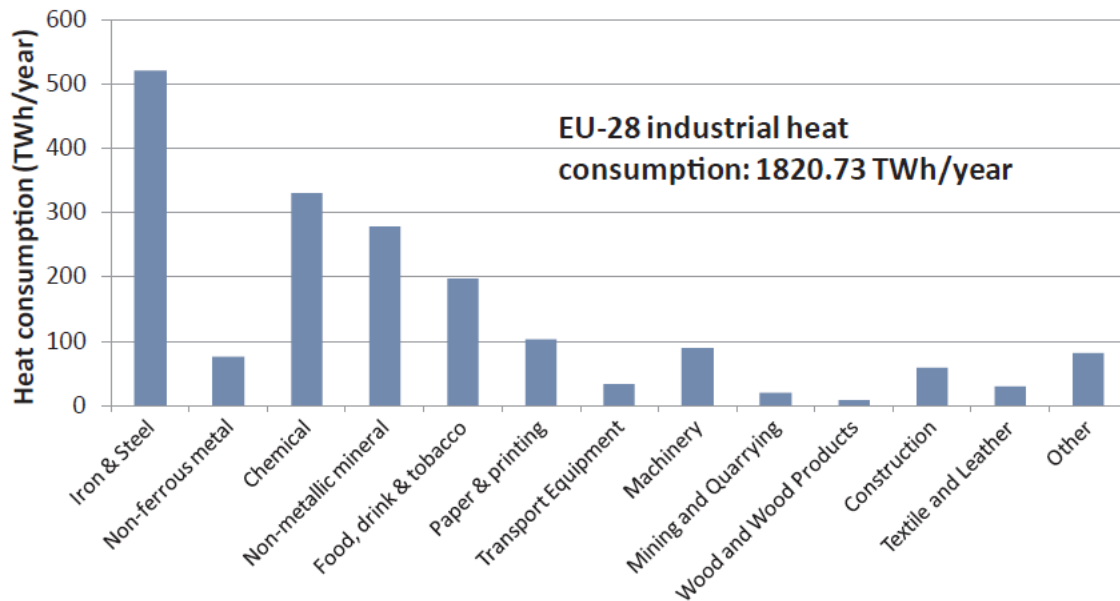


Figure 1: Industrial heat consumption in the EU classified by industrial section (Eurostat, 2015 data) [6]

The industrial energy consumption as a share of the total energy consumption has declined since 2000 as the result of the following three reasons [6]:

- 1) Energy efficiency measures have been implemented in the industry.
- 2) Changes in the type of industries that operate within the EU.
- 3) Increasing energy consumption in other sectors, mainly in buildings, services, and transportation.

Figure 2 shows the annual energy consumption in European Union countries, from 2005 to 2015. It is possible to understand from the figure above that countries like France, Italy, and the United Kingdom have been decreasing their energy consumption across the years, in contrast with countries like Germany that have struggled to decrease their energy consumption due to their industries. Portugal, has not had a significant amount of energy consumption when compared to most of the countries in the EU, but it has been similar across the years.

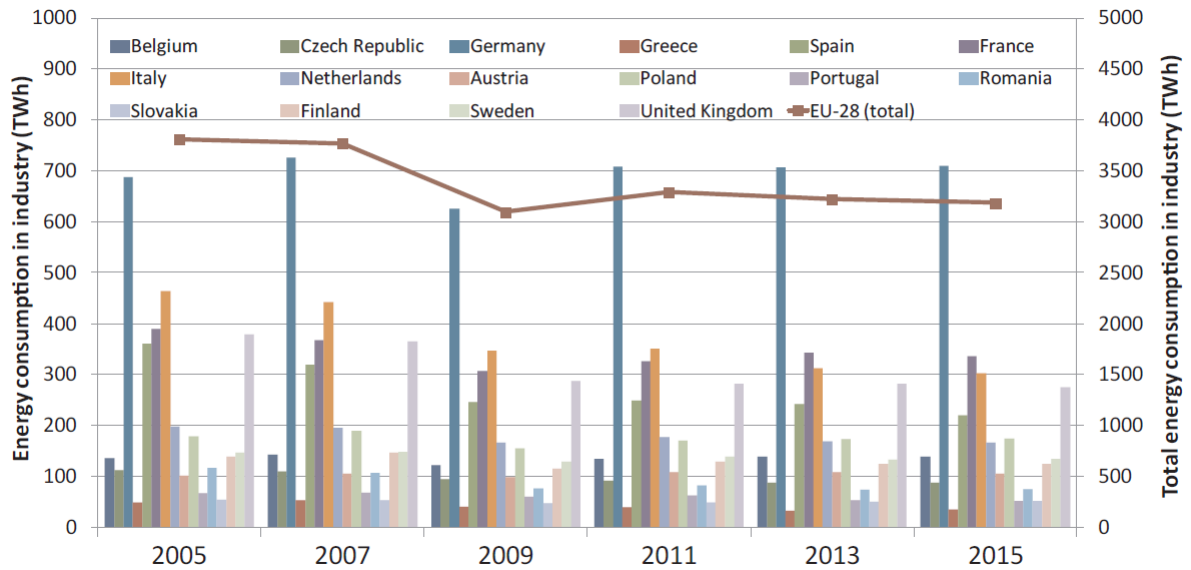


Figure 2: Annual energy consumption in industry in EU countries (left axis) and totals for the EU (red line – right axis) (Eurostat 2005 – 2015 data) [6]

Besides increasing the share of renewable sources, it is necessary to optimize the thermal energy use in the industrial sector. Waste heat recovery is one way to reduce the energy consumption [7]. According to estimates, 20–50% of the energy used in industrial processes is lost as waste heat together with energy losses in electromechanical devices like pumps and motors. The way that heat would be storage will be fundamental for the future applications that he can have [6,8].

In nowadays, the cooling needs are of primary importance, therefore, it is of ultimate importance to understand the efficiency standards in the air-conditioning units since the increase in air-conditioning demand a consequence of the global warming.

According to the International Energy Agency (IEA) without efficiency gains, space cooling energy use could more than double between now and 2030 due to increased activity and use of air-conditioning, as shown in Figure 3 [9].

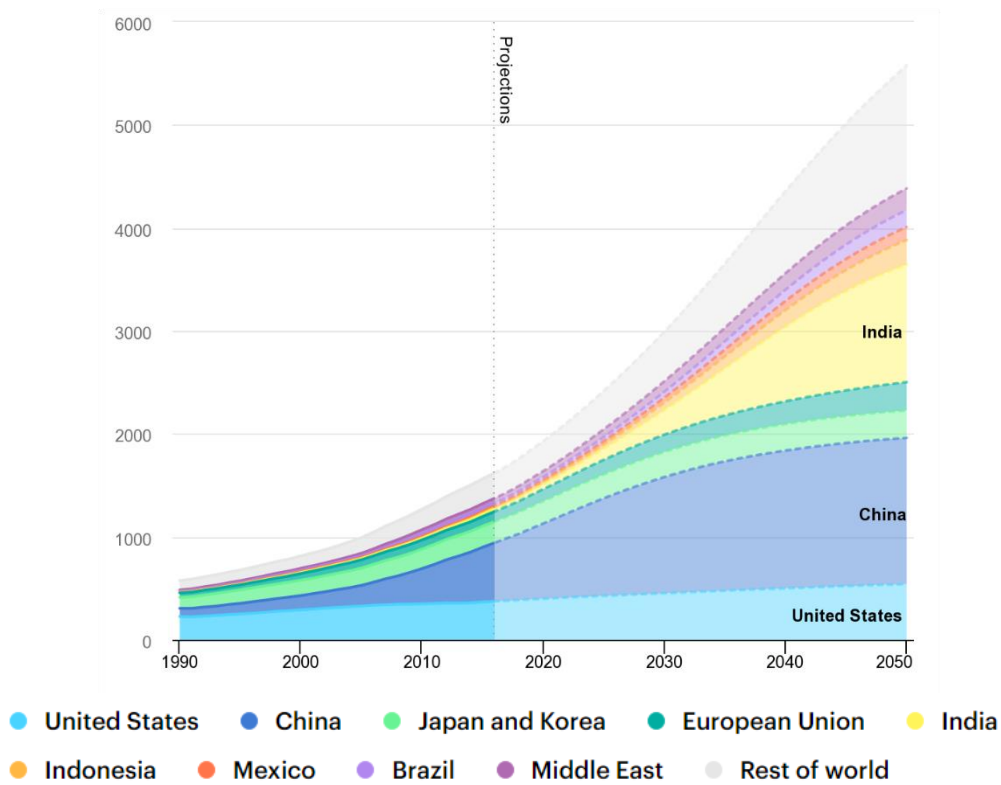


Figure 3: Global air-conditioning stock in millions of units until from 1990 - 2016 and projections until 2050 [9]

CO₂ emissions are predominantly indirect emissions from electricity generation. Emissions intensity is the CO₂ emissions per air-conditioning unit. Efficiency standards are a key measure to avoid the lock-in of inefficient air-conditionings in the coming decades.

To be in line with the Net zero scenario, the average efficiency rating of new air conditioning units would need to increase at least 50% by 2030 in all markets [10].

In this scenario, indirect CO₂ emissions associated with space cooling demand by 2030 fall to about one-third of those in 2021, with a decrease in emissions intensity per unit six times faster than over the past decade, as shown in Figure 4 [10].

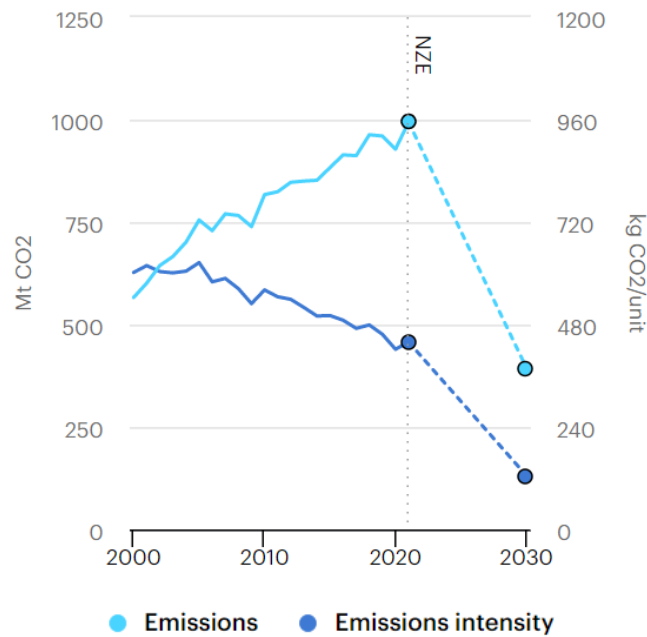


Figure 4: CO2 emissions from, and emissions intensity of, air conditioning in the Net Zero Scenario, 2000-2030 [9]

1.2 Short description of RACE

Race stands for Refrigeration and Air Conditioning Engineering and it is a company with offices in the Algarve, Coimbra, Estremoz, Lisboa and Porto. It is registered with a capital of €3 950 000. The company headquarters is in Porto, where, around 100 people work.

After a path trodden for over three decades as a reference in the Refrigeration and Air Conditioning sectors, RACE decided to take an ambitious step forward and redefine the future of the organisation.

RACE is specialized in designing large-scale HVAC systems, responding with distinction to all associated technical and environmental standards, and using the most advanced solutions for energy efficiency.

RACE has a high level of experience installing and maintaining HVAC systems for diverse locations like shopping centers, supermarkets, hotels, and offices. It is a company with more than 30 years of experience and more than 5000 commercial or industrial installations. RACE gives their services to HVAC systems but also to refrigeration projects, whether it is for industrial refrigeration or commercial refrigeration.

The subject of the present work was proposed by the department of HVAC project from RACE, as a case study. It is based on the KoolSys system, that began to be thought in 2019 but only in the beginning of 2020 it started to be manufactured, until nowadays. The main purpose was to make the brand more competitive on the refrigeration and HVAC market.

KoolSys is a transcritical CO₂ refrigeration plant, that due to a lower condensation temperature (31 °C), it is almost impossible to reject heat by condensation. Therefore, it is needed to use a gas cooling instead of a condenser. With a pressure above the critical pressure (73.8 bar), the gas cooler makes possible to reject heat for the environment. This system is mostly used in medium/large size supermarkets.

This dissertation was proposed in order to understand the possibilities of a heat recovery system that would be use wasted heat by the system and how it could be used as a way to save energy and economic expenses, both from the point of view of the company and the client.

1.3 Objectives

The main goal of this work was to evaluate the potential for waste heat recovery from a trans critical CO₂ refrigeration cycle. The waste heat recovered from the cooling cycle could be used for domestic hot water, and production of indoor heating lowering the energy consumption and the carbon footprint of these system.

This work focused on a specific installation of a KoolSys system in a supermarket in the centre of Portugal. It was evaluated, the released energy from the gas cooler of the system during 7 months, from July/2022 to January/2023 with measured data, in order to understand all the possible variations of heating demand and heat losses to the environment.

Inside this data it was quantified:

- Available heat that is rejected for the system through the environment.
- Demand of HVAC, DHW and/or indoor spacing heating of the space in study.
- Possibility of storage heat that will not have immediate use and possible further use.
- Financial analyses over a waste heat recovery system.

1.4 Structure of the dissertation

This dissertation is divided in five main chapters:

Chapter 1 presents the context of this dissertation, explaining the concerns about the energy consumption and environmental impacts. It also gives a brief explanation about the company, RACE, where this work was carried out, finally there is an explanation about the goals and the methodology used during this work.

Chapter 2 contains the literature review, regarding existing refrigeration systems and the solutions that were already applied in them. It has a brief explanation about the components that are part of a vapor compressor cycle as is the KoolSys system, and clarifications about what is waste heat recovery and how it is possible to recover it, use it or store it until needed.

Chapter 3 presents an explanation of what has been done and all the processes that occur, how it is possible to use a waste heat recovery system, and the advantages and disadvantages of using it with a KoolSys system, for DHW and/or HVAC. It will also include the value of parameters like SCOP (Seasonal coefficient of performance), COP (Coefficient of performance), heating demand, the amount of waste heat available and if it is possible to use it in short-term or long-term, being the storage necessity evaluated.

Chapter 4 is where are discussed the results of the project and the analyses of those results. This analysis will be based on some parameters like investment in the recovery system and a storage unit if needed, available wasted heat and heating demand. That will allow us to understand the reliability of a KoolSys system with a waste heat recovery system to heat up the space.

Chapter 5 presents the conclusions and future works in consequence of the findings of this dissertation.

2 Literature review about refrigeration systems and waste heat recovery systems

2.1 Refrigeration systems

The main goal of a refrigeration cycle is to remove heat from a low temperature source (cold source) to a high temperature source (hot source).

This type of systems, initially it has a lower temperature than the low temperature source, this allows the working fluid to absorb heat during the process. After, the temperature of the fluid will increase, realising heat on the hot source since the fluid temperature is bigger than the hot source temperature.

Refrigeration cycles can be classified as [11]:

- Vapour compression cycle.
- Absorption cycle.
- Inverse Joule Bryton.
- Ejector cycle.
- Adsorption Cycle.

In this project, the refrigeration cycle used is the vapour compression cycle, since it is the cycle that KoolSys plant operates and therefore it is the only which is going to be approached.

2.2 Vapour compression refrigeration systems

The main components of a simple vapour compression cycle are an evaporator, compressor, a condenser, and an expansion valve, as shown in Figure 5.

A more detailed explanation of the steps is explained below [12].

- 1) Compression:** The refrigerant enters the compressor at low temperature and low pressure, in a gaseous state. Here the compression takes place to raise the temperature and the pressure. This process requires work (W_C), may be used an electrical motor. Compressors can be scroll, screw, centrifugal or reciprocating types [12].
- 2) Condensation:** The condenser is essentially a heat exchanger. Heat is transferred from the working fluid to a flow of water. As the working fluid flows through the

condenser, it is in a constant pressure. In this equipment is necessary to have a pressure control device for safety and efficiency reasons [12].

- 3) **Throttling or Expansion:** When the working fluid enters in this device, it expands and release pressure, consequently the temperature drops. The working fluid leaves the valve as a liquid vapor mixture. This device plays two crucial works in the vapor compression cycle. First, they maintain a pressure differential between low- and high-pressure levels. Second, they control the amount of liquid refrigerant entering the evaporator [12].
- 4) **Evaporation:** At this stage the refrigerant is at a lower temperature than its surroundings. Therefore, it evaporates and absorbs latent heat of vaporization. Heat extraction from the refrigerant happens at low pressure and temperature. Compressor suction effect helps maintain the low pressure [12].

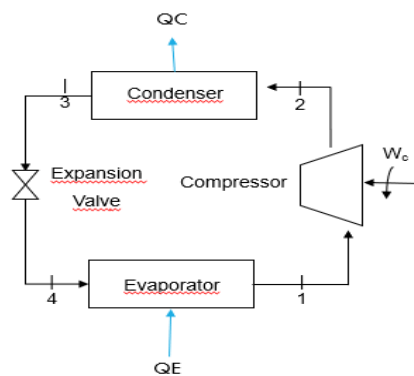


Figure 5: Scheme of a vapour compression cycle

Figure 6 shows, a vapour compression cycle works with two pressure levels, high and low pressures, it has a high pressure in the condenser and low pressure in the evaporator.

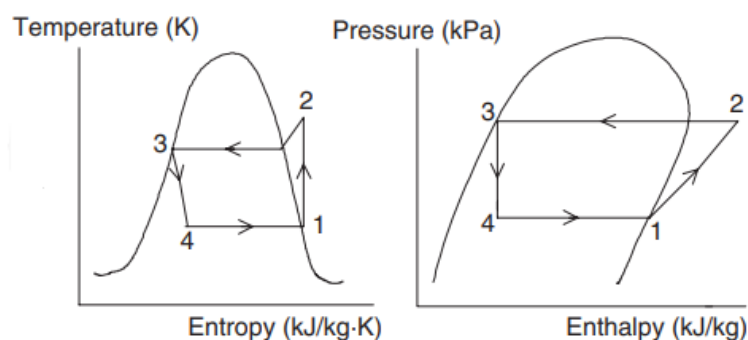


Figure 6: T-s and P-h diagrams of a vapour compression cycle [13]

2.3 Transcritical refrigeration cycle with CO₂

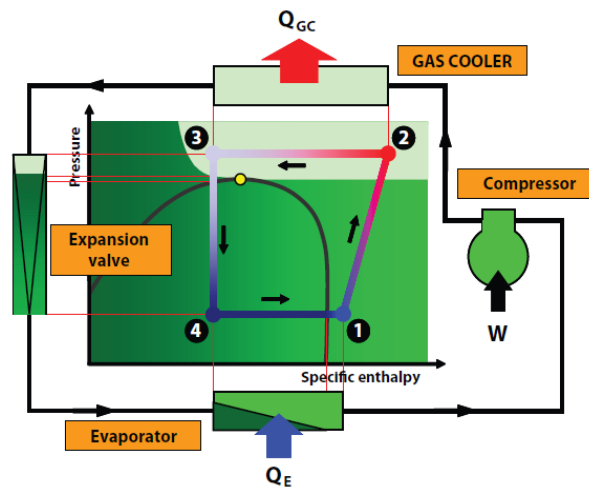


Figure 7: Transcritical CO₂ refrigeration cycle process with main components [14]

Figure 7 shows a transcritical system (vapor compression cycle), where the working fluid is CO₂ and the heat rejection process occurs in the gas cooler, at pressure above the critical point [14].

The use of carbon dioxide has the advantage of being environmentally friendly with reduced greenhouse effect, toxicity, flammability, or impact on the ozone layer [14]. Therefore, CO₂ is a natural refrigerant fluid, and it is non-toxic and non-inflammable [14]. It is characterized for a low global warming potential (GWP=1) and has excellent thermodynamic properties [14, 15].

The critical point, presented in Figure 8, is the thermodynamic condition as of liquid and gas have the same density. In the case of CO₂ this point occurs at a temperature of 31°C and a pressure of 73.8 bar. Above this point is not allowed any condensation of the fluid.

The triple point, presented in Figure 8, represents the thermodynamic condition in which the three states (gas, liquid and solid) exist simultaneously. In the case of CO₂, this occurs at a pressure of 5.2 bar and a temperature of -56.6°C. This condition can originate, in some cases, with an abrupt decrease in pressure, formation of dry ice [14, 15].

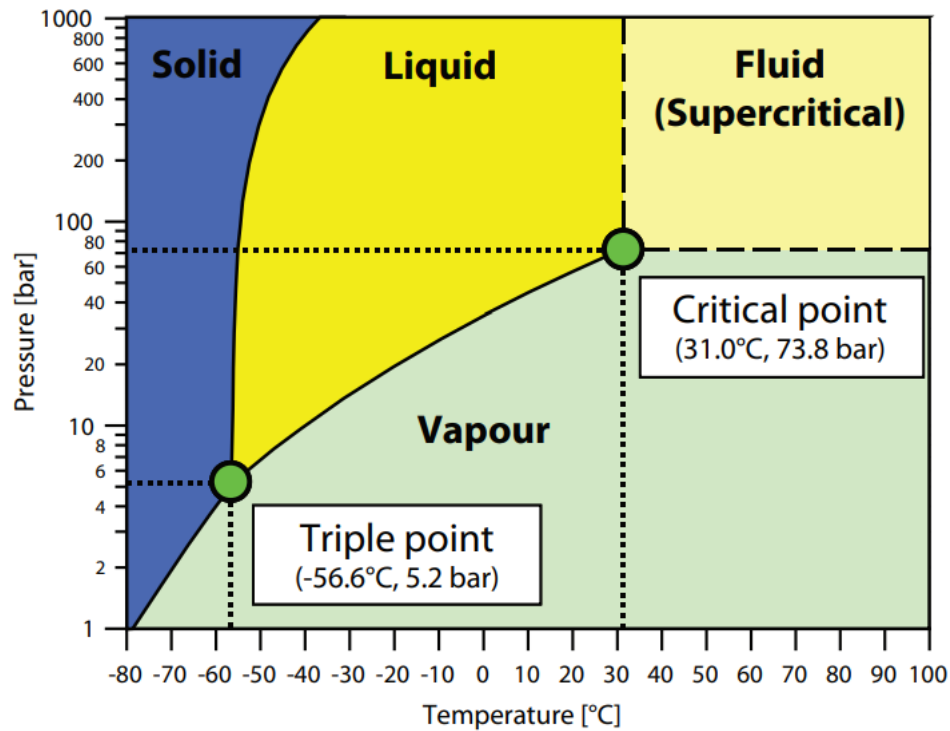


Figure 8: Phase diagram of R744 (CO₂) [14]

The CO₂ have a high thermal expansion coefficient. This means that the fluid has a great facility to expand with the absorption of heat, and, in case of being trapped, this volume increase will increase the pressure exponentially [14, 15].

Because of its low critical temperature, it is practically impossible to reject heat by condensing the carbon dioxide, therefore it is needed to use a gas cooler instead of a condenser, as shown in Figure 9 [14].

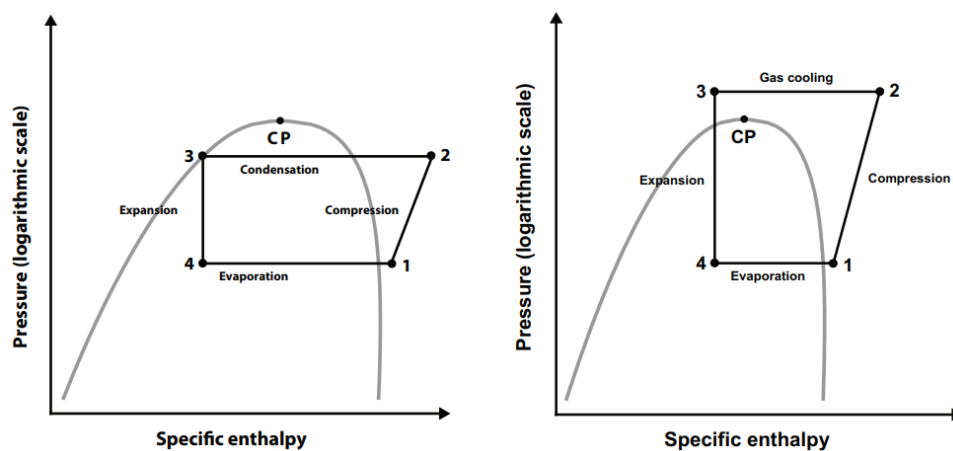


Figure 9: Subcritical (left) and transcritical (right) refrigeration cycle process [14]

2.2.1 Compressors

A compressor has two main functions within the refrigeration cycle, remove the required amount of refrigerant vapor from the evaporator so that the desired temperature and pressure can be maintained in the evaporator and increase the working fluid pressure and temperature [13].

Therefore, refrigerant compressors can be classified by architecture as [13]:

- a) **Hermetic compressor:** this type of compressors, as shown in Figure 10, is used in units that require a small range of temperatures and are mostly used in small equipment where the cost is a major factor. The motor and drive are sealed in compact welded housing. The refrigerant and lubricating oil are no rotating seals separating motor and compressor. It cannot be open in any circumstances. Hermetic compressors can work for a long time in small-capacity refrigeration systems without any maintenance required [13].

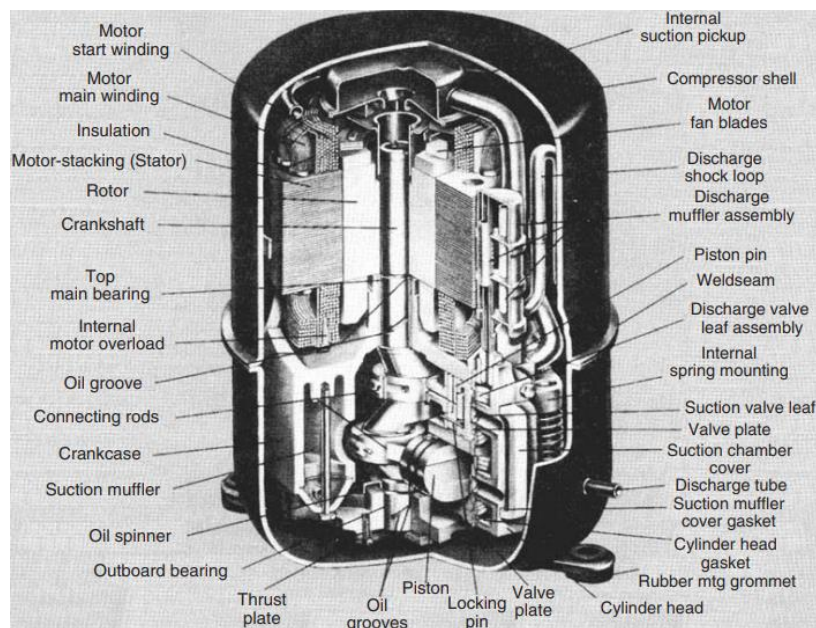


Figure 10: A typical hermetic reciprocating compressor [13]

- b) **Semi hermetic compressor:** In this type of compressor, represented in Figure 11, the refrigerant does not flow across the motor windings since bigger refrigeration compressors are frequently semi hermetic, even if the motor and compressor are housed in the same casing and can be unbolted. The necessity for external motor cooling, which increases efficiency in cooling applications, is not advantageous for refrigeration operations and is significantly more expensive than for hermetic

systems. These compressors were developed to avoid the disadvantages of the hermetic compressors [13].

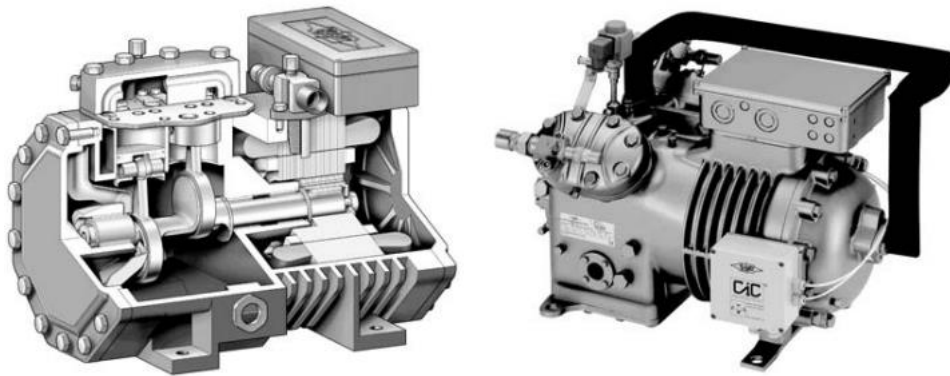


Figure 11: Semi hermetic reciprocating compressor [13]

- c) **Open compressor:** These compressors, shown in Figure 12, have crankshafts that extend through the compressor housings and are externally linked to electric motors. Where the shafts enter the compressor housings, appropriate seals must be utilized to stop the leakage of refrigerant gas or the entry of air (when the crankcase pressure is lower than atmospheric pressure). The motor and compressor are rarely enclosed in the same housing to prevent leakage at the seal [13].



Figure 12: Open type reciprocating compressor [13]

The compressors can also be divided by type:

- i. **Displacement compressors**, use the shaft work to increase the refrigerant pressure by reducing its volume in a closed chamber. The compressors of this group are reciprocating, rotary vane and screw compressors [13].

- ii. **Dynamic compressors**, that increase the refrigerant pressure through a continuous exchange of angular momentum between a rotating mechanical element and the fluid subject to compression [13].

According to the type of compressor used, the compressor can be reciprocating, rotary vane, scroll, screw in case of displacement compressors and centrifugal and turbo compressors in case of dynamic compressors, as shown in Figure 13 [13,16].

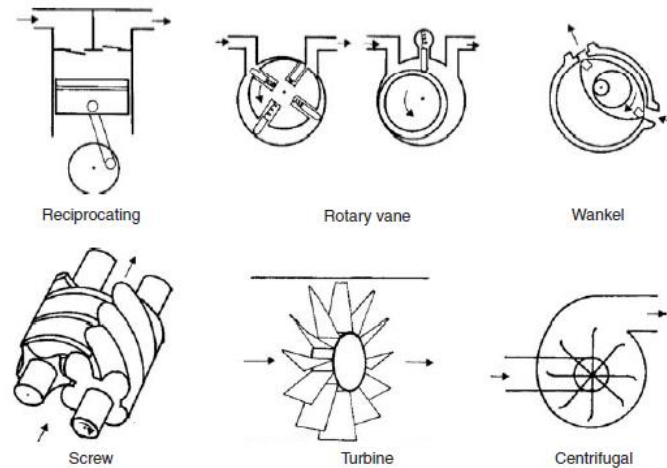


Figure 13: Compressor types (Heap, 1979) [13]

It is expected that the compressors satisfy the following requirements: high reliability, long service life, easy maintenance, easy capacity control, quiet operation, compactness, and cost effectiveness [13]. To select a refrigerant, the criteria considered are the refrigeration capacity, volumetric flow rate, availability, safety, compression ratio and thermal-physical properties of the refrigerant [13,16].

Briefly analysing the compressor types, starting by analysing the displacements compressors. The reciprocating compressors, compress the refrigerant on the for-ward stroke of a piston and are built to be single acting in a large capacity range, up to hundreds of kilowatts.

The advantages of a rotary vane compressor are the fact that has a simple and compact design, which allows easy access and maintenance. It is a single stage compression that results in a very high compression ratio on the order of 7:1, as well as better energy efficiency and reduced risk of fault, a low rotation speed that reduces the vibrations, low cycle temperatures that reduce the oil and energy consumption, and low needs of maintenance [13].

The screw compressors have higher reliability and improved performance, as reduced costs, fewer moving parts, less vibration, and less refrigerant loss [13,16].

The advantages of a scroll compressor are that makes less noise, and have a higher lifetime, achieve higher efficiency, and better part-load performance and operating characteristics [13,16].

Dynamic compressors, such as centrifugal compressor are used at a large-scale system, with large capacities or for high-flow or low-pressure difference applications [13].

Finally, turbo-compressor have a low efficiency, and in this type of compressor, the discharge pressure is limited by the maximum permitted tip speed, this type of compressor is used when high suction volumes at high suction pressures are required [13,16].

2.2.2 Gas cooler

The gas cooler should be designed for the application of CO₂ as refrigerant. Besides high pressures, it is important to address the following conditions:

- Gas cooler outlet temperature.
- Temperature change of the CO₂ during the cooling process.

The gas cooler outlet temperature should be as low as possible to obtain the highest capacity and COP, as shown in Figure 14.

Figure 14 illustrates the changes in state point locations and cycle performance as the gas cooler pressure is either increased or decreased. The state points with index L represent the cycle with a lower pressure, state points with the index H represent the cycle with a higher pressure. The numbers in the Figure 14 show the consequences of a change of ± 5 bar in the gas cooler pressure. By observing the specific enthalpy in the evaporator (Δh_{EVAP}), it is possible to notice that for the cycle with a lower pressure, the refrigerant capacity is only 64% of the nominal cycle process. For the cycle process with a high pressure, the capacity is 106% of that nominal cycle process [14].

The influence of gas cooler pressure on compressor power consumption can be observed by comparing the changes in specific enthalpy in the compressor (Δh_{COMP}). The pressure changes of ± 5 bars cause changes in the compressor power consumption of $\pm 10\%$ [14].

Together the changes in refrigerant capacity and compressor power consumption cause changes in the COP. As shown in Figure 8 for a nominal cycle COP is 2.6, but for both processes with higher and lower pressures, the values of COP are lower, indicating that an optimum gas cooler pressure exists [14].

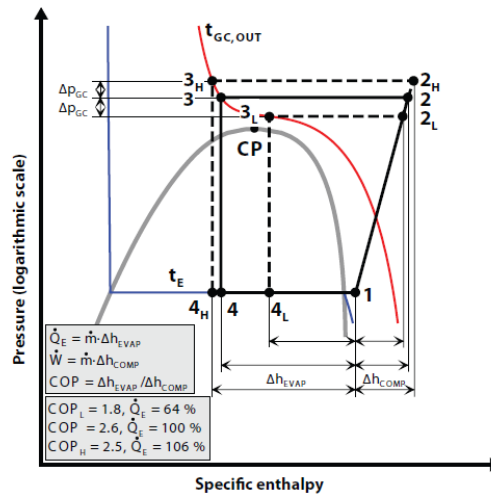


Figure 14: Influence of the gas cooler temperature and pressure on cycle process [17]

Figure 15 shows the variations in refrigerant capacity and COP for a system operating with constant evaporating temperature, superheat, and gas cooler outlet temperature. This model of the compressor reflects “true” compressor behaviour, meaning that isentropic and volumetric efficiencies vary with the operating conditions [14].

As shown, the optimum condition for refrigerant capacity is found for a higher gas cooler pressure than for the optimum condition for COP. For a system operating at conditions close to optimum COP, the extra refrigerant capacity that can be obtained by increasing the gas cooler pressure is relatively small [14].

In almost all practical applications the operating conditions (evaporating temperature, superheat, and ambient temperature) will vary. Consequently, the optimum gas cooler pressure also varies [14].

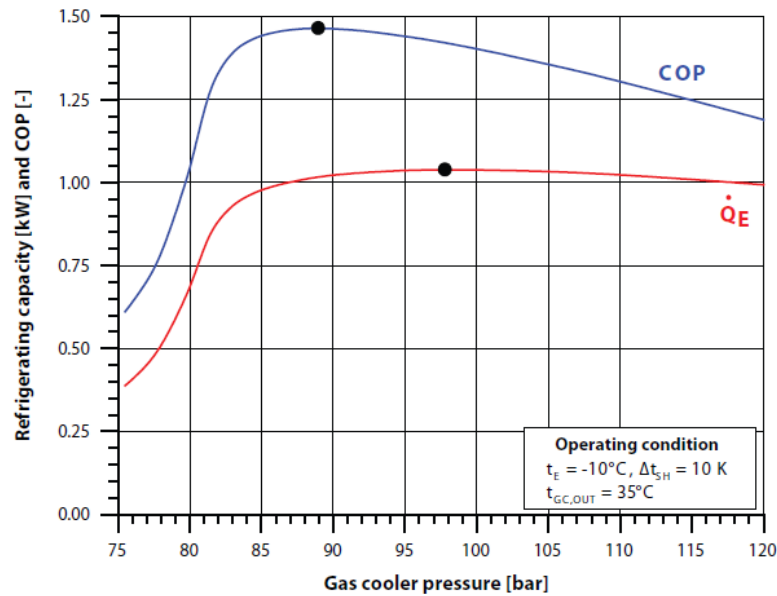


Figure 15: Effect of gas cooler pressure on refrigeration capacity and COP [14]

Unlike condensers where most of the heat exchange takes place at constant temperature, the temperature of the CO₂, steadily drops while the refrigerant flows through the gas cooler [14].

With a standard design, heat would flow from the warmer parts to the colder parts of the gas cooler. This would not only lower the ability to reject heat to the environment but also result in an increased temperature of the carbon dioxide leaving the gas cooler.

For optimum efficiency, the temperature of the CO₂ when leaving the gas cooler should not be more than 2-3K above the temperature of the ambient air before entering in the gas cooler [14].

To minimize this heat flow, the thermal contact between the gas cooler sections must be minimized as shown in Figure 16. In order to get the airflow as cold as possible to the gas cooler, the compressor should be in the air flow behind the gas cooler [14].

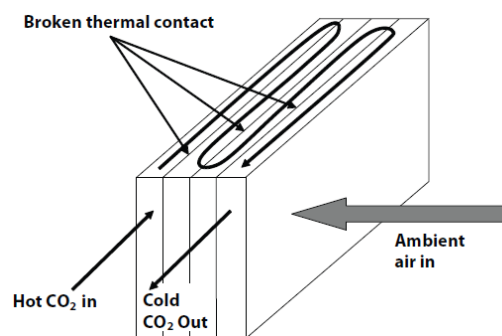


Figure 16: Principle of gas cooler to minimize the refrigerant exit temperature [14]

2.2.3 Expansion valves

The expansion valves are used to reduce the pressure from de condenser/gas cooler (high pressure) to the pressure of the evaporator (low pressure) and, to regulate the liquid refrigerant flow that enters the evaporator according to the equipment and load characteristics [13,17].

The most common types of expansion valves are:

- Thermostatic expansion valves.
- Constant-pressure expansion valves.
- Float valves.
- Capillary tubes.
- Electronic valves.

i. Thermostatic Expansion Valves (TEV)

These valves, represented in Figure 17, regulate how much refrigerant is released into the evaporator and are effectively reduction valves [13].

The capacity and temperature ranges are excellent since there is a wide variety of expansion valve sizes and bulb charges. TEVs drawback is the requirement for relatively strong superheating, which deprives the evaporation process of its heat transfer area [17].

By changing the mass flow of refrigerant in response to the evaporator load, the TEV aims to maintain a constant amount of superheating inside the evaporator under all circumstances. A membrane located within the valve housing compares the temperature before and after the evaporator to achieve this [17].

The TEV needs to be used in conjunction with a bulb to compare the pressures before and after the evaporator. Inside the valve head, a membrane balances the pressure differential between the saturation pressure of evaporation and the pressure in the bulb. The position of a needle and, consequently, the amount of refrigerant entering the evaporator are controlled by the movement of the membrane [17].

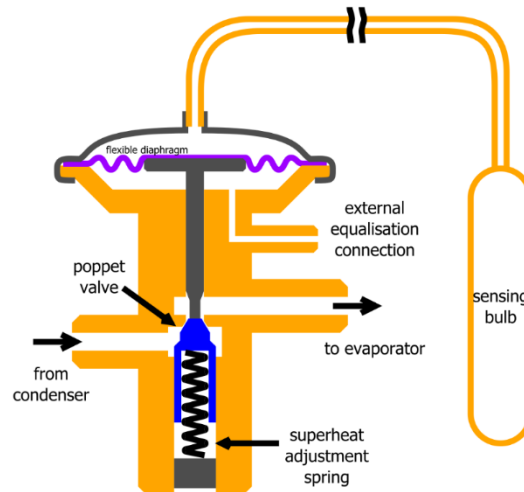


Figure 17: Parts of a thermal expansion valve [17]

ii. Constant-Pressure Expansion Valves

Constant-pressure expansion valves are frequently referred to as automatic expansion valves because they automatically open and close and maintain a constant pressure at the outlet without the intervention of any external mechanical component. These valves keep the exit pressure constant. Based on the suction pressure, they can detect and hold the evaporated. It has a precise match between the refrigerant flow and compressor capacity is made [13].

iii. Float Valves

Both high-side and low-side float valves are available. They are used to regulate the refrigerant flow to a liquid cooler of the flooded type [13].

The low-side float valve assists in keeping the evaporator's liquid refrigerant level consistent. It opens when the evaporator is empty and closes when the evaporator is full of liquid [17].

The high-side float valve controls the system's high-pressure side and keeps the condenser's refrigerant level constant. Condensed refrigerant flows to the float chamber and opens the valve when the compressor is running [17].

The refrigerant flows into the evaporator as a result, where it is stored. The valve opening will close when the liquid level drops in the float chamber, blocking the passage of liquid to the evaporator [17].

iv. Capillary Tubes

A long, wound-up copper tube with a tiny hole known as a capillary tube is used to collect hot, high-pressure liquid refrigerant from the condenser. High pressure is maintained on one side of the tube by this little aperture while low pressure is maintained on the other. The pressure of the refrigerant flowing through the tube is rapidly reduced by friction from its walls [17].

These components, which are frequently used in small hermetic-type refrigerant systems with a capacity of 30 KW, lower the condensing pressure to the evaporating pressure while maintaining a consistent evaporating pressure regardless of changes in the refrigeration load [13].

When using capillary tubes, the tube length is modified to correspond to the compressor capacity while also taking the condenser efficiency and evaporator size into account [13]. A capillary tube is shown in Figure 18.

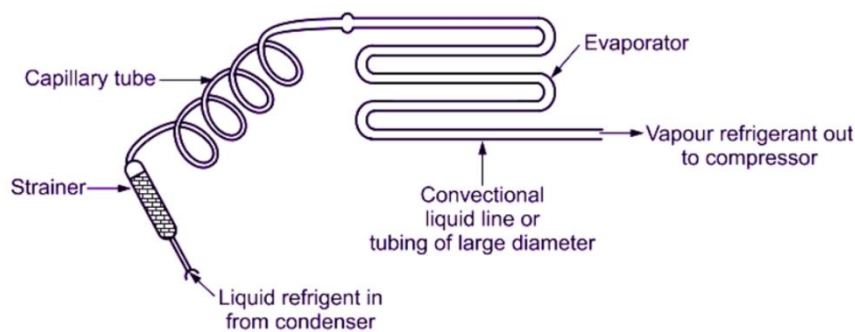


Figure 18: Capillary tube [17]

v. Electronic Expansion Valves (EEV)

The electronic expansion valve (EEV) functions using a far more advanced design. These expansion valve varieties regulate the refrigerant flow that enters a direct expansion evaporator. They carry out this action in response to signals that an electronic controller sends to them. The valve port is opened and closed by a tiny motor known as a step or stepper motor. Step motors do not spin continuously; instead, they are signalled by an electronic controller and perform a partial rotation for each signal received. A gear train that puts a pin in a port where refrigerant flows power the step motor [17].

Figure 19 represents an electronic expansion valve.

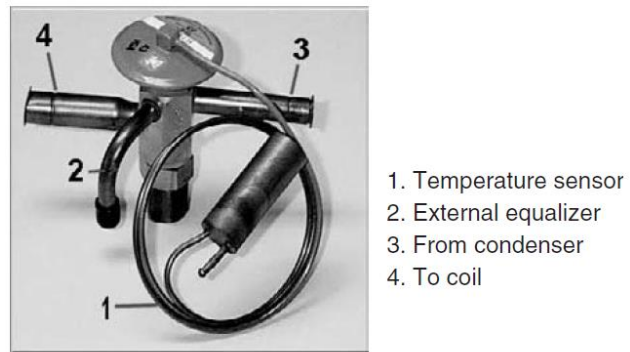


Figure 19: Electronic Expansion Valve [13]

2.2.4 Evaporator

In the evaporator the refrigerant is at a lower temperature than its surroundings. Therefore, it evaporates and absorbs latent heat of vaporization, producing a low-pressure saturated vapor when entering in the compressor without any change in the temperature [13].

Evaporators are divided into two types:

- Direct cooler evaporator, use of outside air that is carried in through ventilation ducting and humidified in a special air handling device. Before being distributed within, this airstream with a higher moisture content (relative humidity) reaches the ideal temperature and has the ideal level of hydration. This conditioned air is essential because it replaces the exhausted room air and helps to control the temperature inside the building. The evaporative cooling effect is larger if the air collects more moisture upon entry due to having a lower initial relative humidity [18].
- Indirect cooler evaporator, outside air is brought inside, through ventilation, to reduce temperatures without combining the two air streams. While the return air from the room passes through the Heat Recovery unit, fresh outside air is passed through the evaporative humidifier and exhaust system without being mixed. The return air may be cooled before being reintroduced in the room if the air temperature after the evaporative humidifier is lower than the room temperature. Because moisture isn't introduced to the supply air and the outside air that is being humidified is vented, this approach has the advantage of successfully cooling the airstream without over-humidifying it. Even in seasonal regions or places where the outside temperature is higher than the inside, indirect evaporative cooling techniques are helpful [18].

i. Air coolers

These coolers, are shown in Figure 20, consist of a network of tubes through which refrigerant travels and are sometimes referred to as direct expansion coils. The exit gas header contains an accumulation of the hot refrigerant vapor. Due to their relatively low-pressure ratios, these direct expansion coils are exclusively employed in positive displacement (inlet) compressor systems [13].

In a flooded coil, the evaporator coil is kept close to full of the liquid refrigerant. A significant amount of heat is transferred by the entire contact of the liquid with the tube walls because they need a lot of refrigerant. A dry coil uses less refrigerant than a wet coil, which lowers the refrigerant charge [13].

A dry coil requires only a small amount of refrigerant, and this reduces the cost of the refrigerant charge. After absorbing heat from the medium to be cooled, the dry expansion coil produces primarily liquid at the input and only superheated vapor at the output [13].

Frosting happens in the air coolers when the surface temperatures drop below 0°C. Thick frost coatings restrict the airflow rate and the amount of interior space accessible while serving as insulation [13].



Figure 20: Room type and Large-scale industrial type air coolers [13]

ii. Liquid Coolers

Both flooded evaporators and direct expansion evaporators, which are commonly shell and tube type heat exchangers, can be used in chilled water systems. In a flooded evaporator, a level valve regulates the amount of refrigerant that floods the heat exchanger's shell side. Through the tubes, cold water is circulated. In contrast, water is transported in the shell of a direct expansion evaporator while the refrigerant is boiled inside the tubes [13].

To ensure that only refrigerant gas leaves the evaporator, the rate of refrigerant flow is throttled. It is important to note that if the refrigerant vaporizes on the outside surface of the tubes the evaporator is a flooded cooler, if it vaporizes inside the tubes the evaporator is a dry cooler [13].

Figure 21 represents a liquid cooler, shell type evaporator.

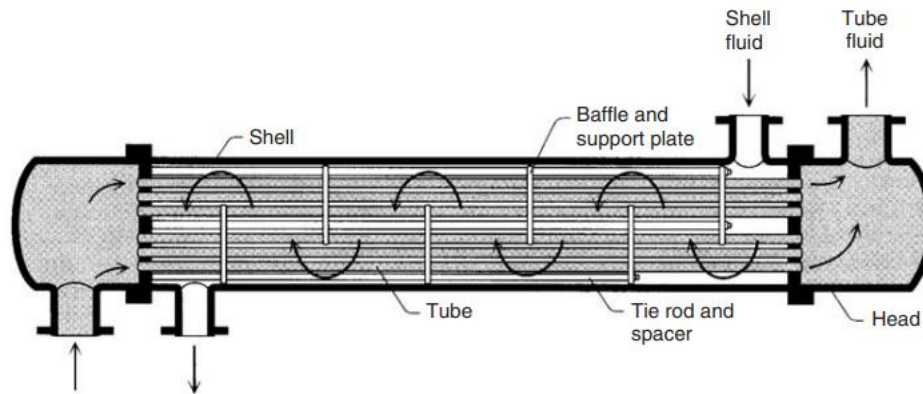


Figure 21: Shell-tube type evaporator [13]

2.4 Heat released by refrigeration cycles through the environment

Industrial waste heat refers to energy that is generated in industrial processes without being out to practice use [7]. Sources of waste heat include for example hot combustion gases discharged to the atmosphere, heated products exiting industrial processes, and heat transfer from hot equipment surfaces. Various studies have estimated that as much as 20 to 30% of industrial energy consumption is discharged as waste heat [7]. While some heat losses from industrial processes are inevitable, losses can be reduced by improving equipment efficiency or waste heat recovery technologies. Waste heat recovery consists in capturing and reusing the wasted heat for heating or generating mechanical or electrical power [7].

The term "total waste heat" typically refers to the total quantity of heat generated after the thermal carrier has reached ambient temperature. Technical potential of waste heat is the term used to describe the portion of this heat that can be recovered. [6].

It is feasible to estimate the entire amount of waste heat that is theoretically recoverable in Europe using waste heat recovery fractions broken down by temperature and industrial sector [6].

As shown in Figure 22, there is just a small amount of waste heat potential exceeding 1000 °C, and it is only seen in the iron and steel sectors. There isn't much waste heat accessible, even in the 500 – 1000 °C temperature range, with the potential being constrained in the cement, iron, and steel industries. The potential grows between 200 and 500 °C, primarily in the pulp and paper industries. The bulk of waste heat is generated between 100 and 200 °C, across most industrial sectors, whereas below 100 °C, it is

largely generated by drying and preheating processes in the food and beverage industry [6].

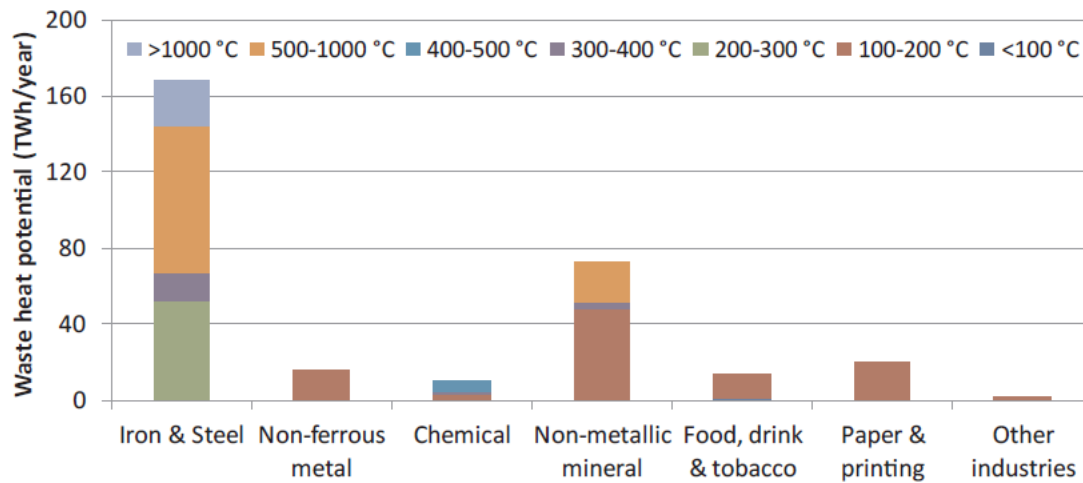


Figure 22: Waste Heat potential per industrial sector and temperature level for EU industry in 2015 [6]

The waste heat potential at temperature level of 100 – 200 °C is high and represents one third of the total one. It is available in a large variety of industries. The waste heat of most sectors is within the that temperature level, which matches the temperature of the exhaust gases of boilers and standard heating equipment [6].

The KoolSys system is mostly applied in large and medium-sized commercial premises and industrial complexes, in order to refrigerate space and food. Therefore, it is important to understand the waste heat potential in that industry.

Figure 23 shows the waste heat potential in food industry in each European Union country.

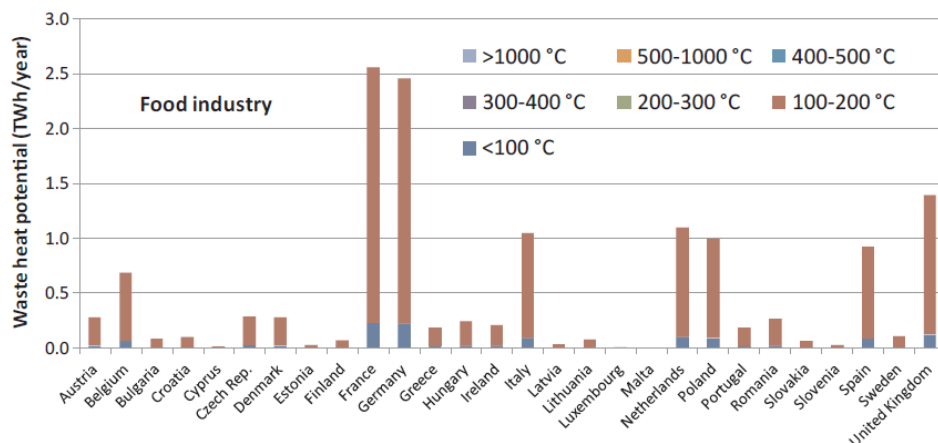


Figure 23: Waste heat potential in each EU country per temperature level in food industry [6]

Heat Waste can be used for many processes, like [7]:

- Preheating boiler inlet water.
- Preheating combustion chamber inlet air.
- Electricity production.
- Heating ventilation and air cooling (HVAC)

2.3.1. Waste heat recovery

There are three essential components required for a waste heat recovery system [7]:

- An accessible waste heat recovery.
- Suitable recovery technology.
- Point of end use for the recovered energy.

Waste heat recovery systems are frequently constrained by factors such as temperature limits and investment return of recovery equipment's. Table 1 presents the heat source range for waste heat recovery range values.

Table 1: Temperature range of waste heat recovery and their application [7]

Heat Source Temperature Range	Waste heat recovery application
High [> 650 °C]	• Combustion Air Preheat • Steam generation for process heating • Steam generation for mechanical/electrical work • Furnace Load preheating • Transfer med-low temperature process
Medium [230 – 650 °C]	• Combustion air preheat • Steam/Power generation • Organic Rankine cycle for power generation • Feedwater/Furnace load preheating • Transfer to low-temperature processes
Low [< 230 °C]	• Space Heating • Domestic hot water • Upgrading via a heat pump to increase temp for end use

The waste heat recovery it is a process that use various technologies for recovery and reuse of the heat rejected by the process. It can be recovery from several sources [19]:

- Exhaust gases.
- Cooling water.
- Solar panels.
- Overheating heat.
- Heat rejected from condensation process in a refrigeration cycle.

The most common technologies for waste heat recovery are [19]:

- Heat exchangers to make direct use of the heat in the same state as it is.
- Heat pumps and vapor recompression, that transform heat to generate more useful work than if it were at its initial temperature.
- Multi-stage operation like multi-effect evaporators or vapor expansion.

2.5 Industrial refrigeration

Most of the industrial refrigeration is made through compression refrigeration systems.

2.4.1. Use of petrol engines

It is feasible to employ a gasoline engine to convey mechanical energy to the compressor of the refrigeration system rather than an electric motor. In the food business, chemical industry, and ice production, this kind of technology has been tested. An absorption chiller, which enables savings of 77% in the consumption of initial primary energy, can be used in a refrigeration system powered by a gasoline engine. Savings without the absorption chiller are around 52%. [19].

Even though this type of engine allows great savings in energetic terms, these measures need considerable investments in equipment's and have an increase in emissions of NO_x. The payback period with absorption chiller or without are 4 years and 2 years, respectively [19].

2.4.2. Thermal accumulation

Thermal accumulation it is an old technology that allow the production of ice when the electricity is cheaper and the storage of the ice until this could be used to refrigerate. It is a simple technology.

2.4.3. Energy accumulation

To be able to accumulate and use some of the heat wasted for DWH, is needed that the accumulator has a latent material, usually a phase changing material like paraffin.

The paraffin has the advantage that when reach 65°C and change the phase from solid to liquid, absorbs over five times more heat than water, stopping it from evaporate and preventing condensation inside the accumulator.

2.4.4. New working fluids in refrigeration systems

Due to Montreal protocol, the research efforts are focused on development of alternative refrigerants to CFCs and HCFCs. Most of these refrigerants can save energy between 2 and 20%, according to studies [19]. One solution is the use of natural refrigerants like CO₂.

Natural refrigerants are chemical compounds that are produced in nature's biochemical processes. They do not destroy the ozone layer and make an insignificant contribution or even zero for global warming. The use of carbon dioxide has the advantage of being environmentally friendly with reduced greenhouse effect, toxicity, flammability, or impact on the ozone layer [14]. Therefore, CO₂ is a natural refrigerant fluid, and it is non-toxic and non-inflammable [14]. It is characterized for a low global warming potential (GWP=1) and has excellent thermodynamic properties [14, 15].

3 KoolSys system model and waste heat recovery system

3.1 Data Assembly

This dissertation is based on a KoolSys system that use R744 (CO₂) as a refrigerant fluid and is intended for refrigeration plants with application in large and medium-sized commercial premises and industrial complexes. In this work, it was studied the energy released from the gas cooler of the Koolsys system through the environment, the demand for HVAC and/or DHW of the supermarket and if it was needed the use of an accumulator for energy storage. To determine that possibility, RACE has enabled data from the KoolSys plant, since they receive data such as pressure, temperatures, and mass flow from the system.

In this work, the KoolSys plant studied is located at a supermarket in the centre of Portugal and According to Climate Data, Table 2 shows the average, maximum and minimum outside temperature during the year of 2022, in the county in question.

Table 2: Climate data of a county in the centre of Portugal [20]

	Janeiro	Fevereiro	Março	Abril	Maió	Junho	Julho	Agosto	Setembro	Outubro	Novem- bro	Dezembro
Temperatura média (°C)	8.9	9.8	12.2	14.3	17.6	21.5	23.2	23.7	21.5	17.3	12.1	9.7
Temperatura mínima (°C)	5.1	5.6	7.5	9.3	12	15.1	16.4	17.1	15.6	12.9	8.4	6.2
Temperatura máxima (°C)	13.5	14.8	17.5	19.6	23.5	28.4	30.8	31.4	28.2	22.5	16.5	14.1
Chuva (mm)	80	61	68	65	60	17	6	10	40	100	88	89
Umidade(%)	81%	74%	68%	68%	61%	55%	52%	51%	56%	70%	76%	79%
Dias chuvosos (d)	7	5	6	7	6	3	1	2	4	7	7	7
Horas de sol (h)	5.3	6.5	7.0	8.2	9.7	10.9	11.4	10.9	9.3	7.2	5.8	5.5

3.2 Characterization of KoolSys plant considered in this case study.

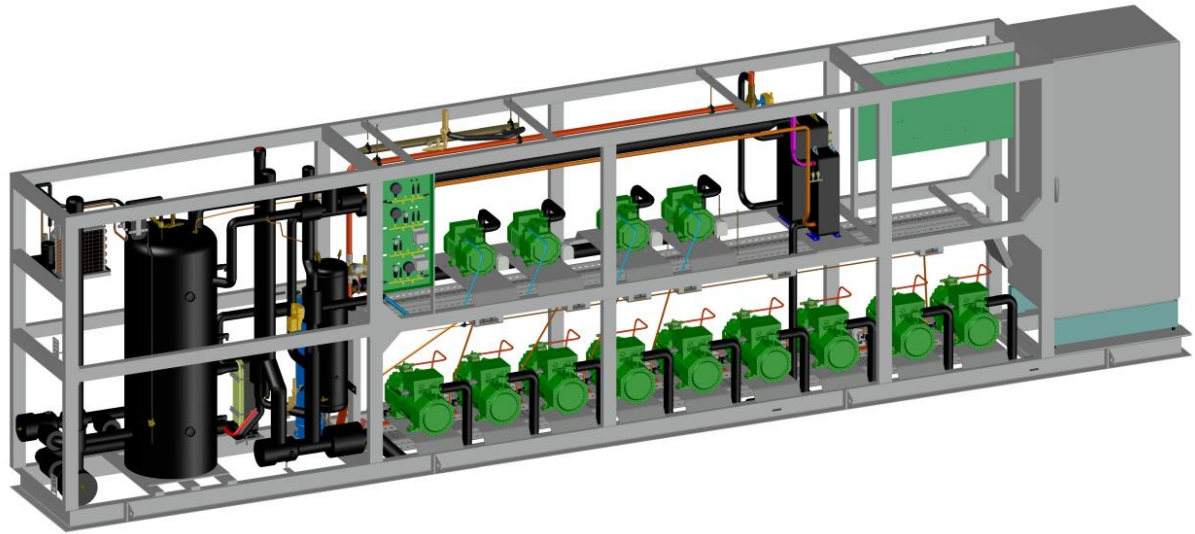


Figure 24: KoolSys system and componentes

The KoolSys plant, as shown in Figure 24, is a compression chiller plant and has R744 (CO_2) as working fluid. This plant works in booster system and is intended for use in refrigeration plants in large and medium-sized commercial premises and industrial complexes.

This plant is set up with two stages of compression. The first stage of compression (low temperature compressors) is destined for compressing gas from equipment with an evaporation temperature in the order of -30°C and the second stage of compression (medium temperature compressors) is responsible for the compression of the mixture of three different origins: low temperature compressor discharge, the gas that comes from the medium temperature (equipment's with an evaporation temperature of -10°C) and the gas from the liquid deposit, also called flash gas. Each of these compression stages has semi-hermetic compressor, and in each stage, at least one compressor is equipped with a frequency variator.

The discharge of the medium temperature compressors, after passing for an oil separator goes to the gas cooler. This zone is called the high-pressure zone and it is regulated for an electronic expansion valve, also called transcritical valve. The expanded fluid is below the critical point, and it will be directed for the liquid deposit.

The liquid deposit has a stable pressure of about 38 bar and is regulated for other electronic expansion valve, also called flash gas valve. This deposit allows a separation between liquid and gas. The liquid it will be used by the installation equipment's and the gas goes to the flash gas valve or for the auxiliary compressor, also called parallel compressor.

This equipment has as goal to increase the energetic efficiency of the installation. For small gas flow rates, the flash gas valve will expand the fluid which will then be mixed with the medium temperature suction. Whenever the flow rate increases above a set value, the gas-flash valve will close and the gas in the reservoir will be compressed by the auxiliary/parallel compressor. A frequency inverter can also be applied to this compressor.

Four regions with different pressure levels are thus established:

- **High pressure line:** from the discharge of the medium temperature to the transcritical valve. This zone must support a pressure of 120 bar.
- **Liquid Line:** corresponds to the area from the liquid receiver to the suction of the auxiliary compressor and gas-flash valve. It should withstand a pressure of 60 bar.
- **Medium temperature suction line:** Is the line where the suction of the medium temperature compressors, the discharge of the low temperature compressors and the gas-flash valve and shall be dimensioned to withstand a pressure of 52 bar.
- **Low temperature suction line:** Is the line where the suction of low temperature compressors and is dimensioned to support a pressure of 30 bar.

All the regions mentioned above are protected by a security valve, who ensure the pressure relief in case of maximum pressure. This valve will protect the system independently from the required refrigerant power. This plant has two security valves placed in parallel, and each one of those will ensure the total flow rate necessary.

The plant will be connected to a plate heat exchanger, designed to exchange heat between the liquid receiver and a self-contained cooling system, called an emergency unit. This unit, electrically connected to a UPS or generator in the plant, is designed to delay the tripping of the liquid receiver safety valve in case of power failure.

In parallel with the R744 circuits, this assembly will have an oil circuit that ensures the separation and distribution of oil to the compressors, essential for their good operation.

This consists of a separator, an oil reservoir, an oil filter, an oil control solenoid valve and individual oil level regulators for each compressor.

Other equipment includes solenoid valves for the injection of gas and liquid in the medium temperature suction, filters in the medium and low temperature liquid and suction lines, and two heat exchangers for superheating the low temperature suction and the parallel compressor. These are located between the low temperature suction line and the liquid line, and the other between the suction line of the parallel compressor and the fluid return line from the gas cooler.

The plant can have copper or stainless steel piping. The joining of the piping will be done mostly by permanent connections, namely brazing or welding, and a small part by screwed and crimped connections.

This project was certified as an assemblage.

3.3 Methodology

The goal of this project was to determine the possibility of a recovery wasted heat system. To start this project, it was needed to know the available heat that was wasted for the gas cooler of the KoolSys plant. RACE collects data from a KoolSys plant, in this case located in the centre of Portugal, such as pressure in the gas cooler, inlet and outlet temperature in the gas cooler, among many other data, not relevant in this dissertation.

Pressure and temperatures are measured with sensors of pressure and temperature respectively. The calculation of the flow rate that goes through the compressors is explained in Annex A.

In this dissertation it was made an analysis from July/2022 through January/2023 from the collected data, to determine the available energy exiting the gas cooler through the environment.

This method will be applied to all the months in study, and since the data is collected from thirty seconds range, it was possible to verify the energy availability by hour, day, and month.

The next step was to compare the available energy in an average day of each one of these months, by hour, with the requirements of heating loads of the space and the DHW demand. This would make us understand if the demand of heating loads and DHW could be satisfied and if it was needed an energy storage deposit.

In this case, as it is shown in Table B 8, Annex B, the energy available from the system is at all hours, and in all the months in study, higher than the demand of the heating loads. Therefore, it is not necessary to use an energy storage unit, since we have at all the time, energy available that we can use for HVAC.

Knowing this possibility, it was necessary to create an installation that allow us to use the available energy, which instead of being dissipated into the environment, would be used as electrical energy for the AQS and/or HVAC system.

The final step, it was the design of a wasted heat recovery system and compare the costs of installation with the costs if there was no recovery system and therefore all electrical power were consumed by the grid.

4 Results and discussion

In this chapter, the results obtained will be shown and discussed to understand the availability of a waste heat recovery system. In the first point, the results will be presented based on what has been discussed in the previous chapters. On point 4.2, the waste heat recovery system will be explored in more depth, which will include a comparative financial analysis of the use of the system.

4.1 Results

As explained in chapter 3, the process started with the evaluation of the data measured from the KoolSys plant. In this sense, and in order to determine the available heat rate from the KoolSys it was used the equation (4.1), as shown.

$$\dot{Q} = \Delta h \times \dot{m} \quad (4.1)$$

With,

$\dot{Q}$ – heat rate released by the gas cooler [kW]

Δh – Enthalpy variation between inlet and outlet of the gas cooler [kJ/kg]

$\dot{m}$ – mas flow rate [kg/s]

Enthalpies were discovered using an excel add-in function called, Coolprop, with function of pressure and temperature in the gas cooler.

Since it was a very large amount of data, and it is only available from July to January, with some missing data, it was chosen to evaluate the available hourly energy and hourly heating demand, for an average day of each month. The average day of each month was determined based on an average of 7 days of each month, being each of those a different day of the week, from Sunday through Saturday.

An example of the data obtained is shown in Table B 1 and the results obtained are represented in Table B 2, in Annex B.

With these results the hourly average demand was calculated for an average day of each month using the equation (4.2):

$$E = P \times \Delta t \times 0.000287 \quad (4.2)$$

With,

E – Energy [kWh]

P – Power [kW]

Δt – Time step [s]

The energy behaviour on those average days is shown in Figure 25.

Figure 26 represents that the available power is higher during the summer and lower during the winter. This goes as expected, due to the average environment temperature at those periods, since during the summer the outside temperature is much higher than during the winter. Therefore, July and August are the months with the most released energy, contrary to December and January which are the months with less released energy.

Additionally, in Figure 26 it is also represented that, during working hours of the facility, between 8 am and 10 pm, exists much more energy released by the KoolSys system when compared with the non-working hours, 10 pm to 8 am.

The data from Figure 26 are detailed in Table B 3, Annex B.

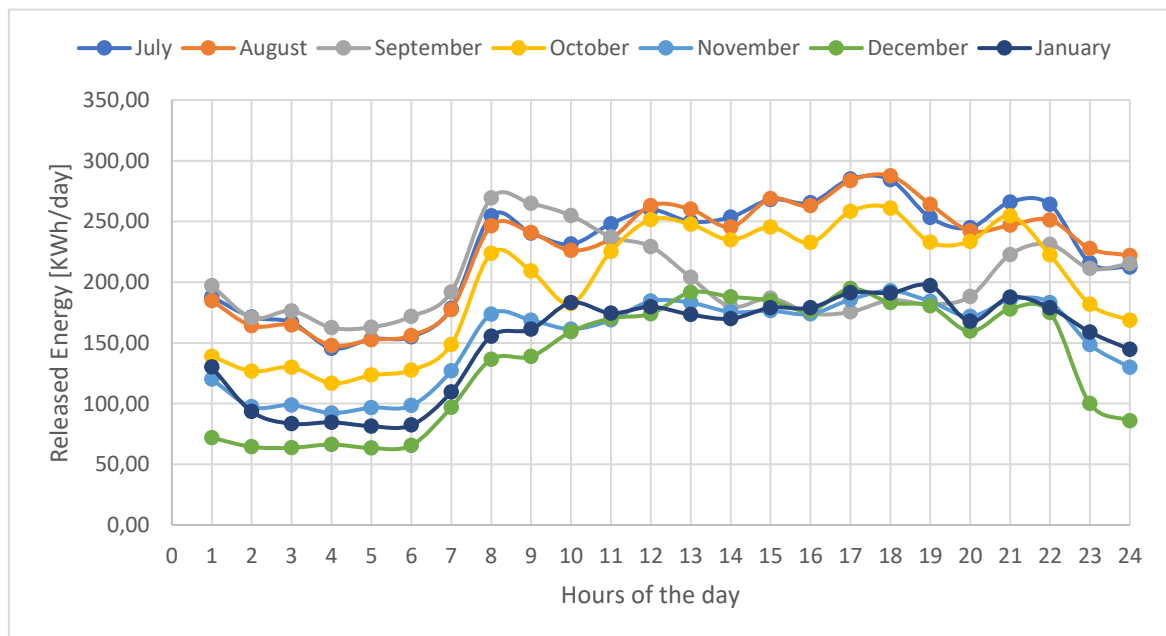


Figure 25: Average daily thermal energy rejected from July to January [kWh/day]

The required heating loads for each hour of each day are shown in Figure 26 and are detailed in Table B 5, Annex B. It is important to notice that the space heating demand could not be calculated because of lack of data, and therefore it was done a simulation using the same space and temperatures of the KoolSys plant location to determine those demand, hourly, daily, and monthly.

Figure 26 demonstrates the energy requirements for heating space for an average day of each month.

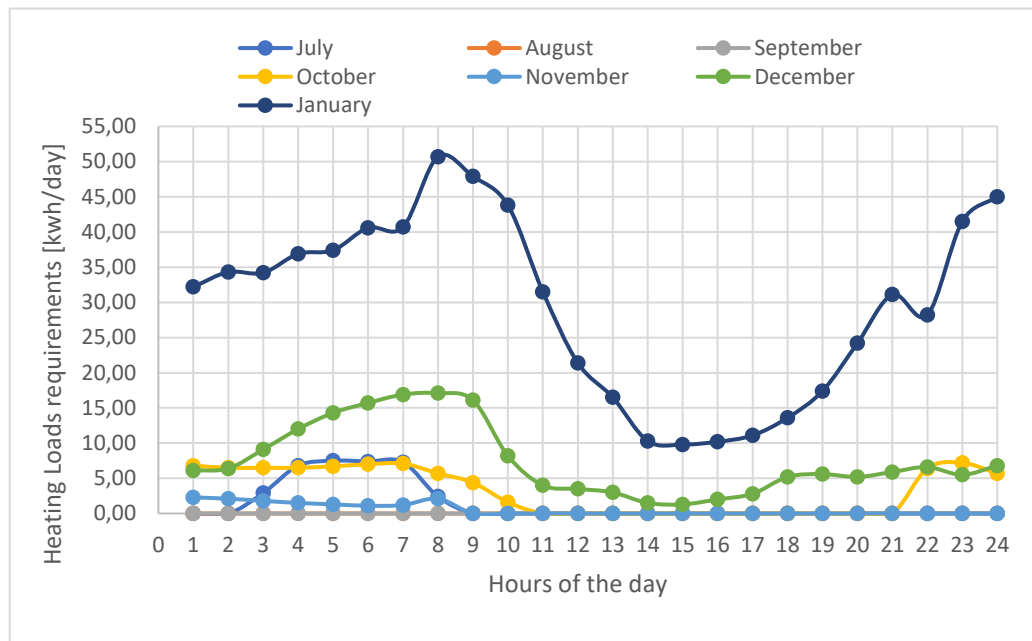


Figure 26: Energy requirements by hour for heating space in kWh/day

When compared with Figure 25, Figure 26 shows that there is no hour in any month in study that will have more energy needed than the one available. Therefore, we can conclude that there is no need to use an energy storage unit. This comparison is detailed in Table B 8, Annex B.

4.2 Waste heat recovery system

To develop a waste heat recovery system, it was needed to create a system that could recover the wasted heat by the gas cooler. It was determined that the following items should be included in the connection system, between the KoolSys plant, the DHW deposit and the HVAC system:

- Circulator pump.
- Heat exchanger 50 KW.

- Coil deposit (water heater) for the DHW system.
- Rooftop (Heating battery) for the HVAC system.
- Valves and accessories.

Figure 27 represents the installation needed for this waste heat recovery system.

This system is composed by a circulator pump, in the cold-water line, to return the cold water to the heat exchanger in order to warm up with the heat released by the gas cooler of the KoolSys system and be re-used for DHW and/or HVAC.

The DHW deposit needs to be a coil deposit, in order to be able to use the heat coming from the heat exchanger to warm up the water inside the deposit. The same will happen with the roof-top (heating battery), it receives hot water coming from the heat exchanger and use that heat to warm up the space.

The valves are of extremely importance in this system, the motorised valve on/off will ensure that the water leaving the heat exchanger will only pass through, for the deposit or for the roof-top, if there is a need for heating one of them, or both. The sectioning valves allow the water to pass by them, and when fully opened, decrease the pressure drop. The backstop valve prevents the water from coming back when it has passed by the pump, if that valve did not exist the system would collapse.

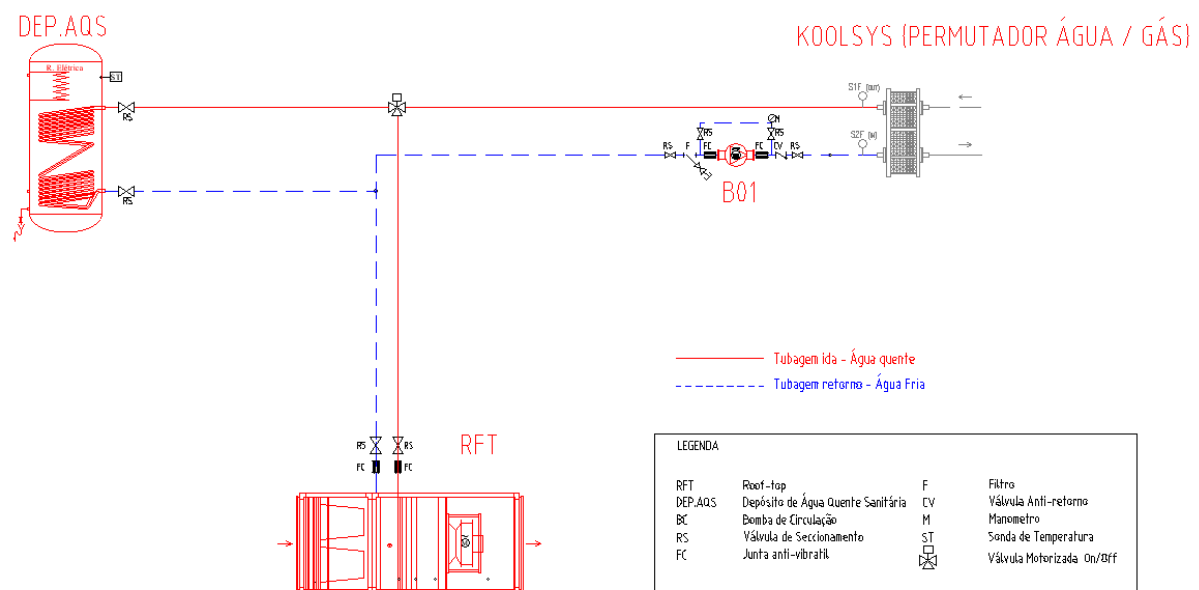


Figure 27: Waste heat recovery system [AutoCad]

4.2.1. Financial analysis of the system without heat recovery

Since data were measured only from 7 months of the year, from July to January, only these months were considered to perform this financial analysis.

First, it was necessary to determine the costs associated to the heating demand for HVAC and DHW, without the recovery system. Assuming the electricity from the grid cost is 0.15€/kWh it is possible to determine those costs.

The monthly heating demand for HVAC are shown in Table B 7 and the demand for DHW are shown in Table B 10, in Annex B.

- **HVAC electricity cost**

From Table B 7, was calculated the HVAC demand from July to January. Thus, to calculate those demand costs, it was used equation (4.3):

$$HVAC \text{ needs costs } [€] = \frac{0.15 \times \sum_{July}^{Jan} \text{Central Heating Coil Load}}{SCOP} \quad (4.3)$$

The HVAC demand for the seven months in study are 30917 kWh, with an electricity price of 0.15€/kWh and a SCOP of 3.52, as shown in Table B 11.

Therefore, HVAC demand costs are 1317.49€ for those 7 months.

- **DHW electricity cost**

From Table B 10, is possible to retrieve the data from the demand from July through January. Thus, to calculate those demand it was used equation (4.4):

$$DHW \text{ needs costs } [€] = 0.15 \times \sum_{July}^{Jan} \text{Carga} \quad (4.4)$$

The DHW demand for the seven months in study are 4528 kWh that with an electricity price of 0.15€/kWh gives a DHW demand costs of 679.2€ for those 7 months.

- **Total Cost**

The total cost is the sum of the DHW and the HVAC demand costs, what gives a total cost of 1996.69€ for the 7 months, with an average of 285.241€/month, without the recovery system.

4.2.2. Financial analysis with recovery system

The components of this system, and respective prices and quantities are explained in Table 2. The deposit with a coil it will be used to substitute the original electric water heater 300L, that is why the cost is only 270€.

As is possible to retrieve from Table 3, the total investment needed to implement this system is 16023.40€.

Table 3: Quantities and prices of the components needed to the wasted heat recovery system.

REF.	DESIGNATION	QUANTITY	UNIT PRICE	TOTAL PRICE
1	Additional recovery battery for KoolSys plant	1	5 040.80 €	5 040.80 €
2	Additional battery for hot water in Roof-Top (HVAC unit)	1	3 564.00 €	3 564.00 €
3	Additional coil inside the deposit for DHW	1	270.00 €	270.00 €
4	Circulator pump and accessories	1	1 856.25 €	1 856.25 €
5	Isolated pipe between KoolSys, DHW deposit and Roof-Top	1	2 256.20 €	2 256.20 €
6	Electric installation associated	1	1 011.15 €	1 011.15 €
7	Comand	1	2 025.00 €	2 025.00 €
	Total			16 023.40 €

In order to calculate the period of return on investment, is necessary to divide the total cost of the components by the monthly cost of the system without the recovery system.

$$\frac{\text{Total investment cost in the recovery system}}{\text{Monthly cost from electricity from the grid}} = \text{Return on investment time} \quad (4.5)$$

Using equation (4.5), it is possible to determine the return-on-investment time, the total investment cost in the recovery system is 16023.40€ as shown before, and the monthly cost from electricity from the grid is 285.241€/month, as calculated before.

Thus, the return-on-investment is equal to 4.68 which gives 4 years and approximately 8 months, to recover the amount invested in the system.

5 Conclusions

Nowadays, one of the major concerns in the world, mostly in Europe, is the energy efficiency of equipment and processes, especially in HVAC and refrigeration systems. RACE is a Refrigeration and air conditioning engineering company, which is not indifferent to these concerns. In this sense, for this project, the department of HVAC proposed a case study based on the KoolSys system, which the main purpose was to make the brand more competitive on the refrigeration and HVAC market.

Throughout this investigation a literature review was performed in order to understand the background of the transcritical refrigeration cycle with CO₂ and the possibility of waste heat recovery systems applied to them. In the first stage, it was performed a study about refrigeration cycles, and more specifically of transcritical cycles with CO₂ as the working fluid. Moreover, it was performed a study about the CO₂ characteristics and the main components of the cycle. Finally, it was analysed the possibilities of waste heat recovery systems, how could they work and be implemented with different capacities.

KoolSys is a transcritical CO₂ refrigeration cycle, and due to that working fluid, and is low critical temperature, it is practically impossible to reject heat by condensing the carbon dioxide, therefore it is needed to use a gas cooler instead of a condenser, the other components are the same as a vapor compression cycle.

The objective of this project was to understand the availability of using a waste heat recovery system, that could use the released heat from the gas cooler of the KoolSys system, to DHW and/or HVAC. In order to do this, was duly analysed the data collected such as temperature, pressure and mass flow rate, to determine the available power that went through the environment as from gas cooler. Those values were compared with the demand for DHW and HVAC, to understand the real possibility of implementing a waste heat recovery system. When this possibility was verified, a recovery system was designed.

After the recovery system was designed, it was made a financial analysis to the components necessities for its correct function and further confront with the costs for DHW and HVAC, with electricity from the grid. In this sense, it was concluded that it will take 4 years and 8 months to recover the amount invested in this recovery system for

the company. According to RACE HVAC Project Head, the time range is considerable for a company which is pursuing faster results.

Finally, it is recommended for future studies on this regard namely a more efficient way to analyse the data in order to get more accurate results which are closer to reality, creating a program in a programming language, i.e python. Additionally, the company should on a first stage study the demand for DHW and/or HVAC to be more precise when analysing the results, and to achieve better results. It is also recommended that a more detailed study of the heat himself could be more suitable for the veracity of this theme, as well as a more detailed study using as an average the days with the most and least available energy rejected from the system, as well as the demand.

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ANNEX A: CO₂ flow rate in the system

This annex is destined to the presentation of the calculation of the flow rate of the compressor.

A.1. Calculation of the flow rate

To carry out a study of the operating conditions of compressors it is necessary to use performance charts of the same, provided by the manufacturers. Those maps can be represented, according to the norms EN 12900 e AHRI 570, through a third order polynomial.

For compressors operating in subcritical regime:

$$Y = C_1 + C_2 \times T_e + C_3 \times T_c + C_4 \times T_e^2 + C_5 \times T_e \times T_c + C_6 \times T_c^2 + C_7 \times T_c^3 + C_8 \times T_e^2 \times T_c + C_9 \times T_e \times T_c^2 + C_{10} \times T_c^3 \quad (\text{A. 1})$$

With,

Y – represents the variable to be calculated

C₁ to C₁₀ represent the compressor coefficients

T_e represents the evaporator temperature

T_c represents the condensation temperature

For compressors operating in transcritical regime:

$$Y = C_1 + C_2 \times T_e + C_3 \times P_{GC} + C_4 \times T_e^2 + C_5 \times T_e \times P_{GC} + C_6 \times P_{GC}^2 + C_7 \times P_{GC}^3 + C_8 \times T_e^2 \times P_{GC} + C_9 \times T_e \times P_{GC}^2 + C_{10} \times P_{GC}^3 \quad (\text{A. 2})$$

With,

P_{GC} – represents the pressure in the gas cooler.

These polynomials, based on empirical tests, make it possible to calculate certain compressor characteristics depending on operating conditions (ambient temperature, evaporating temperature and superheating), such as electric power, flow rate or current. Knowing one of these characteristics, for certain operating conditions, and using a linear approximation (interpolation or extrapolation), it is possible to calculate the others [21].

In this part it will be calculated the flow rate knowing the electric power used. With the following operation conditions:

- Evaporation temperature MT of -10°C.
- Ambient temperature of 30°C, that correspond to an exit temperature of the gas cooler of 30.6°C and a gas cooler pressure of 79.1 bar.
- Overheating at the entrance of the compressor of 10K.
- Electric power of the compressor of 15 KW.

The representative coefficients of the operation in study are in Table A.1 to A.4.

Tabel A 1: Regression coefficients coefficients of a compressor at 25Hz in transcritical mode

C_1	528
C_2	16.4
C_3	-1.95
C_4	0.240
C_5	-0.0300
C_6	0.00
C_7	0.00
C_8	0.00
C_9	0.00
C_{10}	0.00

Tabel A 2: Regression flow rate coefficients of a compressor at 70Hz in transcritical mode

C_1	1748.45
C_2	55.02
C_3	-6.26
C_4	0.84
C_5	-0.08
C_6	0.01
C_7	0.01
C_8	0.00
C_9	0.00
C_{10}	0.00

Tabel A 3: Regression power coefficients of a compressor at 25Hz in transcritical mode

C_1	-5362
C_2	-255.3
C_3	240.2
C_4	-2.950
C_5	3.490
C_6	-1.350
C_7	-0.0100
C_8	0.00
C_9	0.00
C_{10}	0.00

Tabel A 4: Regression power coefficients of a compressor at 70Hz in transcritical mode

C_1	-14357
C_2	-659.35
C_3	632.29
C_4	-7.3300
C_5	8.7100
C_6	-3.4500
C_7	-0.030000
C_8	0.010000
C_9	-0.010000
C_{10}	0.010000

Through equation A.2 and tables C 1 to C 4, it is possible to calculate the values of the polynomial, Y.

Using and interpolation it is possible to calculate the flow rate ($\dot{m}$).

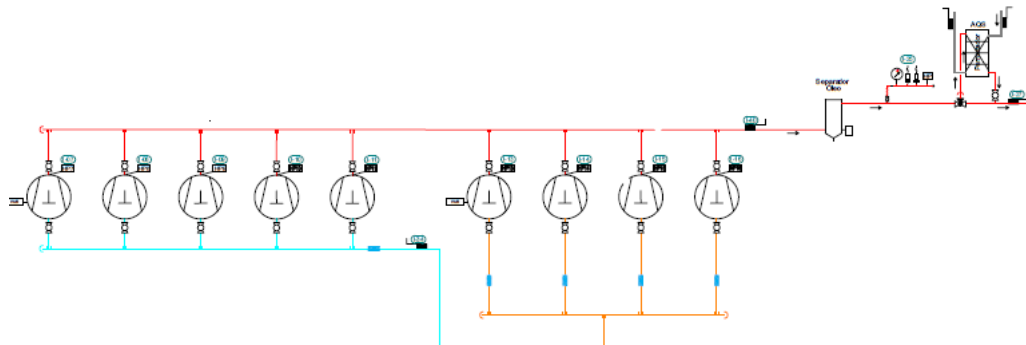


Figure A1 - Compressors MT and heat exchanger line

ANNEX B: Results obtained from the data collected

Due to the amount of data obtained, Table C 1 and Table C 2 represent the data that was measured and therefore treated. This is an example of the data, since it was measured in thirty seconds range, and it was not possible to allocate all the data in this section.

Table B 1: Data collected by RACE from the KoolSys system.

Date/Time	GC_Inlet_Temp [°C]	gasCoolerPressure [bar]	GC_Out_Temp [°C]	Ext_Temp [°C]	MT_Comp_flow rate [kg/s]	IT_Comp_flow rate [kg/s]
07/01/2023 03:57	66,4	50,0	12,7	9,50	0,300	0,0400
07/01/2023 03:57	66,2	47,2	13,7	9,60	0,300	0,0400
07/01/2023 03:58	64,7	46,6	13,1	9,50	0,300	0,0400
07/01/2023 03:58	63,0	46,8	12,5	9,40	0,300	0,0400
07/01/2023 03:59	61,4	47,4	12,0	9,40	0,290	0,0400
07/01/2023 03:59	60,2	48,3	11,7	9,40	0,280	0,0400
07/01/2023 04:00	59,6	49,3	11,7	9,40	0,270	0,0400
07/01/2023 04:00	59,4	50,2	11,8	9,50	0,270	0,0400
07/01/2023 04:01	59,6	50,8	12,0	9,30	0,260	0,0400
07/01/2023 04:01	60,1	50,4	12,3	9,30	0,250	0,0400

Table B 2: Data calculated from the KoolSys collected data

Date/Time	GC mass flow CO2 (kg/s)	GC_h_inlet (kJ/kg)	GC_h_outlet (kJ/kg)	Heat released (kW)
07/01/2023 03:57	0,330	502	233	88,6
07/01/2023 03:57	0,340	505	426	27,2
07/01/2023 03:58	0,340	503	426	26,7
07/01/2023 03:58	0,340	501	424	26,1
07/01/2023 03:59	0,330	499	231	87,5
07/01/2023 03:59	0,320	496	230	85,8
07/01/2023 04:00	0,310	494	230	82,3
07/01/2023 04:00	0,310	493	230	80,4
07/01/2023 04:01	0,300	493	230	78,7
07/01/2023 04:01	0,300	494	232	78,1

Table B 3: Released hourly energy for an average day of each month in kWh.

	Julho [kWh]	Agosto [kWh]	Setembro [kWh]	Outubro [kWh]	Novembro [kWh]	Dezembro [kWh]	Janeiro [kWh]
00:00 - 01:00	188	185	197	139	120	72	130
01:00 - 02:00	171	164	171	127	97	64	93
02:00 - 03:00	167	165	176	130	99	64	83
03:00 - 04:00	145	148	162	117	92	66	85
04:00 - 05:00	153	152	163	123	97	63	81
05:00 - 06:00	155	156	172	127	98	66	82
06:00 - 07:00	178	177	192	149	127	97	109
07:00 - 08:00	255	246	269	224	174	136	156
08:00 - 09:00	240	241	265	209	169	139	161
09:00 - 10:00	231	226	255	182	161	159	183
10:00 - 11:00	248	236	237	225	169	170	174
11:00 - 12:00	260	263	229	251	184	174	180
12:00 - 13:00	250	260	204	248	183	192	173
13:00 - 14:00	253	245	180	235	175	188	170
14:00 - 15:00	268	269	187	245	176	185	179
15:00 - 16:00	265	263	174	233	174	176	179
16:00 - 17:00	285	284	175	258	186	195	191
17:00 - 18:00	284	288	185	261	193	183	191
18:00 - 19:00	253	264	182	233	184	180	197
19:00 - 20:00	245	242	188	234	172	160	168
20:00 - 21:00	266	247	223	255	186	178	188
21:00 - 22:00	264	251	231	222	183	175	179
22:00 - 23:00	216	228	211	182	149	100	159
23:00 - 24:00	212	222	215	168	130	86	145

Table B 4: Released average energy for each day of each month [kWh]

	July [kWh]	August [kWh]	September [kWh]	October [kWh]	November [kWh]	December [kWh]	January [kWh]
Média:	5459,65	5300,49	4866,43	4678,10	3584,38	3594,97	3156,28
Total:	169249	164315	145992	145021	107531	111444	97844,6
Dia 1	4870,05	5914,36	4541,91	5072,00	4707,46	3249,28	3595,94
Dia 2	5124,79	5411,93	4535,46	4195,39	4714,36	2876,75	3659,41
Dia 3	5234,17	5543,53	4590,97	4357,42	4685,84	2985,61	3633,04
Dia 4	5124,74	5410,92	4391,45	4507,41	3671,94	3279,61	2895,30
Dia 5	5049,23	4605,43	4518,01	4521,48	3525,88	2758,01	2191,95
Dia 6	5910,65	5465,51	4530,05	4635,14	3376,68	3089,26	1492,28
Dia 7	5836,52	5439,63	4541,51	4914,96	3628,28	3076,51	3814,03
Dia 8	5518,82	5398,76	4685,91	4879,80	4009,16	4438,41	3871,89
Dia 9	5517,77	5325,10	4490,33	4819,43	3771,85	3711,52	3664,07
Dia 10	5502,29	5312,70	4557,75	4533,70	3770,93	3470,80	3682,00
Dia 11	5449,50	5279,72	4287,48	4816,56	3072,55	3736,76	3679,64
Dia 12	5437,04	5260,98	5211,59	4731,58	3593,62	4102,98	3507,46
Dia 13	5637,11	5354,63	5340,67	4761,60	3603,30	3727,28	3313,91
Dia 14	5515,81	5338,79	5444,98	5703,85	3635,67	4008,48	3482,12
Dia 15	5511,19	5324,38	5258,76	4759,78	3636,72	3745,30	3464,28
Dia 16	5510,10	5313,76	4885,97	4647,95	3583,52	3643,73	3508,40
Dia 17	5509,01	5312,14	4616,29	4724,38	3556,62	3776,48	3537,64
Dia 18	5509,97	5231,95	4386,38	4544,17	3674,42	3820,14	2653,06
Dia 19	5518,60	5309,55	4698,35	4128,49	3382,52	3832,06	4180,29
Dia 20	5530,26	5219,70	4800,48	4752,89	3565,18	3793,35	3633,18
Dia 21	5514,99	5268,11	4545,37	4790,19	3639,79	3802,79	3152,23
Dia 22	5428,11	5490,96	4917,18	4557,82	2348,52	3773,41	2804,51
Dia 23	5491,11	5551,90	4980,82	4373,05	3392,94	3777,42	2720,08
Dia 24	5306,07	5616,56	5385,64	4339,71	3365,71	3796,52	2699,06
Dia 25	5371,43	5581,21	5213,28	4659,15	3338,44	3215,12	2730,09
Dia 26	4836,90	5324,91	5056,47	4914,03	3290,44	3791,29	2602,32
Dia 27	5354,12	5104,50	5277,42	4836,50	3277,29	3760,56	3066,66
Dia 28	6131,63	5140,16	5365,21	4638,64	3236,16	3581,09	2610,65
Dia 29	5624,82	4105,94	5636,73	4616,99	3178,50	3561,95	2582,99
Dia 30	5646,66	5203,60	5300,38	4625,44	3297,07	3640,57	2702,66
Dia 31	5725,73	5153,84	-	4661,49	-	3621,01	2713,49

Table B 5: Heating loads requirement by day of each month in kWh (Simulated)

	July [kWh]	August [kWh]	September [kWh]	October [kWh]	November [kWh]	December [kWh]	January [kWh]
00:00 - 01:00	0,00	0,00	0,00	6,80	2,30	6,10	32,2
01:00 - 02:00	0,00	0,00	0,00	6,50	2,10	6,40	34,3
02:00 - 03:00	2,90	0,00	0,00	6,50	1,80	9,10	34,2
03:00 - 04:00	6,80	0,00	0,00	6,50	1,50	12,0	36,9
04:00 - 05:00	7,50	0,00	0,00	6,70	1,30	14,3	37,4
05:00 - 06:00	7,40	0,00	0,00	7,00	1,10	15,7	40,6
06:00 - 07:00	7,30	0,00	0,00	7,10	1,20	16,9	40,7
07:00 - 08:00	2,40	0,00	0,00	5,70	2,10	17,1	50,7
08:00 - 09:00	0,00	0,00	0,00	4,40	0,00	16,1	47,9
09:00 - 10:00	0,00	0,00	0,00	1,60	0,00	8,20	43,8
10:00 - 11:00	0,00	0,00	0,00	0,00	0,00	4,00	31,5
11:00 - 12:00	0,00	0,00	0,00	0,00	0,00	3,50	21,4
12:00 - 13:00	0,00	0,00	0,00	0,00	0,00	3,00	16,5
13:00 - 14:00	0,00	0,00	0,00	0,00	0,00	1,50	10,3
14:00 - 15:00	0,00	0,00	0,00	0,00	0,00	1,30	9,8
15:00 - 16:00	0,00	0,00	0,00	0,00	0,00	2,00	10,2
16:00 - 17:00	0,00	0,00	0,00	0,00	0,00	2,80	11,1
17:00 - 18:00	0,00	0,00	0,00	0,00	0,00	5,20	13,6
18:00 - 19:00	0,00	0,00	0,00	0,00	0,00	5,60	17,4
19:00 - 20:00	0,00	0,00	0,00	0,00	0,00	5,20	24,2
20:00 - 21:00	0,00	0,00	0,00	0,00	0,00	5,90	31,1
21:00 - 22:00	0,00	0,00	0,00	6,40	0,00	6,60	28,2
22:00 - 23:00	0,00	0,00	0,00	7,20	0,00	5,50	41,5
23:00 - 24:00	0,00	0,00	0,00	5,70	0,00	6,80	45,0

Table B 6: Heating loads requirement by day of each day of each month in kWh (Simulated)

	July [kWh]	August [kWh]	September [kWh]	October [kWh]	November [kWh]	December [kWh]	January [kWh]
Dia 1	34,0	0	0	78,0	13,0	181	711
Dia 2	2	0	0	50,0	25,0	150	790
Dia 3	5	24,0	0	36,0	26,0	127	806
Dia 4	0	37,0	0	92,0	152	155	512
Dia 5	20,0	6,00	0	113	155	329	167
Dia 6	0	0	0	138	186	334	149
Dia 7	0	0	0	143	294	485	269
Dia 8	0	23,0	7,00	155	362	639	379
Dia 9	0	0	0	127	268	279	451
Dia 10	0	0	0	137	252	343	376
Dia 11	0	0	0	119	285	104	114
Dia 12	0	0	5,00	142	99,0	133	93,0
Dia 13	0	0	75,0	133	115	315	107
Dia 14	0	0	103	142	114	106	131
Dia 15	0	31,0	80,0	117	103	489	142
Dia 16	36,0	45,0	21,0	80,0	138	744	189
Dia 17	12,0	0	48,0	113	120	772	266
Dia 18	11,0	0	14,0	135	112	520	285
Dia 19	0	0	77,0	130	157	277	344
Dia 20	27,0	0	52,0	107	244	246	419
Dia 21	18,0	0	0	43,0	512	244	452
Dia 22	0	0	0	25,0	627	118	531
Dia 23	0	0	0	89,0	521	194	591
Dia 24	0	0	0	87,0	173	187	615
Dia 25	0	43,0	0	0	105	84,0	739
Dia 26	2,00	45,0	0	0	86,0	98,0	526
Dia 27	52,0	45,0	3,00	38,0	122	121	418
Dia 28	0	7,00	72,0	8,00	158	170	464
Dia 29	22,0	0	51,0	0	139	140	250
Dia 30	46,0	0	23,0	13,0	131	532	285
Dia 31	56,0	0	[-]	40,0	[-]	666	345

Table B 7: Heating Loads requirements for HVAC by month in kWh (Simulated)

Air System Simulation Results (Table 1) :

Month	Central Cooling Coil Load (kWh)	Central Cooling Eqpt Load (kWh)	Central Unit Clg Input (kWh)	Central Heating Coil Load (kWh)	Central Heating Coil Input (kWh)	Supply Fan (kWh)	Vent. Reclaim Device (kWh)
January	0	0	0	11914	11914	0	0
February	0	0	0	6622	6622	0	0
March	0	0	0	5459	5459	0	0
April	855	855	221	3722	3722	0	0
May	1388	1388	345	2849	2849	0	0
June	7072	7071	1769	632	632	0	0
July	11138	10665	2637	349	349	0	0
August	12089	11837	2946	307	307	0	0
September	5783	5731	1400	632	632	0	0
October	1872	1872	487	2632	2632	0	0
November	23	23	6	5795	5795	0	0
December	0	0	0	9288	9288	0	0
Total	40221	39443	9811	50200	50200	0	0

Table B 8: Hourly available energy after retrieving the demand for heating loads [kWh]

	Julho [kWh]	Agosto [kWh]	Setembro [kWh]	Outubro [kWh]	Novembro [kWh]	Dezembro [kWh]	Janeiro [kWh]
00:00 - 01:00	188	185	197	132	118	66,0	98,0
01:00 - 02:00	171	164	171	120	95,0	58,0	59,0
02:00 - 03:00	164	165	176	123	97,0	54,0	49,0
03:00 - 04:00	139	148	162	110	91,0	54,0	48,0
04:00 - 05:00	146	152	163	117	95,0	49,0	44,0
05:00 - 06:00	148	156	172	120	97,0	50,0	42,0
06:00 - 07:00	171	177	192	142	126	80,0	69,0
07:00 - 08:00	252	246	269	218	172	119	105
08:00 - 09:00	240	241	265	205	169	123	113
09:00 - 10:00	231	226	255	181	161	151	139
10:00 - 11:00	248	236	237	225	169	166	143
11:00 - 12:00	260	263	229	251	184	170	158
12:00 - 13:00	250	260	204	248	183	189	157
13:00 - 14:00	253	245	180	235	175	186	160
14:00 - 15:00	268	269	187	245	176	183	169
15:00 - 16:00	265	263	174	233	174	174	169
16:00 - 17:00	285	284	175	258	186	192	180
17:00 - 18:00	284	288	185	261	193	178	177
18:00 - 19:00	253	264	182	233	184	175	180
19:00 - 20:00	245	242	188	234	172	154	144
20:00 - 21:00	266	247	223	255	186	172	156
21:00 - 22:00	264	251	231	216	183	168	151
22:00 - 23:00	216	228	211	175	149	94,0	117
23:00 - 24:00	212	222	215	163	130	79,0	100

Table B 9: Hourly consumption profile of hot water during an average day of each month in litres (Simulated)

Temperatura nominal de consumo: 50°C (N.B. existem válvulas misturadoras)												
Temperaturas de abastecimento ao depósito (°C):												
	Jan 13	Fev 13	Mar 14	Abr 15	Mai 16	Jun 18	Jul 20	Ago 20	Set 19	Out 17	Nov 15	Dez 13
Perfis de consumo (1)												
hora	Jan	Fev	Mar	Abr	Mai	Jun	Jul	Ago	Set	Out	Nov	Dez
01												
02												
03												
04												
05												
06												
07												
08	50	50	50	50	50	50	50	50	50	50	50	50
09	200	200	200	200	200	200	200	200	200	200	200	200
10	5	5	5	5	5	5	5	5	5	5	5	5
11	10	10	10	10	10	10	10	10	10	10	10	10
12	15	15	15	15	15	15	15	15	15	15	15	15
13	20	20	20	20	20	20	20	20	20	20	20	20
14	50	50	50	50	50	50	50	50	50	50	50	50
15	50	50	50	50	50	50	50	50	50	50	50	50
16	20	20	20	20	20	20	20	20	20	20	20	20
17	15	15	15	15	15	15	15	15	15	15	15	15
18	10	10	10	10	10	10	10	10	10	10	10	10
19	10	10	10	10	10	10	10	10	10	10	10	10
20	15	15	15	15	15	15	15	15	15	15	15	15
21	20	20	20	20	20	20	20	20	20	20	20	20
22	50	50	50	50	50	50	50	50	50	50	50	50
23												
24												
diário	540	540	540	540	540	540	540	540	540	540	540	540

Table B 10: Heating water requirements by month for AQS system in kWh (simulated)

Balanço energético mensal e anual						
Mês	Rad.Horiz. kWh/m ²	Rad.Inclin. kWh/m ²	Desperdiçado kWh	Fornecido kWh	Carga kWh	Apoio kWh
Janeiro	65	106	,	346	725	379
Fevereiro	84	123	,	369	644	275
Março	137	170	,	464	695	231
Abril	169	180	,	477	658	181
Maio	206	195	,	492	653	162
Junho	227	203	,	489	596	107
Julho	239	221	,	530	591	61
Agosto	212	217	1,	540	590	51
Setembro	159	189	,	498	590	91
Outubro	106	147	,	430	650	220
Novembro	68	112	,	347	668	321
Dezembro	55	90	,	301	713	412
Anual	1726	1952	1,	5283	7774	2491

Table B 11: Seasonal coefficient of performance (SCOP) of the HVAC system

EFICIÊNCIA SAZONAL

SCOP/SEER (1)	3.52	3.98	
Eficiência energética sazonal (aquecimento $\eta_{s,h}$ / arrefecimento $\eta_{s,c}$) (2)	137.61	156	%
Classe de eficiência energética Eurovent	B	B	

