

Faculdade de Engenharia da Universidade do Porto



**Development of an R744 Booster Refrigeration
Unit with Dual-Stage Compressors**

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Master's in Mechanical Engineering
Fluids and Energy

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Resumo

A presente dissertação apresenta o projeto feito no âmbito do Mestrado em Engenharia Mecânica da Faculdade de Engenharia da Universidade do Porto.

Deste modo, procedeu-se ao desenvolvimento de uma central de refrigeração transcrítica que incorpore compressores de duplo estágio num sistema booster de compressão paralela. Esta integração permite a separação dos estágios de evaporação, conferindo um funcionamento independente à central. O fluido de trabalho utilizado foi o R744 (CO₂), uma vez que existem limitações impostas pela comissão europeia que, não só proíbem utilização de alguns gases fluorados, mas também impõe a redução da utilização e disponibilidade destes. Por outro lado, existe um aumento vertiginoso do custo dos gases fluorados permitidos, o que torna a sua utilização menos favorável.

Numa primeira fase, começou por se estudar as tecnologias de refrigeração existentes, focando particularmente a atenção naquelas que utilizam o fluido de trabalho pretendido. Após a revisão de conceitos inicial, focou-se a atenção na definição dos requisitos do projeto. Localização, temperaturas exteriores e potência frigorífica requerida são alguns dos parâmetros fundamentais para o desenvolvimento da central. De seguida, desenvolveu-se o diagrama da central, abordando a função principal de alguns componentes. Após a definição do diagrama da central, procedeu-se à seleção dos componentes da mesma, começando pelos compressores, uma vez que, são o componente principal de qualquer instalação frigorífica. De seguida procedeu-se à seleção da tubagem do sistema, válvulas de expansão e depósito de líquido como componentes principais. Filtros, permutadores de calor e outras válvulas também foram selecionados de modo a garantir o funcionamento correto da central. O isolamento térmico e o sistema de lubrificação também são abordados, assim como o sistema de emergência.

Por fim, simulou-se em software as diferentes condições de funcionamento da central projetada, comparando-a com uma central de referência de potência semelhante. Avaliaram-se ambas as centrais a nível económico de custo inicial e a nível de performance, comparando-as com o objetivo de concluir se a versatilidade da central projetada apresenta vantagens relativamente à central de referência.

Com o trabalho efetuado foi possível chegar a um projeto de uma central que cumpre os requisitos definidos inicialmente. Por um lado, o menor número de compressores utilizados permite uma central mais compacta e consequentemente mais económica. Por outro lado, não apresenta vantagens significativas a nível de eficiência energética quando comparada com a central de referência.

Abstract

This work presents a master's degree dissertation in Mechanical Engineering in the Faculty of Engineering of the University of Porto.

The aim of this dissertation is to develop a parallel compression transcritical booster refrigeration system that uses dual-stage compressors. Integrating dual-stage compressors allows the complete separation of evaporation stages, which makes them independent from each other. The working fluid used was R744 (CO₂) as there are limitations imposed by the European Commission that not only prohibits the use of some fluorinated gases, but also imposes a reduction in their use and availability. On the other hand, there is a steep increase in the cost of allowed fluorinated gases, which makes their use less favourable.

Primarily, it was necessary to study existing refrigeration technologies, prioritizing the ones that utilize the pretended working fluid. After an initial concept revision, the project condition was the focus of the attention. Location, exterior temperatures, and cooling load are fundamental parameters for projecting a refrigeration unit. After defining the refrigeration unit diagram, the components were selected, beginning with the compressors, as they are the main component of any refrigeration unit. Next, it was selected the refrigerant piping, the expansion valves and the liquid receiver as the main components. Filters, heat exchangers and other valves were also selected to ensure the correct operation of the refrigeration unit. Thermal insulation and the lubrication system are also covered, as it is the emergency system.

Lastly, a software was used to simulate different operating conditions of the projected refrigeration unit, comparing the results with a reference booster refrigeration system of similar power. Both refrigeration units were evaluated in terms of initial cost and performance, comparing them in order to evaluate if the versatility of the designed plant presents any advantages in relation to the reference unit.

Through this work it was possible to achieve a refrigeration unit project that meets the requirements initially defined. On the one hand, the smaller number of compressors used allows for a more compact and therefore more economical unit. On the other hand, it does not present significant advantages in terms of energy efficiency when compared with the reference refrigeration unit.

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José Miguel Martins Soares de Vasconcelos Ligeiro

“How hard can it be?”

Jeremy Clarkson

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Abbreviations and Symbols

A	Area	[m ²]
COP	Coefficient of Performance	
D	Diameter	[m]
DHW	Domestic Hot Water	
E_{Annual}	Annual energy consumption	[kWh]
F	CO ₂ emissions per kWh	[kg CO ₂ /kWh]
f	Darcy's coefficient of friction	
GWP	Global Warming Potential	
h	Enthalpy	[J/kg]
HP	High Pressure	
HFC	Hydrofluorocarbons	
l	Length	[m]
LP	Low Pressure	
LT	Low Temperature	
m	Mass	[kg]
$\dot{m}$	Mass Flow Rate	[kg/s]
MT	Medium Temperature	
ODP	Ozone Depletion Potential	
p	Pressure	[Pa]
Q	Heat	[J]
$\dot{Q}_L$	Cooling load	[W]
q_L	Cooling effect	[J/kg]
Re	Reynolds Number	
s	Entropy	[J/kg*K]
T	Temperature	[°C]
TEWI	Total Equivalent Warming Impact	
u	Velocity	[m/s ²]
V	Volume	[m ³]

$\dot{V}$	Volume flow rate	[m ³ /s]
W	Compression Work	[J]
$\dot{W}$	Compression Power	[W]
Δp	Pressure difference	[Pa]
ΔT	Temperature difference	[°C]
ε	Effective roughness height	[m]
η	Efficiency	
μ	Dynamic viscosity	[pa*s]
ν	Kinematic viscosity	[m ² /s]
ρ	Density	[kg/m ³]

Chapter 1

Introduction

In this first chapter, it is presented the contextualization of the project and the reasons that led to its realisation. Afterwards, are also presented the objectives of this project, as well as the hosting company, followed by, the structure of the dissertation, which is fundamental to understand the creation of the project.

1.1 - Contextualization and Motivation

With carbon emission levels on the rise and ozone layer destruction, a rapid and far-reaching action is needed to counteract global warming. Among many available strategies, CO₂ is nowadays more and more often proposed as a refrigeration fluid solution for heating, cooling and refrigeration purposes, since it has an extremely low global warming potential (GWP) and is natural.

Refrigerants should be selected regarding their potential influence on the global environment (ODP – Ozone Depletion Potential, GWP) as well as their possible effects on the local environment. Evaluation of the environmental performance requires a lifecycle approach. Concerning global climate change, the TEWI - Total Equivalent Warming Impact approach is generally used as the basis. The TEWI is a way of assessing global warming by combining the direct contribution of refrigerant emissions into the atmosphere, with the indirect contribution of the carbon dioxide and other gas emissions resulting from the energy required to operate the refrigerating system over its operational life [1]. The lower the TEWI value of a system, the lower its contribution to the increase in the greenhouse effect, and, therefore, a more environmentally friendly system.

$$TEWI = GWP * m_{released} + F * E_{annual} \quad (1)$$

Where:

GWP – Global Warming Potential

$m_{released}$ - Mass leaked

F – CO₂ emissions per kWh

E_{annual} - Annual energy consumption

As equation (1) suggests, the total equivalent warming impact is divided in two parts: the direct effect, which results from the release of refrigerant to the atmosphere, and the indirect effect, which results from the energy consumption of the system.

Refrigeration systems use HFC (hydrofluorocarbons) gases in most applications, from cold display areas to deep freezer rooms and anything in between. As a result, the Environmental Committee of the European Parliament, the European Council and the European Commission agreed on a phase-down of the consumption of fluorinated green-house gases (F-Gases) by 79% by the year 2030 [2]. This value defined in 2015 was based on the average consumption between 2009 and 2012. The percentage to reduce is not the same for every gas, it depends on their GWP. Thus, the restrictions on gases with a GWP of 2000 will be higher than those on gases with a GWP of 500. The GWP of CO₂ is the reference value and is 1. The remaining values are defined in tonnes of CO₂ equivalent, i.e., one tonne of a gas with a GWP of 1000 has the same global warming potential as one thousand tonnes of CO₂. As a consequence, it was adopted a control system for hydrofluorocarbons gases manufacturers, which obliges them to report all production, in order to control the consumption of those gases [2].

New regulations also include, for several applications, a maximum admissible GWP value of refrigerant. This means since 2020 some refrigerants and technologies were already limited certain segments and, up to the end of that year, more prohibitions and limitations have taken place. After 2022, new centralised refrigeration systems for commercial use with cooling loads greater than 40 kW, multiple cooled spaces and at least two compressors, refrigerants with GWP higher than 150 are not allowed. The exception to this are the primary units in cascade systems, where the gases with GWP up to 1500 can be used. There are also new regulations regarding containment, leak checks and marking. Regular checks are mandatory from a refrigerant charge higher than 5 tonnes of CO₂ equivalent i.e., 3,5 kg of R134a or 1,3 kg of R404A.

The quantitative limitation (Figure 1) is the biggest challenge of the new F-Gas Regulation. It describes the total amount of consumption allowed, i.e., refilling, leakage quantities, losses due to damages and maintenance.

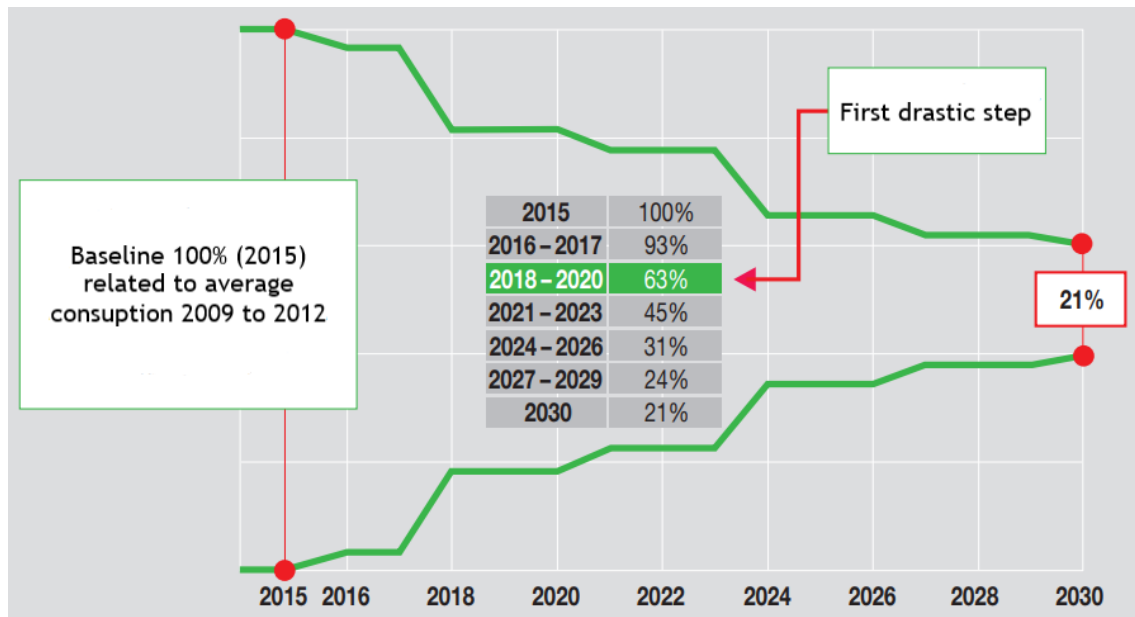


Figure 1 - Quantitative limit "phase-down" until 2030 (adapted from [2]).

Considering an average GWP value of 2250 for all HFC's used as refrigerants in the EU, an average GWP of clearly below 500 would be the result of the quantitative limit of 21%. All the refrigerants with a GWP below 500 currently available as an alternative, are flammable, some are also toxic, with the exception of the CO₂. This low average value allows the conclusion that the reduction of consumption by means of the planned use bans (up to 2022) alone will not be enough to reach the quantitative limit. Adding to that, there is an increasing need of refrigeration systems, so it is easy to conclude that alternative refrigerants and/or technologies will have to be used also in fields of application that are not directly affected by the bans.

With all these restrictions on hydrofluorocarbons gases it becomes necessary to engineer more environmentally friendly solutions. Besides, the lack of availability of these refrigerants is drastically increasing their prices and it is expected to continue to rise, making the issue, from the point of view of the companies, not only environmental but mainly financial. Table 1 presents the Hydrofluorocarbons gases price evolution.

Table 1 - Hydrofluorocarbons gases price evolution [3].

Price table		October 2017	October 2018	March 2022
Description	Capacity [kg]	Price per kg [€]	Price per kg [€]	Price per kg [€]
R134a	60	17,07	36,07	38,14
R407F	55	21,11	50,24	52,86
R410A	50	37,13	61,83	63,81
R448A	50	29,16	36,14	38,07
R744	37,5	2,10	2,10	2,15

As it is demonstrated in Table 1 there is an increase in price of R134a and other HFC gases between October 2017, June 2018 and March 2022 and it can be expected to continue to increase. In the same table, it can be seen that the price of CO₂ (R744) remains fairly constant throughout the years.

1.2 - The company, RACE

The company **RACE**, S.A. – **R**efrigeration & **A**ir-**C**onditioning **E**ngineering – originates from the merger of two companies Selfrio S.A. and Sistavac S.A. Selfrio S.A., the first of the companies to emerge, was founded in 1985 and Sistavac S.A. appeared later in 1992 with the introduction of the HVAC area, being 100% owned by Selfrio S.A. It is in 2011 that the merger of the companies occurs, which now adopt the name Sistavac S.A. In 2017, RACE is launched, rebranding the company.

The company develops solutions for commercial and industrial refrigeration, including the design and installation of refrigeration plants and has an after-sales department. Besides refrigeration it also develops air conditioning projects. It has a team off around 300 employees, and it is present not only in Portugal, but also in Brazil and Romania [4].



1.3 - Objectives

The aim of this thesis is to develop a refrigeration unit that integrates dual-stage compressors on a traditional booster refrigeration system and utilizes CO₂ as the refrigerant in transcritical state. These compressors will provide complete separation of the evaporation stages, allowing the unit to operate in different modes.

Thus, the main objectives to be achieved are as it follows:

- Develop a refrigeration unit diagram with parallel compression and utilizing dual-stage compressors;
- Integrate dual-stage compressors on a standard booster refrigeration system;
- Dimensioning and selecting all the components of the refrigeration unit;
- Estimate the total component cost of the projected refrigeration unit;
- Comparing the efficiency of the projected refrigeration unit with an existing one;
- Approaching the world of commercial refrigeration.

1.4 – Structure

This work is structured in 6 chapters:

Chapter 1 presents the context of this dissertation and the motivation behind it. The company that proposed the project and its objectives are also presented as well as, the structure of the document.

In Chapter 2, one can find, the state of the art for this work, the theoretical refrigeration cycles, the actual refrigeration cycles as well as their variations and characteristics. Finally, CO₂ booster systems are explained.

Next, in Chapter 3, it can be found the problem definition for this thesis. Project conditions and requirements are established. Alternative operating conditions are also defined.

After the problem definition in chapter 4, it is proposed a solution for the problem. A diagram containing the relative position of different components is presented, and all components are analysed and dimensioned according to their function.

Furthermore, in chapter 5, it is presented a project's analysis and comparison with a reference CO₂ booster system. The results are shown and analysed.

Finally, in Chapter 6, the conclusions and future work are presented.

Chapter 2

Refrigeration Cycles

It is a well-known fact that heat flows in the direction of decreasing temperature, i.e., from a high temperature region to the most stable state which is ambient temperature.

Nevertheless, the reverse process (i.e., heat transfer from low to high temperature) does not occur by itself (Clausius Definition of Second Law of Thermodynamics) [5]. This process requires a special device which is the refrigeration system.

The main objective of a refrigeration system is to remove heat from a material, which can be in a solid, liquid, or gas state, allowing it to be below ambient temperature.

Nowadays, a refrigeration system works by pressurizing and depressurizing a fluid in a cycle, with the goal of cooling it down, so it can absorb heat from a material or space that is being cooled and reject excess heat to the environment.

2.1 - Reversed Carnot Cycle

The simplest refrigeration cycle is the reversed Carnot cycle. Carnot cycle is a totally reversible cycle which consists of two reversible isothermal processes and two isentropic processes.

This cycle represents the maximum efficiency of a refrigeration system for a given temperature limit (difference between the temperature of the cold and hot side).

Since it is a fully reversible cycle, all four processes can be reversed and still maintain the amount of energy constant. This will reverse the direction of the heat and work interactions, therefore producing a refrigeration cycle. Figure 2 shows the direction of the fluid and the steps that it takes in other to heat transfer on a T-s diagram.

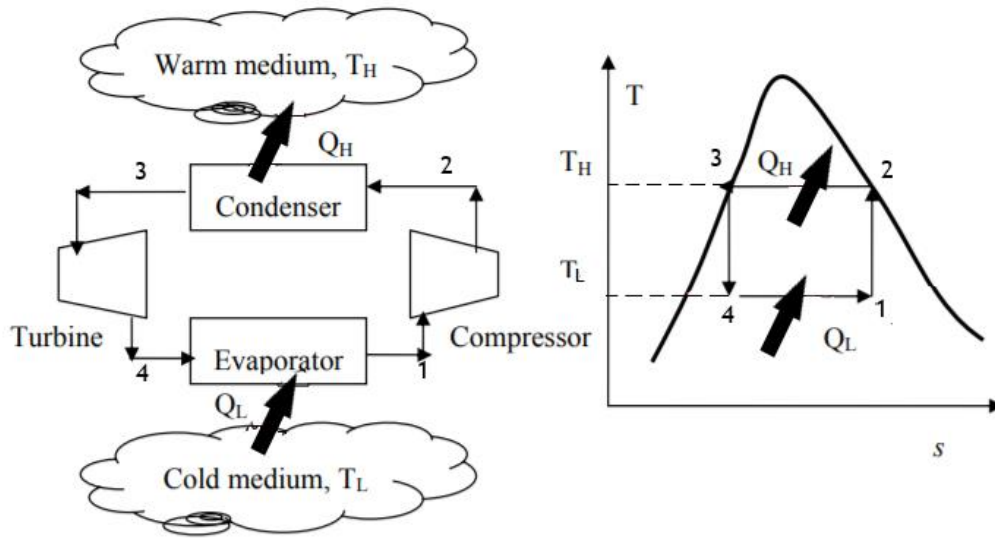


Figure 2 - Schematics of a Carnot refrigerator and T-s diagram of a reversed Carnot cycle [6].

The reversed Carnot cycle can be described in 4 different steps:

4-1: The fluid passes through the evaporator (which is a heat exchanger) absorbing heat from the cold medium. This process is isothermal and isobaric and the fluid changes phase from liquid to gas.

1-2: The fluid is compressed up the condensation pressure in a compressor. This is an isentropic process.

2-3: After the compression the fluid passes through the condenser where it changes phase from saturated gas to saturated liquid, rejecting heat to the warm medium, this is an isothermal and isobaric process.

3-4: The fluid is expanded in a turbine until it reaches the evaporation pressure, this process is an isentropic expansion.

We can conclude by analysing the Figure 2 this cycle operates between two temperatures, T_L , cold medium (temperature of evaporation) and the T_H , hot medium (temperature of condensation). Therefore, to calculate the COP, we just need to know the temperature in which the cycle operates. The temperature in equation (2) must be in K.

$$COP_{r.Carnot} = \frac{T_L}{T_H - T_L} \quad (2)$$

However, there are some limitations when trying to use this cycle in the real world. One of the main issues is the fact that the compression happens when the fluid is in a two-phase mixture (liquid and gas mixture). This will result in an unsteady compression that reduces the life span of the compressor and can hydrolock it. The same can be said about the expansion as a two-phase expansion in a turbine is also unsteady and reduces the life span of it.

Adding to that, the amount of work that can be obtained from the turbine is negligible when compared to the work needed to compress the fluid in the compressor, making it a complex and expensive option that is not worth it.

There is also the problem of not considering the necessary ΔT that heat exchangers need to reject or absorb heat, therefore it requires an infinite heat transfer area.

As a consequence, the reversed Carnot cycle is considered to be a reference theoretical cycle only, and it's used as a reference when comparing real refrigeration cycles [7].

2.2 - Ideal Vapour Compression Refrigeration Cycle

Due to the limitations mentioned on the reversed Carnot Cycle, there is a need to alter the system to obtain a feasible refrigeration cycle, mainly, the turbine is changed for an expansion valve and the refrigerant needs to be completely vaporized before it is compressed.

Figure 3 refers a typical layout of a conventional refrigeration system as well as the corresponding T-s diagram.

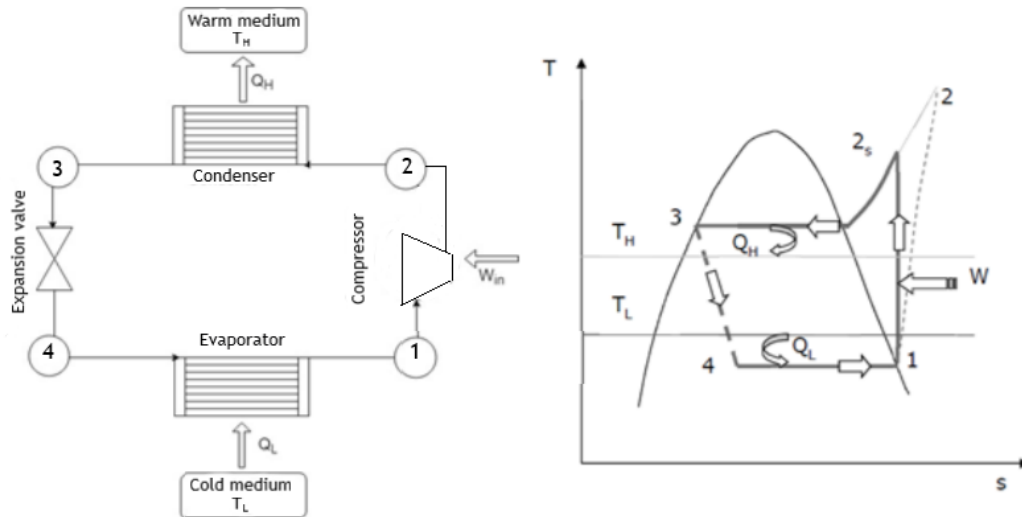


Figure 3 - Schematic of a refrigeration system and the corresponding T-s diagram (adapted from [7]).

When applying the first law of thermodynamics, we can obtain the energy balance on each component of the system.

Since the fluid is completely vaporized before it is compressed, then it must be cooled down in gas phase before it reaches condensation temperature where it changes phase to liquid.

This process is isobaric, which means it maintains the same pressure throughout, and therefore, it is better illustrated in a p-h diagram as the one in Figure 4.

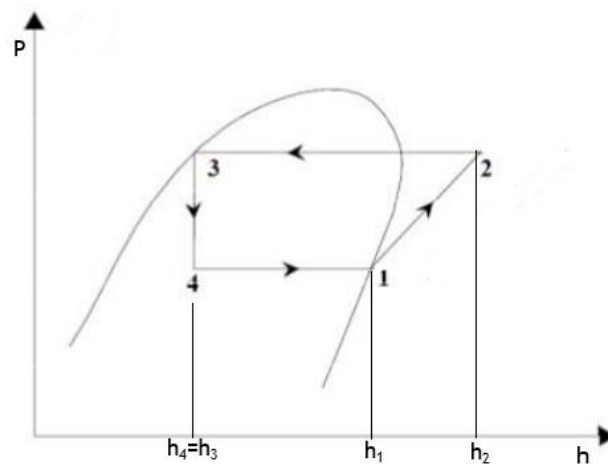


Figure 4 - p-h diagram of a conventional refrigeration system [7].

As in the reversed Carnot cycle this can be described in 4 different steps:

4-1: The fluid passes through the evaporator absorbing heat from the cold medium. This process is isothermal and isobaric and the fluid changes phase from liquid to gas.

1-2: The fluid is compressed up to the condensation pressure in a compressor. This is an isentropic process (maintains its entropy, s , in an ideal compression 1-2s).

2-3: The fluid passes through the condenser cooling down and changing phase until it becomes saturated liquid. This is also an isobaric process.

3-4: The fluid passes through the expansion valve reducing its pressure till it reaches the evaporation pressure and changing it into a mixture of liquid and gas. This process is isenthalpic as suggested by the diagram in Figure 4.

On the evaporator there is no work being done, therefore there is only a mass flow through it. The cooling effect (q_L) can be obtained by the next equation:

$$q_L = h_1 - h_4 \quad (3)$$

The respective cooling capacity ($\dot{Q}_L$) depends on the capacity of the compressor and the thermodynamic properties of the fluid, it is given by the equation:

$$\dot{Q}_L = \dot{m} * (h_1 - h_4) \quad (4)$$

The compressor power input ($\dot{W}$) can be obtained by the equation (5):

$$\dot{W} = \dot{m} * (h_2 - h_1) \quad (5)$$

On the condenser, the rate of heat released to the hot side ($\dot{Q}_H$) is obtain using equation (6):

$$\dot{Q}_H = \dot{m} * (h_3 - h_2) \quad (6)$$

The performance of this refrigeration systems is evaluated by calculating the COP of the cycle. This can be done using the next equation:

$$COP_{Cycle} = \frac{\dot{Q}_L}{\dot{W}} = \frac{h_1 - h_4}{h_2 - h_1} \quad (7)$$

The efficiency of a refrigeration cycle can be determined by comparing it with the equivalent reversed Carnot cycle using equation (8) [7]:

$$\eta_{cycle} = \frac{COP_{Cycle}}{COP_{r.Carnot}} \quad (8)$$

2.3 - Actual Vapour Compression Cycle

In the ideal vapour compression refrigeration cycle some losses are ignored, mainly the pressure drop that exists in multiple components such as the pipes of the system and the heat transfer to and from the pipes and the air surrounding the installation.

Figure 5, presents a actual vapour compression refrigeration cycle:

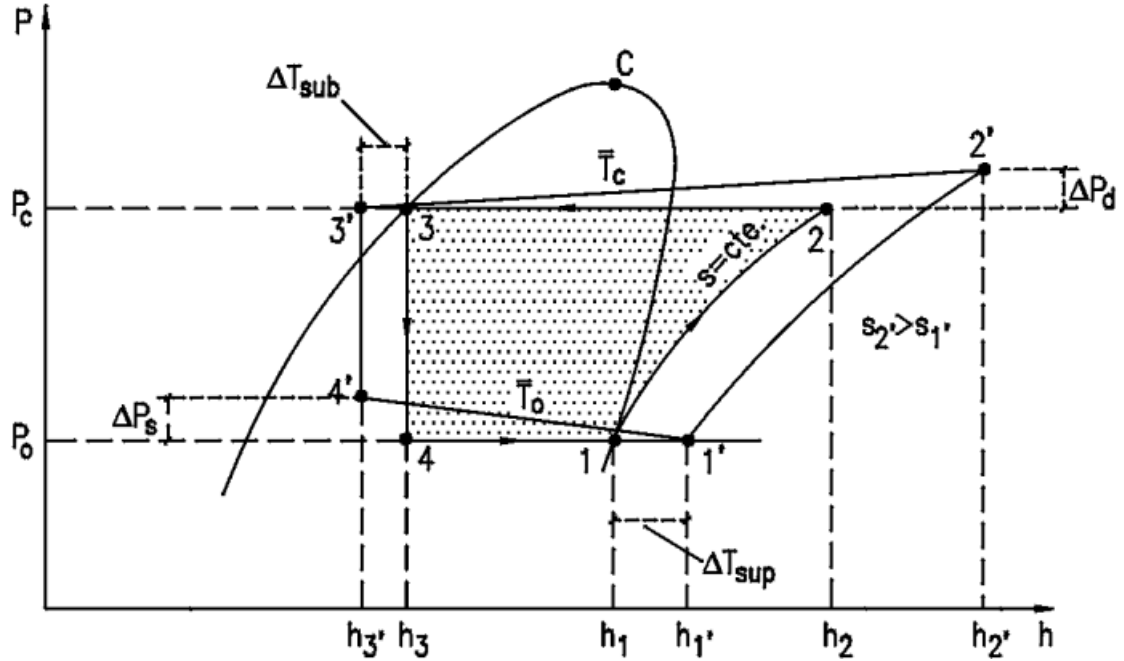


Figure 5 - P-h diagram of an actual refrigeration cycle [7].

As the diagram suggests, in an actual refrigeration cycle ($1' \rightarrow 2' \rightarrow 3' \rightarrow 4'$) there is a pressure drop in both the condenser and the evaporator, which means that the rejection and absorption of heat in the heat exchanger is not an isobaric process.

Thanks to that pressure drop, the compression work (W) done by the compressor is also higher than in the ideal refrigeration cycle (both pressure and enthalpy are higher in $2'$ than 2).

There are also some degrees of superheating (ΔT_{SUP}) and subcooling (ΔT_{SUB}). It is important to guarantee that there is only liquid between the condenser and the expansion valve because this increases the cooling effect (q_L) and moves state $4'$ further to the left in the diagram meaning its enthalpy is less than in state 4. The superheating of the fluid from state 1 to $1'$ has a similar influence on the cooling effect of the system and also guarantees that the compressor only compresses gas which increases its life span.

2.4 - Staged Compression

Staged compression is usually done using two compressors called two-stage compression.

In this case, instead of the compression process being carried out in a single stage between the maximum condensing pressure and the minimum evaporating pressure, it is fractioned into one or more intermediate pressures, using several compressors or one stage compressor.

The curve $1 \rightarrow 3'$ on the p-h diagram in Figure 6 shows the compression process for a single compression system. However, if the compression is fractioned using multiple compressors and cooling the refrigerant in between the compressors (using water or air heat exchangers/intercoolers), as it is presented on the schematic of Figure 6, it would be obtained a compression curve according to the p-h diagram, process $1 \rightarrow 2$ and $2' \rightarrow 3$. The process $2 \rightarrow 2'$ concerns the intermediate cooling of the refrigerant at a constant pressure. It is verified that the compression work is less in the second case. With the $1 \rightarrow 2$ evolution of the refrigerant being common to both processes, it is observed that $h_3 - h_{2'}$ is less than $h_{3'} - h_2$. It can also be seen that the maximum temperature reached by the refrigerant at the second compressor outlet is lower in this case, $T_3 < T_{3'}$, since state 3 has a lower enthalpy than $3'$ at the same pressure [7].

The first compressor called low pressure (LP) compressor pressurizes the fluid up to an intermediate pressure. Then the refrigerant is cooled in an intercooler before entering the high pressure (HP) compressor that takes it to condensation pressure.

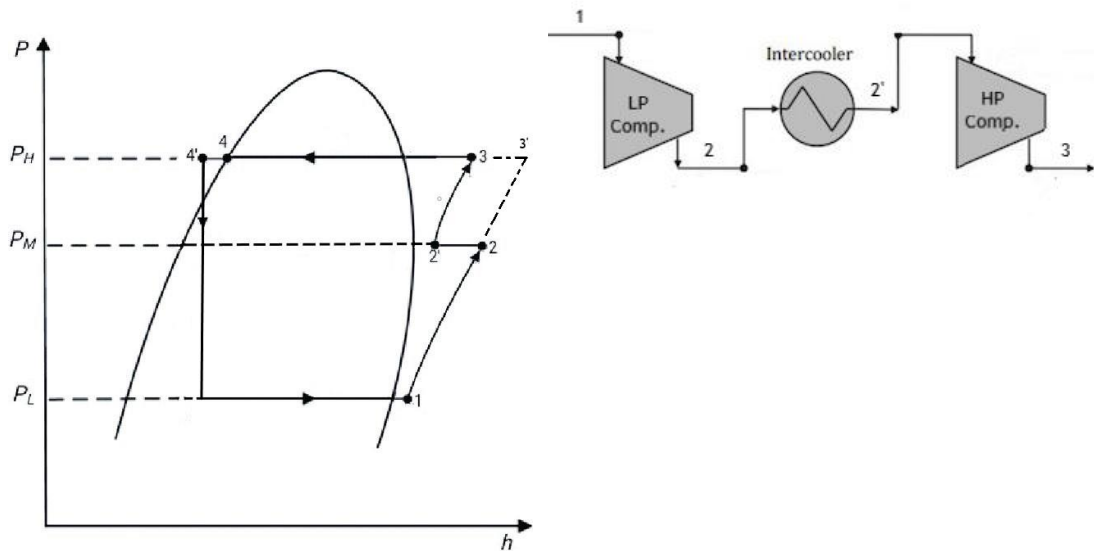


Figure 6 – Two-stage compression p-h diagram and schematic (adapted from [8]).

For the advantages to take place, the fluid must be cooled between the compressors. Otherwise, the cycle behaves the same as a single-stage compression cycle. There are also some disadvantages, in particular, it should be noted that the choice between a system with several compression stages or just one, also involves an economic study. In the first case there is an increase in the cost of the system, which will have to be compared with the eventual reduction in power consumption, i.e., the operating costs. This is more significant the lower the evaporation temperature [7].

According to [7], if the working fluid in the refrigeration system was a perfect gas, it can be concluded that the medium pressure (P_M) to be selected, in the case of a system with two stages of compression, is given by equation (9):

$$P_M = (P_e * P_c)^{0.5} \quad (9)$$

Where:

P_e is the intake pressure of the LP compressor.

P_c is the discharge pressure of the HP compressor.

Although this last equation is only valid for fluids that are considered to behave as perfect gases, it serves as a first approximation for determining the medium pressure in the case where the working fluid is a vapour. It can also be demonstrated that the maximum COP of the cycle is achieved for a medium pressure slightly higher than the one resulting from the equation (9) [7].

2.5 - Staged Expansion

A more detailed analysis of the expansion process reveals the existence of some inefficiencies. Thus, as seen already, saturated liquid enters the expansion system and is expanded to the evaporating pressure. This refrigerant's isenthalpic expansion, is represented in Figure 7 by the evolution $4' \rightarrow 8'$.

In state $8'$ there is a higher percentage of vapour than in state 8. This vapour that emerged on the expansion valve does not create cooling effect, since, in the evaporator, only a phase change occurs (the saturated liquid removes energy from the space to be cooled by changing its phase to gas). However, this vapour has to be compressed to the condensation pressure, which introduces some power losses since the vapour did not create any cooling effect.

Let us assume now that the expansion process is interrupted in state 7 at an intermediate pressure between the condensation and the evaporation one. The humid vapour that is created after the expansion valve is a mixture of saturated liquid and saturated vapour, state $7f$ and $7g$, respectively. That two-phase fluid is then directed to a liquid receiver (schematic in Figure 7), where the phases of the fluid separate from each other.

At this point, the saturated liquid in the receiver, state $7f$, is expanded for a second time, to the evaporation pressure, state 8. This state has less saturated vapour than state $8'$, which increases the creating cooling effect [7].

Staged expansion refrigeration cycles, much like the staged compression refrigeration cycles, usually consider only two stages called two-stage expansion cycles.

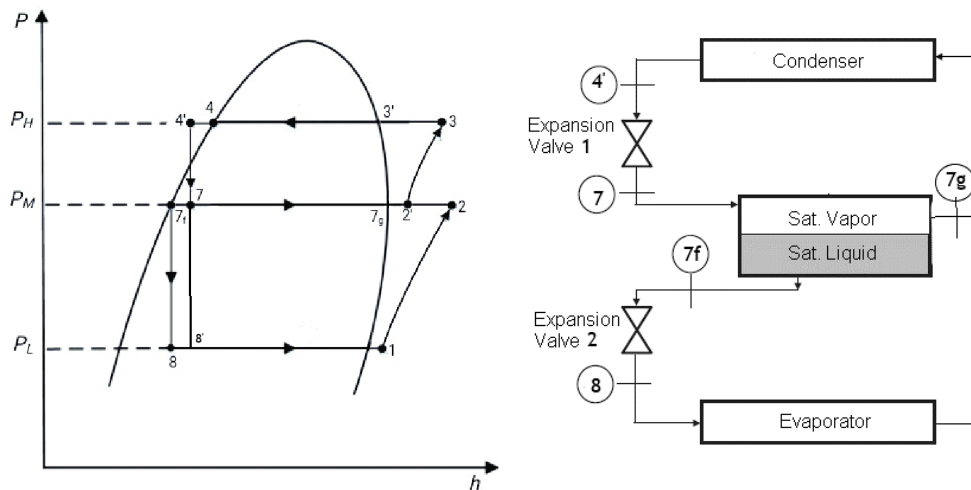


Figure 7 - Two stage expansion p-h diagram and schematics (adapted from [8]).

The saturated vapour in the liquid receiver, state $7g$, is then compressed directly to the condensing pressure. It could be expanded to the evaporation pressure and compressed after mixing with the vapour from the evaporator outlet. However, this process would represent an additional energy consumption, since it would require more compression work.

This has two main advantages. First the LP compressor can be smaller than the equivalent in a single expansion cycle. This happens because it does not need to compress all the working fluid in the cycle, it just compresses the one that passes through the evaporator. Also, the cooling effect is greater using this cycle, because, as said before, state 8 has less vapour than state 8' ($X_8 < X_{8'}$) and so its enthalpy is lower. The main disadvantage of a two-stage expansion refrigeration system is the increased complexity and the need for extra parts to make it work.

2.6 - Cascade Vapour Compression Cycle

Cascade refrigeration cycles are an alternative to the single or two-stage compression refrigeration cycle. They are intended to obtain lower temperatures in the evaporator, and consequently operate with different refrigerants on each stage. Different refrigerants are used as the temperatures involved in each stage favour the thermodynamic properties of different working fluids.

Cascade refrigeration system use two different refrigerants and two different cycles. The HP (high pressure) cycle and the LP (low pressure) cycle are connected by a heat exchanger. This heat exchanger acts as the evaporator in the HP side and as the condenser on the LP side.

Figure 8 presents a p-h diagram of a conventional cascade refrigeration system.

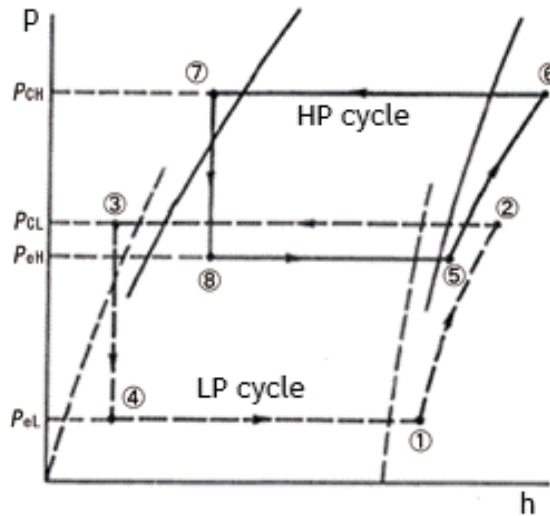


Figure 8 - p-h diagram of a cascade refrigeration system [9].

As the p-h diagram suggests the condensation temperature of the LP cycle is higher than the evaporation temperature of the HP cycle, which ensures the heat transfer between cycles. In Figure 9 it is presented a typical cascade system layout:

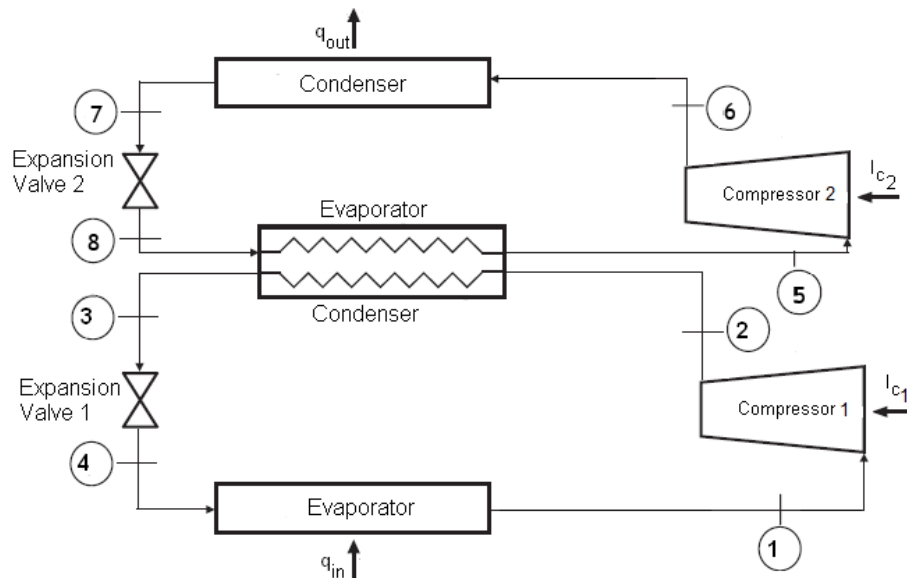


Figure 9 - Layout of a cascade refrigeration system [10].

2.7 - Transcritical Vapour Compression Cycles

The standard transcritical refrigeration cycle operates just like any other vapour compression cycle.

Every fluid has a point called “Critical Point”. In a phase diagram, the critical point or critical state is the point at which two phases of a substance initially become indistinguishable from one another. The critical point is the end point of a phase equilibrium curve, defined by a critical pressure P_c and critical temperature T_c . At this point, there is no phase boundary [11].

For a fluid to be in a transcritical state, its pressure must be above the pressure of the critical point. In the case of CO_2 , the critical pressure and temperature are 7,38 MPa and 31,1°C, respectively [7]. Figure 10 presents a phase diagram with saturated liquid and vapour lines and the respectively critical point.

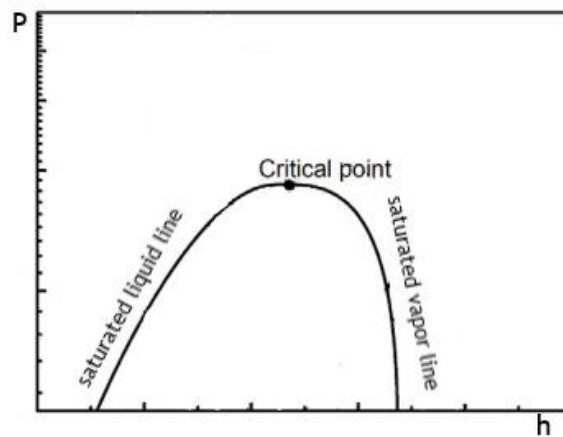


Figure 10 - Critical point position (adapted from [12]).

In the transcritical cycle, most of the time the heat rejection takes place above the critical point. In the supercritical region, condensation does not occur, and the pressure is independent of the temperature. So, the condenser in the conventional vapour compression cycle is replaced by a gas cooling device called “gas cooler”. Nevertheless, the heat absorption occurs by evaporation of the refrigerant at low pressure similarly to the conventional subcritical cycle. Figure 11 displays the basic transcritical cycle [12].

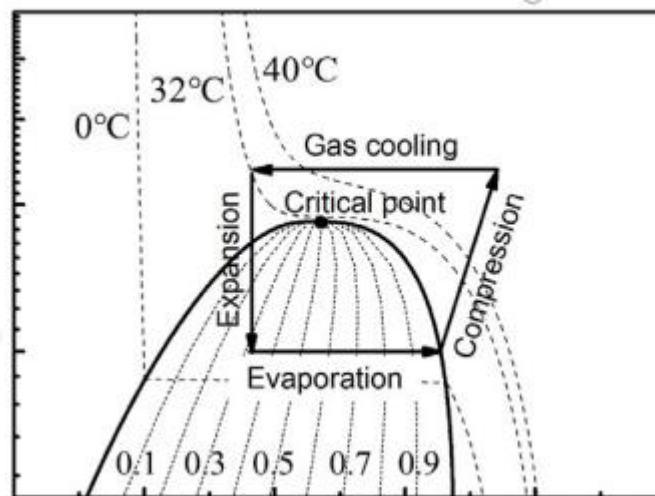


Figure 11 - p-h diagram of a transcritical refrigeration cycle (adapted from [12]).

2.8 – Two-Stage Evaporation Cycle (Booster System)

A two-stage evaporation cycle is a refrigeration system that has two or more evaporators at different temperatures. They can provide different spaces with different temperatures and are called “booster CO₂” when the refrigerant used is the R744 (CO₂). Figure 12 displays a typical layout of a booster refrigeration system:

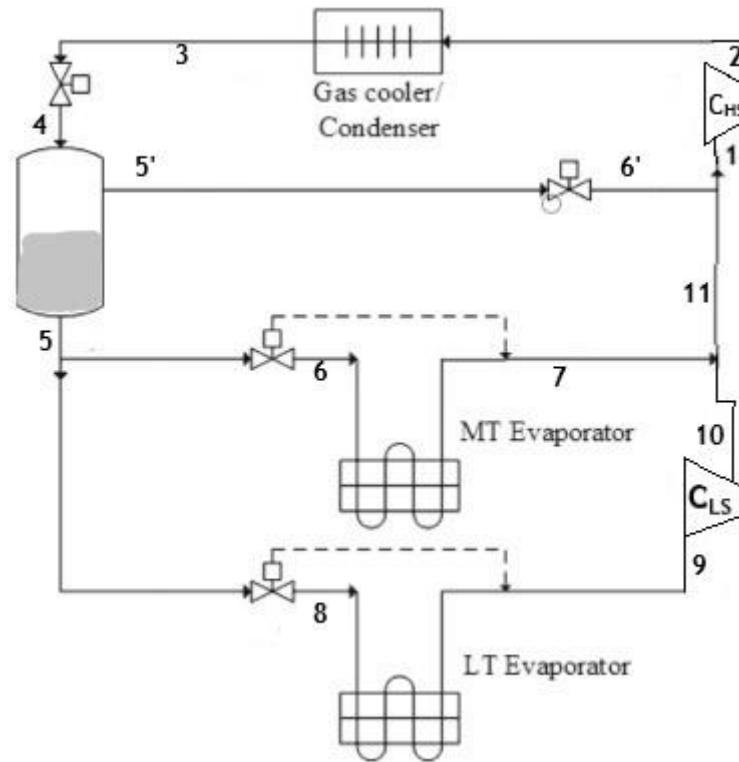


Figure 12 - Booster refrigeration system (adapted from [13]).

Two-stage evaporation cycles are mostly used in commercial refrigeration systems (supermarkets). One or a set of evaporators is used in the freezing area, also called low-temperature or negatives (LT) evaporators and the other one/set is used in the cold area, also called medium-temperature or positives (MT) evaporators. Figure 13 displays a p-h diagram of a booster refrigeration system.

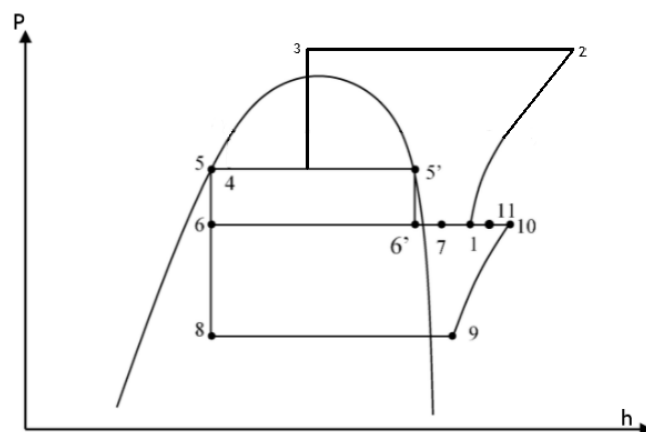


Figure 13 - p-h diagram of a booster refrigeration cycle (adapted from [14]).

This type of refrigeration systems has two different compressor sets. One of them is used to compress the vapour from the LT evaporators up to the pressure of the exit of the MT evaporators (LT compressors). The other set (MT compressors) compresses a mixture of the vapour from the LT compressors (state 10 in Figure 13), from the MT evaporators (state 7 in Figure 13) and the two-phase fluid from the expansion of the liquid receiver gas (state 6' in Figure 13).

2.9 – Booster System with Parallel Compression

Much like the standard booster system, the parallel compression one has two sets of evaporators and two main sets of compressors. In addition to those, this configuration of the system includes an extra compressor called auxiliary compressor, also called parallel compressor.

Figure 14 presents a layout of a booster refrigeration system with parallel compression.

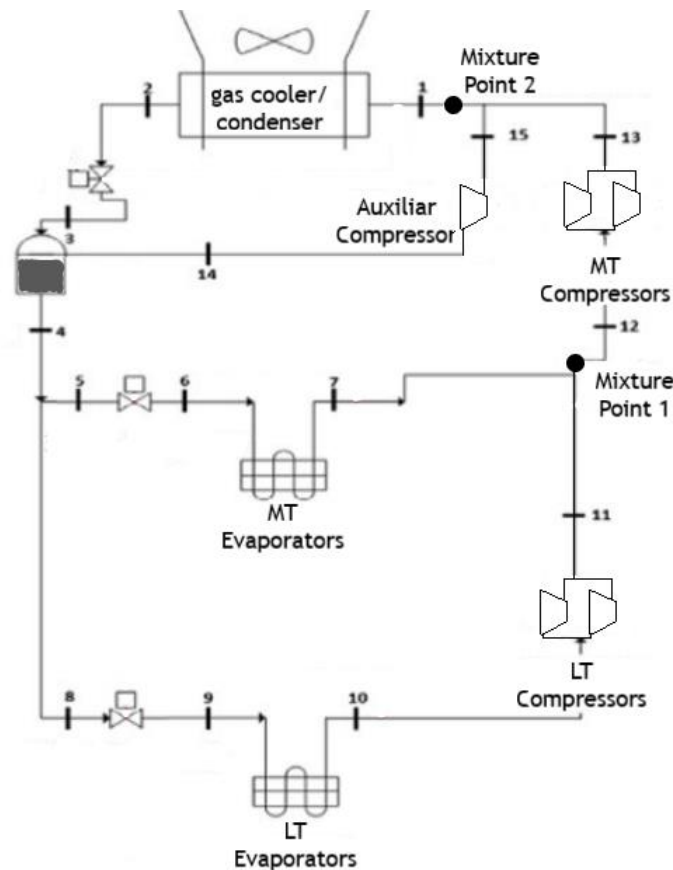


Figure 14 - Booster refrigeration system with parallel compression (adapted from [15]).

The main difference between the parallel compression and the standard booster refrigeration system is the replacement of the flash gas valve with an auxiliary compressor. This auxiliary compressor is used to control the pressure in the liquid receiver by compressing the unused vapour from the tank.

This type of system has some advantages. First, it reduces the amount of vapour to be compressed in the MT compressors, which means they can be of lower power and dimensions. Still, the main advantage is the overall energy savings, increasing subsequently the COP of the system. This occurs because of the pressure difference between the liquid receiver and the condensation pressure, which is less than the pressure difference between the MT evaporator and the condensation one. Figure 15 displays a p-h diagram of this configuration.

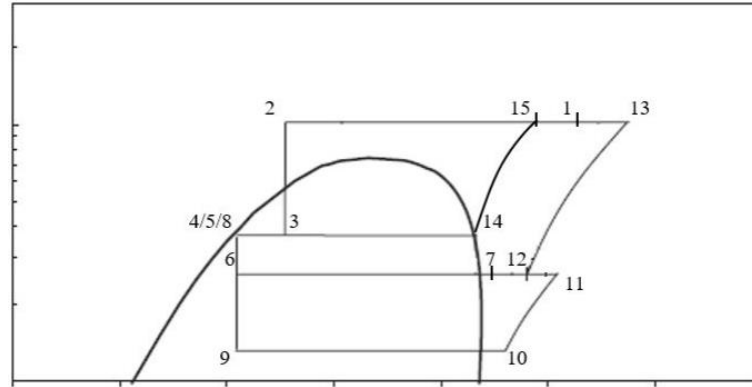


Figure 15 - p-h diagram of a booster refrigeration cycle with parallel compression [15].

Chapter 3

3 Problem Definition

At this stage, it is important to define the design conditions of the proposed transcritical CO₂ refrigeration system. As discussed before due to F-GAS regulations and environmental concerns, the system must be based on a transcritical CO₂ cycle in order to reduce its environmental impact. Another requirement is integrating dual-stage compressors on a parallel compression booster system. Parallel compression is a necessity since the refrigeration unit is supposed to be installed in Portugal, which has a warm climate compared to the northern Europe. As project conditions, it is also considered an outside temperature of 38°C. According to [3], the temperature difference between the outside temperature and gas cooler outlet temperature should be about 2 K.

The refrigeration unit must provide the refrigerant flow and the cooling effect into two different sets of evaporators at different temperatures. These different evaporation temperatures are called Positives or MT and Negatives or LT. The Positives, MT, are the evaporators at higher temperature and the Negatives, LT, are the lower temperature ones. This refrigeration unit has different cooling load requirements for each set of evaporators, the MT requires 100 kW and the LT 30 kW.

The evaporation temperatures are also pre-determined, according to the intended applications. The MT evaporators require a temperature of -10°C and the LT requires -32°C. To prevent liquid carryover towards the compressors a 5 K of useful superheating is ascribed to the working fluid at the outlet of the evaporators.

The non-useful superheating is also a parameter to consider. This phenomenon occurs on refrigerant pipes that carry the fluid to and from the evaporators. Pipe length and surroundings conditions depend on the physical space that is different from the one where the refrigeration unit will operate. However, it is possible to predict, that the non-useful superheating will be between 2 K and 5 K [3]. Since it is not constant, the refrigeration unit should have mechanisms to allow it to maintain operation over the entire range of non-useful superheating.

At this point it is possible to establish the main project conditions. It is then requested, considering the following data on Table 2, to project a refrigeration unit and selecting the required components.

Table 2 – Main project conditions.

Parameter	Value / Parameter
Location	Portugal
Working fluid	CO ₂ / R744
LT Cooling load	30 kW
MT Cooling load	100 kW
LT temperature	-32°C
MT temperature	-10°C
Useful superheating	5 K
Gas cooler outlet temperature	41°C

Dimensioning the gas cooler is also outside the scope of this project. This is because refrigeration units can be equipped with a heat exchanger in series with the gas cooler that provides DHW – Domestic Hot Water for the building in which it is installed, improving the global efficiency of the system. Even though this heat exchanger is usually installed before the gas cooler and inside the frame of the unit, it will not be considered in this project.

Project conditions define the maximum cooling load in extreme outside conditions. Nonetheless, projecting a refrigeration unit for this exact condition is not good practice, as outside climate conditions change over the course of days and months. Cooling load requirements also change, depending on usage and outside temperature. Therefore, up next, it is described a total of 12 different conditions and cooling requirements that the refrigeration unit must be able to operate at and provide. Table 3 presents all the operating modes of the refrigeration unit.

Table 3 – Extra operating modes.

Colling load	Gas cooler outlet temperature		
MT= 100 kW and LT= 30 kW	41°C	33°C	15°C
MT= 0 kW and LT= 30 kW			
MT= 100 kW and LT= 0 kW			
MT= 50 kW and LT= 15 kW			

These extra operating modes are as follows: the refrigeration unit must provide the full cooling load with three different gas cooler outlet temperatures. It needs to be capable of turning off either one of the evaporations stages and maintain the other at full capacity for those same gas cooler outlet temperatures. Finally, it must be able to operate at 50% cooling capacity.

Chapter 4

4 Determined Solution

4.1 – Refrigeration Unit Diagram

The adopted solution was engineered to cope with project conditions, as well as the determined extra operating modes, and allows for a complete separation of evaporation stages.

In Figure 16 it is possible to observe the refrigeration unit diagram, which includes all the components necessary for its operation.

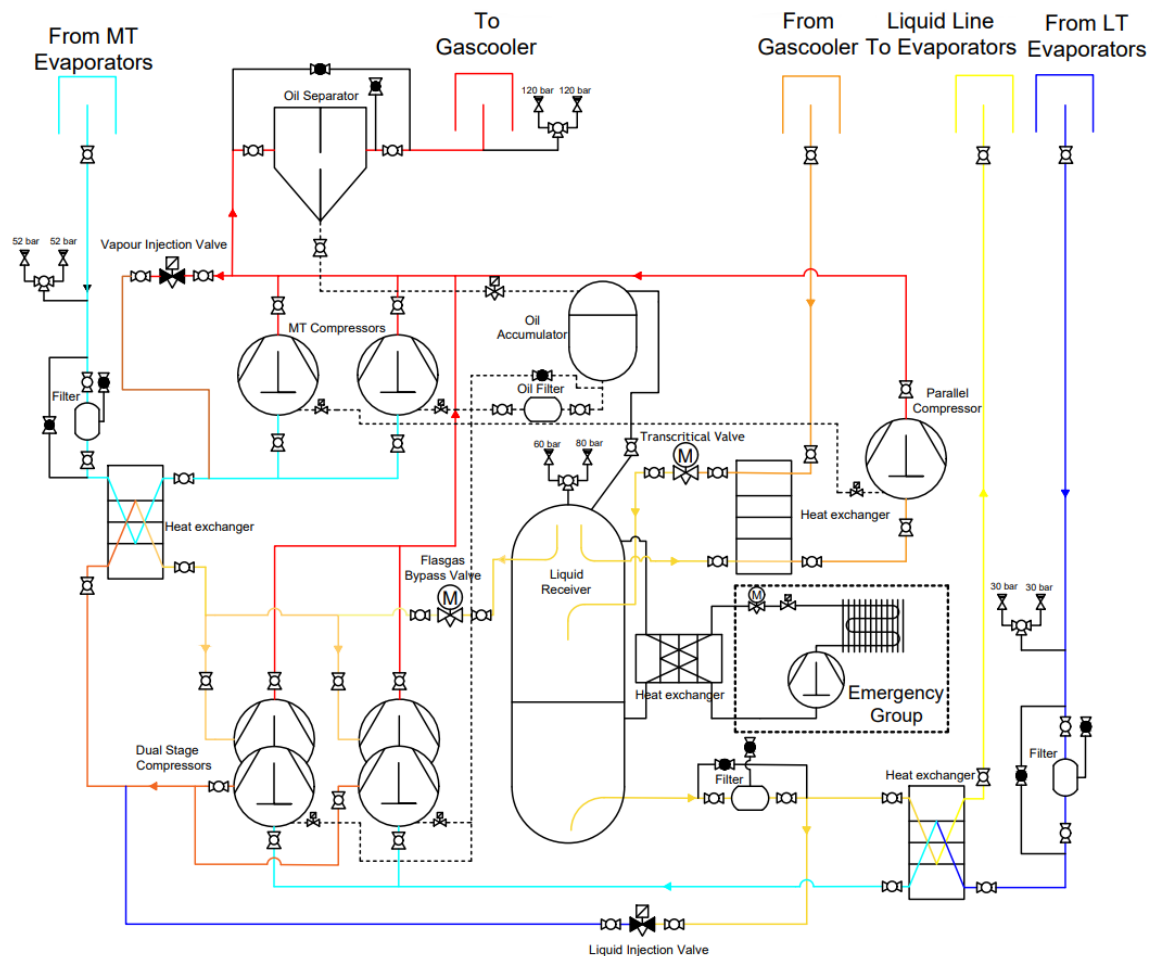


Figure 16 - Refrigeration unit diagram.

As is seen in Figure 16, this solution employs more compressors on each compression stage than the systems discussed previously. In fact, this refrigeration unit integrates components that are not mentioned in chapter 2. Among them are internal heat exchangers, injection valves, filters, and the entire oil system, which are critical to the correct operation of the refrigeration unit.

Another visible component is the safety valves. They serve as automatic pressure relief to prevent the system pressure from rising, which is all the very important because of the high expansion coefficient of the R744. By preventing the pressure from rising above a certain value, these valves guarantee the protection of equipment and the people around it. They are placed at strategic points in the refrigeration unit so as not to trap the CO₂ and protect the pressure limits of the equipment and the installation itself. In this case they are placed at the inlet of the compressors in the liquid receiver and after the oil separator. As one can observe on the diagram and in Figure 16, these valves are installed in pairs to ensure redundancy in the system. Figure 17 presents the safety valves.



Figure 17 - Safety valves [16].

Cut-off valves are also visible on the diagram. These are located before and after each component, which allows for the component to be completely isolated from the unit. They impede refrigerant to flow in certain operations, such as maintenance procedures or replacing components.

Bypass valves are also visible. These are open when performing filter maintenance, as it is a relatively frequent task, and allow the refrigeration unit to maintain operation during these moments.

Another visible component is the charge valves. The function of these valves is to allow refrigerant to be added or removed from the system. In addition, they allow quick and easy pressure measurement in a given section of the unit. Some of these valves have an integrated charge port, which eliminates the necessity of the charge valves as it is seen in Figure 18.

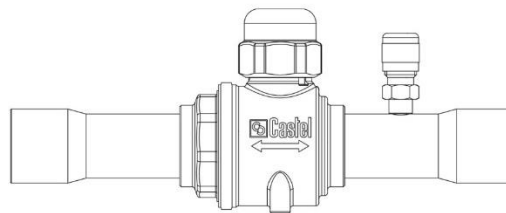


Figure 18 - Cut off valve with integrated charge port [17].

The other visible components on the refrigeration unit diagram are discussed and dimensioned in section 4.2, where it is addressed their size, function, and operation.

There are also some components that are not visible in the diagram of Figure 16. The most important is the frame. This frame is a structure of steel that is powder-coated and provides a location and mounting points to all other components. It is also equipped with lifting points, so that the unit can be moved via a crane or a forklift. Since all components are secure to the inside of the frame, it protects them in case of a small drop during transportation. Figure 19 presents a refrigeration unit frame.

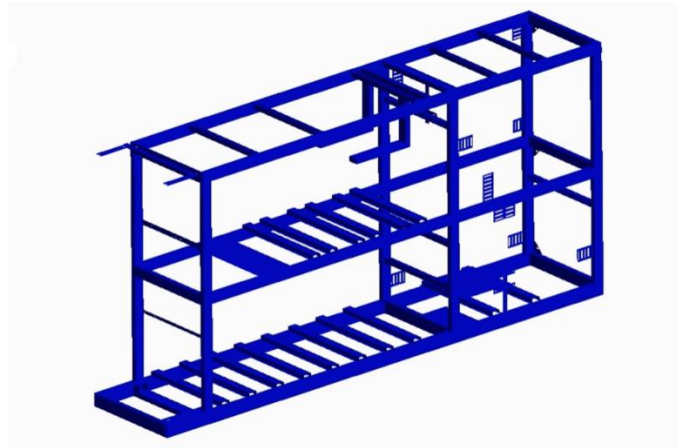


Figure 19 - Refrigeration unit frame [3].

Appendix A presents the refrigeration unit diagram, complete with all the selected components.

4.2 – Equipment Dimensioning

4.2.1 - Compressors

Any commercial or industrial refrigeration unit must comprise one or more compressors. They are one of the most important components of the refrigeration system. Their function is to compress the working fluid, changing its characteristics in both pressure and temperature. This makes it possible to transfer heat to the warm medium, and consequently absorb heat from the cold medium, creating a refrigeration cycle, as discussed on Chapter 2.

The selected compressors are from the manufacturer *Dorin Innovation* and the main compressors are CD2S range. These are dual-stage compressors that have a low pressure inlet and outlet, as well as a high pressure inlet and outlet. The low pressure outlet can be directly connected to the inlet of the high pressure side. However, as it was seen on section 2.4, this does not provide any benefits when compared with single stage compression. Therefore, the fluid is pass through a heat exchanger before it enters the high pressure side of the compressor. Figure 20 presents the CD2S compressors.



Figure 20 – Dual-stage compressor CD2S (adapted from [18]).

As per the refrigeration unit diagram on section 4.1, the final solution employs 5 compressors. Two belong to the previously mentioned range and are responsible for the refrigerant flow through the LT evaporators. The flow of refrigerant through the MT evaporators is accomplished by two other compressors. The fifth compressor is the parallel one and acts in conjunction with the flashgas bypass valve, controlling the pressure inside the liquid receiver, removing the flashgas from it and compressing it to gas cooler pressure in the case of the parallel compressor. These compressors are from the range CD-H. They are a single stage compressor able to operate in supercritical conditions and perfectly suitable for commercial refrigeration. All compressors are semi-hermetic, which allows for maintenance operations, as well as parts replacement.

To select the compressors, Dorin software [19] was used. This software was designed by the manufacturer of the compressors and hence it is the best tool to aid in their selection. It integrates a lot of different features and modes, but, in the booster calculation option, there is no option to use the CD2S range of compressors. Because of this limitation, it was decided in conjunction with the manufacturer that the best course of action, in other to integrate dual-stage compressors on a traditional booster refrigeration system, is to separate the LT and MT compression.

This means the final refrigeration unit will act like two independent and separate refrigeration units, and be capable of providing LT@-32°C and MT@-10°C cooling loads

independently. However, they will be integrated in the same refrigeration frame and share all other components with each other. This meant there was a need to run several calculations on the software, using different calculation modes to calculate the full pretended system.

Selecting the dual-stage compressors on the software is done by choosing a compressor from the catalogue, and then entering the project conditions. The software then exports the cooling capacity as well as various other parameter of the selected compressor, based on the previous conditions. The selected dual-stage compressors are a pair of CD2S2500 compressors, as these are the smallest dual-stage compressors that can ensure the required cooling load.

Figure 21 presents the selection window of the selected compressor.

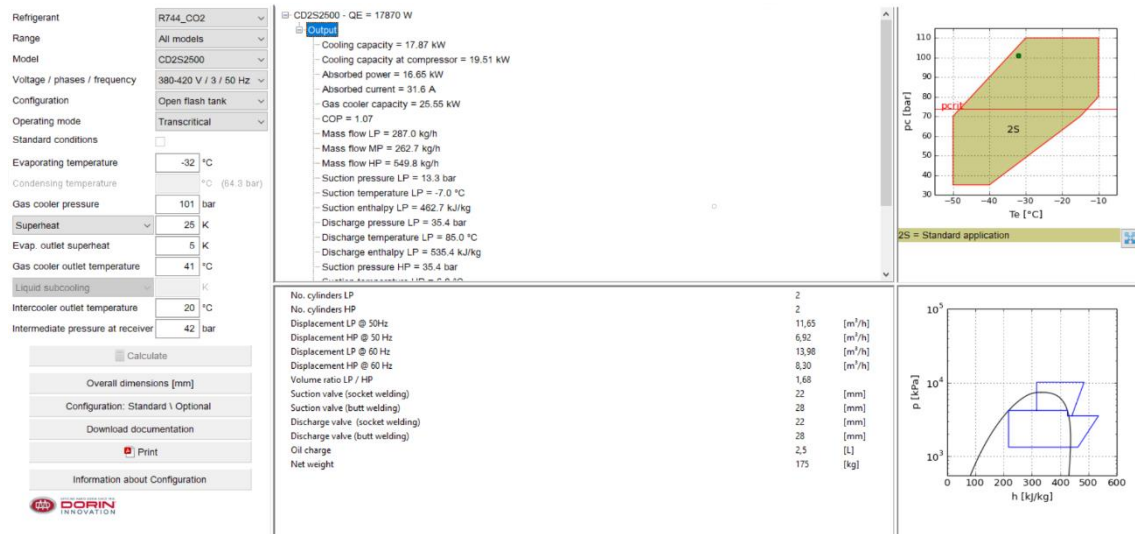


Figure 21 – Dual-stage compressor selection window [19].

As it is seen, the cooling capacity of one of the selected compressors is more than half of the LT cooling capacity. In order to maintain a correct operation, and to ensure the correct refrigerant flow at different conditions, one of these compressors will be equipped with a frequency inverter that can modulate its operating speed.

Selecting the other three compressors is done using the same software but in a different calculation mode. The calculation mode in this case is the MT transcritical system (CO₂) which allows for the selection of MT compressors in both parallel compression of flash gas by-pass configurations. According to [3], if there are just two MT compressors, one should be bigger than the other. The larger is continuously on and the smaller one is equipped with an inverter, to modulate the refrigerant flow through the evaporators, adapting the unit to different cooling requirements and conditions.

Selecting the parallel compressor was also done in the same calculation mode. This compressor should be as small as possible, in order to minimize the number of start and stops that it has to endure. Since the dual-stage compressors were selected in flash gas by-pass configuration, i.e. at full cooling load, the flash gas generated from the liquid flow rate to the LT evaporators is compressed on the high pressure side of the dual-stage compressors. As a consequence, the parallel compressor is dimensioned directly with the MT compressors.

Selecting these compressors begins with imputing the working conditions, which are the same parameters as in the selection of the dual-stage compressors as well as, the

cooling load. Afterwards, the compressors are chosen and, if they are supposed to be equipped with an inverter, that option is selected as well. Then, the software outputs the compressor reserved capacity. If this reserved capacity is negative, then the selected compressor group is too small and, if the reserved capacity is too high, then the compressors are larger than what is necessary. According to [3] a reserved capacity between 5% and 15% is adequate.

Figure 22 presents the selection window for both MT and parallel compressors.

The screenshot displays a software interface for selecting compressors. It includes input fields for voltage (380-420 V / 3 / 50 Hz DOL), inverter status, and model numbers (CD 2000H, CD 2400H, CD 3401H). A central panel lists system parameters: Refrigerant (R744_CO2), Required capacity (100 kW), Evaporating temperature (-10 °C), Suction line outlet superheat (5 K), Evaporator outlet superheat (5 K), Gas cooler outlet temperature (41 °C), Gas cooler pressure (101 bar), and Receiver pressure (42 bar). A right-hand sidebar shows a 'System' summary with values like Cooling capacity (MT) = 100.00 kW, Absorbed power = 61.62 kW, COP = 1.62, and Reserve capacity MT = 12.1%. Below this is a tree view of the selected components: Compressors, MT 1, MT 2, Parallel compressors, MT pri, @ Evaporator, @ Intermediate pressure receiver, @ Gas cooler, and Air conditioning.

Figure 22 - MT and parallel compressor selection window [19].

Various simulations were made in order to select the compressors seen in Figure 22. The final MT compressors selected are: one of CD 3401H, one of CD 2000H and the parallel compressor selected is a CD 2400H, as is seen on Figure 22. The reserved capacity is the same for both the MT and the parallel compressors and it is 12,1%, which is in the range of the values defined. Even so, this is not a unique solution, and another combination of compressors could have been chosen.

It is also important to state that, according to the manufacturer [18] their compressors requires a minimum vapour superheat of 10K, and, to prevent ice formation on the outside of the compressors, the refrigerant must be at least at 0°C on the intake of the compressor.

In Appendix B the selected compressors overall dimensions are presented. In addition, in Appendix C, the application envelopes provided by the manufacturer are shown.

4.2.2 – Refrigerant Piping

Refrigerant pipes are an important part of a refrigeration system, since they connect all components together taking the refrigerant to and from them. They also determine the size of other components, such as thermal insulation and the oil separator and, therefore, they need to be selected up next.

When selecting the piping size, it is necessary to consider the average velocity and pressure drop of the fluid, as well as the associated costs. Smaller diameter pipes have a higher average velocity passing through, but they also have a higher pressure drop.

Regarding costs, generally, smaller diameter pipes have lower associated costs. However, this does not mean they are preferable, as they imply greater pressure losses, which increases the compressor work, energy consumption and associated costs that overtakes the pipe material savings. Table 4 shows the average velocity intervals and limits on different sections of the refrigeration unit according to [3].

Table 4 - Average velocity limits [3].

Pipe / line	Average velocity [m/s]
Suction to LT compressors	3 - 12
Suction to MT compressors	5 - 15
Compressor discharge	5 - 12
High pressure expansion	2 – 4
Liquid	≤ 1
Parallel compression	12

The refrigerant piping selection is an iterative process. It begins by choosing an average velocity from Table 4 for a given section of the refrigeration unit and then calculate the corresponding inner pipe diameter using equations 10 and 11.

$$\dot{V} = A * u \quad (10)$$

$$A = \frac{D^2 * \pi}{4} \quad (11)$$

Where:

- $\dot{V}$ is the volumetric flow rate [m³/s]
- u is the average velocity value [m/s]
- A is the area of the cross section of the pipe [m²]
- D is the pipe inner diameter [m]

Next, a pipe is selected approximating its inner diameter with the calculated diameter from the equations (10) and (11). These pipes are selected from table 5 which contains the characteristics of pipes according to the DIN-EN SS standard [16]. By approximating the diameter, the average velocity is different from the one selected previously and, therefore, it must be calculated using the previous equations (10) and (11). If the calculated velocity is outside the pre-determined interval, then it is necessary to adjust the pipe size to correct the issue. If not, it is important to calculate the theoretical pressure drop of the pipe, so that becomes the next step.

According to [20], the pressure drop on a uniform horizontal pipe can be calculated using equation (12).

$$\Delta p = f \frac{l}{D} \frac{\rho u^2}{2} \quad (12)$$

Where:

- Δp is the total pressure drop [Pa]
- f is Darcy's friction coefficient
- l is the pipe length [m]
- ρ is the fluid density [kg/m³]

Pressure drop can also be calculated as a function of pipe length and in that case its units are [Pa/m].

Darcy's friction coefficient is calculated using either Colebrook equation (13) or Haaland equation (14):

$$\frac{1}{\sqrt{f}} = -2 \log_{10} \left(\frac{\varepsilon}{D} + \frac{2,51}{Re \sqrt{f}} \right) \quad (13)$$

$$\frac{1}{\sqrt{f}} = -1,8 \log_{10} \left[\left(\frac{\varepsilon/D}{3,7} \right)^{1,11} + \frac{6,9}{Re} \right] \quad (14)$$

Where:

- f is Darcy's friction coefficient
- ε is the pipe's effective roughness height [m]
- Re is the flow's Reynolds number

Colebrook's equation (13) does not allow the determination of Darcy's coefficient (f) in any other way than iteratively. Haaland's equation (14) avoids the iterative calculation and results in less than 2% difference when compared to Colebrook's results. This approximation is acceptable since an uncertainty up to 10% on the value of f is accepted [20]. So, this was the equation used.

The Reynolds number can be calculated using equation (15).

$$Re = \frac{u \cdot D}{\nu} = \frac{\rho \cdot u \cdot D}{\mu} \quad (15)$$

Where:

- ν is the fluid's Kinematic viscosity [m²/s]
- μ is the fluid's dynamic viscosity [pa*s]

An acceptable pressure drop value as a function of length is around 0,1% of the pressure at the start of the pipe. Higher values can be accepted in applications of reduced lengths.

As is seen in Table (4), most sections have a maximum and minimum velocity limit. In these cases, the velocity value to consider, when calculating the pipe diameter, should be close to the maximum. This ensures the velocity in the pipe is never lower than the minimum value, when the refrigeration unit is not operating at full capacity. On the sections that have a maximum velocity value, or only have one value, that was the value used when starting the calculation to select the pipe diameter.

The following Table 5 presents pipe dimensions of DIN-EN SS standard.

Table 5 - Refrigerant piping dimensions [16].

Pipe type	N. size [mm]	N. size [inch]	Inner diameter [m]	Outer diameter [m]	Wall thickness [mm]
DIN-EN SS 10	10	0,38	0,0126	0,0172	2,3
DIN-EN SS 15	15	0,50	0,0167	0,0213	2,3
DIN-EN SS 20	20	0,75	0,0223	0,0269	2,3
DIN-EN SS 25	25	1,00	0,0285	0,0337	2,6
DIN-EN SS 32	32	1,25	0,0372	0,0424	2,6
DIN-EN SS 40	40	1,50	0,0431	0,0483	2,6
DIN-EN SS 50	50	2,00	0,0545	0,0603	2,9

SS on the pipe standard refers to the pipe material, which, in this case, is 304L stainless steel. This material was chosen for its corrosion resistance, because of acid and impurities that exist inside the system.

The low velocity limit on the liquid line is due to the expansion valves used before the evaporators. These valves are located just before their respective evaporators, which means they are outside the refrigeration unit this thesis relates to. However, the feed pipe for these valves is part of the unit, so, it must be selected.

As a practical example of selecting a specific pipe the liquid line will be used.

According to [19] the mass flow rate and the fluid density in the liquid line is:

$$\begin{cases} \dot{m} = 2075 \text{ [kg/h]} = 0,576 \text{ [kg/s]} \\ \rho = 900 \text{ [kg/m}^3\text{]} \end{cases}$$

Using equation (16) the volume flow rate is calculated.

$$\dot{V} = \frac{\dot{m}}{\rho} = 0,00064 \text{ [m}^3\text{/s]} \quad (16)$$

Because this line only has one value for the velocity that is selected: $u=1 \text{ [m/s]}$

Combining equations (10) and (11) the calculated diameter comes to: $D= 0,0286 \text{ [m]}$, this is close to the internal diameter of the DIN-EN SS 25 pipe which is $0,0285 \text{ [m]}$. Using the previous equations results in an average velocity of $u'=1,00 \text{ [m/s]}$, which is within the allowable velocity interval. Pressure drop calculation is up next.

According to [20], the stainless steel pipe effective roughness height, ϵ , is $0,015 \cdot 10^{-3} \text{ [m]}$ and according to [21] the fluid's dynamic viscosity at the corresponding pressure and temperature, μ , is $91,3 \cdot 10^{-6} \text{ [Pa}\cdot\text{s]}$. Applying these values to equations (12) and (14) and (15), Darcy's coefficient equates to $f=0,01829$, and the pressure drop as a function of length equates to $\Delta p/m=291,2 \text{ Pa/m}$. Assuming that the liquid receiver is at 4 MPa (40 bar) the pressure loss is $0,007\%/m$, which is acceptable.

A confirmation method for these calculations was developed using Danfoss's software Coolselector2 [16]. Figure 23 displays the recommended pipe for the liquid line of the refrigeration unit. This recommendation is based on the refrigerant mass flow, as well as the cooling loads, the average velocity in the pipe and the pressure drop.

Figure 23 presents the software window of pipe selection.

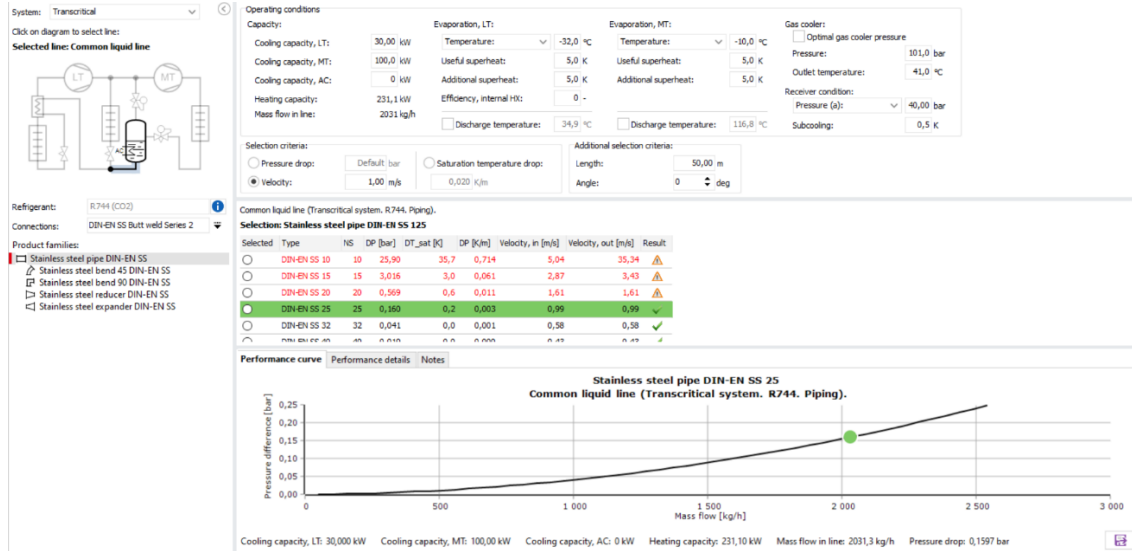


Figure 23 - Software recommended pipe size [16].

As it is seen, the calculation software also recommends the same size pipe as the manual calculations. Therefore, it is determined that the liquid line on the refrigeration unit project is DIN-EN SS 25.

All other pipes on the refrigeration unit were sized in the same manner. Firstly, a manual calculation was made, followed by a software confirmation. Table 6 presents the selected pipe size for each location of the refrigeration unit.

Table 6 - Pipe size according to location / application.

Location/application	Pipe/Line	Velocity [m/s]	Δp [Pa/m]	Δp [%/m]
Liquid line	DIN-EN SS 25	1,00	291,2	0,007
LT evaporators return line	DIN-EN SS 20	10,24	1433,0	0,108
DSC* LT discharge to heat exchanger	DIN-EN SS 15	11,19	3967,0	0,170
DSC* MT suction from heat exchanger	DIN-EN SS 15	12,17	4300,6	0,185
MT evaporators return line	DIN-EN SS 25	10,40	2189,4	0,083
Flash gas/parallel compressor suction	DIN-EN SS 20	11,26	6170,9	0,154
Parallel compressor discharge	DIN-EN SS 15	12,05	16802,2	0,166
To Gas Cooler	DIN-EN SS 25	10,49	5454,5	0,054
From Gas Cooler to transcritical valve	DIN-EN SS 25	2,84	1492,7	0,015
DSC* LT individual suction line	DIN-EN SS 15	9,13	1634,6	0,123
DSC* LT individual discharge line	DIN-EN SS 10	9,83	4360,6	0,187
DSC* MT individual suction line	DIN-EN SS 10	10,69	4779,9	0,205
DSC* MT individual discharge line	DIN-EN SS 10	3,21	1433,9	0,014
MT compressors individual suction line	DIN-EN SS 20	8,49	1981,7	0,020
MT compressors individual discharge line	DIN-EN SS 15	6,76	4016,1	0,039
Flash gas bypass line	DIN-EN SS 15	-	-	-
Liquid injection line	DIN-EN SS 10	-	-	-
Vapour injection line	DIN-EN SS 10	-	-	-

*DSC: Dual-stage compressor.

4.2.3 – Transcritical Valve

The transcritical valve is a high-pressure expansion valve located between the gas cooler and the liquid receiver. Its job is to provide a first reduction in pressure from around 100 bar (10 MPa) in the gas cooler, to 40 bar (4 MPa) in the liquid receiver. With this expansion, it is possible to obtain humid vapour that will reach the liquid receiver.

Choosing this component can be done by using catalogues or the software from the manufacturer. The valve family chosen for this application is the Danfoss CCMT valves. These are an electric operated valve designed specifically for operation in CO₂ systems.

Figure 24 shows the manufacturer software, where the transcritical valve was dimensioned [16]. Input parameters of the program are the cooling loads, in both evaporators' stages, the temperature in those stages, the degrees of useful superheating, the pressure in the liquid receiver, as well as the gas cooler outlet temperature and pressure. The output of the software is the mass flow through the transcritical valve.

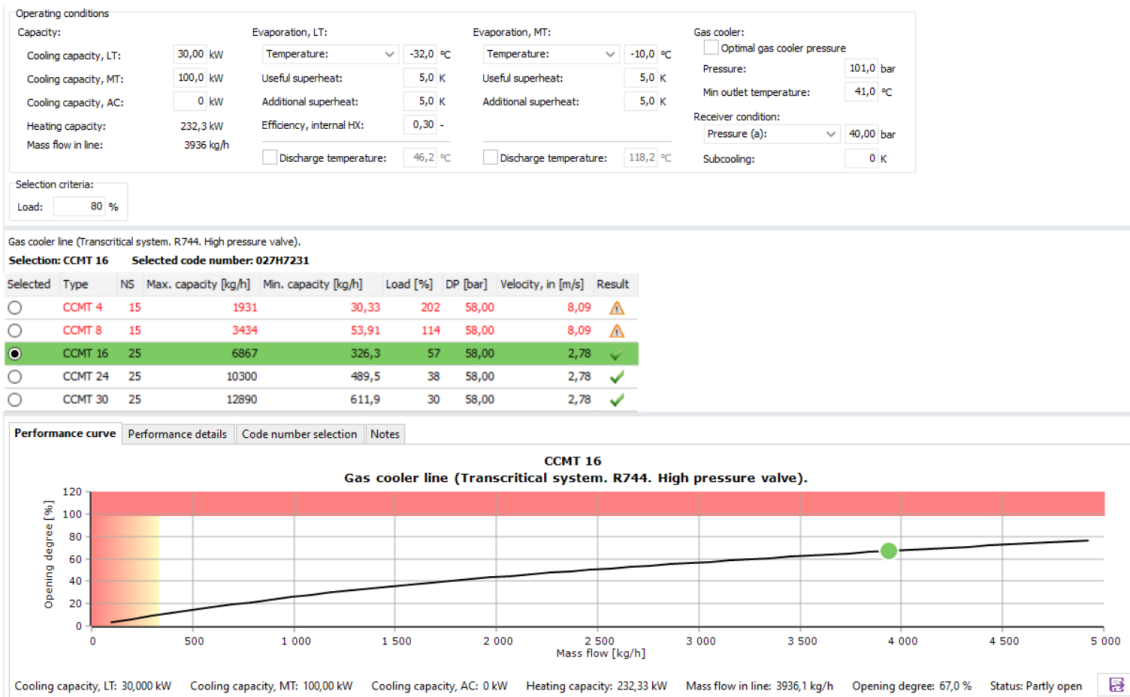


Figure 24 - Transcritical valve selection using Coolselector2 software [16].

The software selects the CCMT 16 valve. The system mass flow rate is between both the maximum and minimum values and the velocity of the fluid on the valve is between the limits mentioned in section 4.2.2.

The connection on this valve is also favourable, because, as shown in Figure 25, they match the dimensions of the pipes selected for the gas cooler outlet into the transcritical valve.

System diagram	
System details	
CCMT 16	
	Maximum operating temperature [°C]= 60,0
	Minimum operating temperature [°C]= -40,0
	Opening degree [%]= 67,75
	Choked= False
	Valve state= Open
	Capacity [%]= 56,87
	Maximum capacity [kg/h]= 6867
	Nominal size inlet [mm]= 25,00
	Nominal size inlet [inch]= 1,00
	Inlet diameter [mm]= 28,50
	Nominal size outlet [mm]= 25,00
	Nominal size outlet [inch]= 1,00
	Outlet diameter [mm]= 28,50
	Available connections:
	DIN-EN Butt weld. Size: 25 t=2,6 mm
	DIN-EN SS butt weld. Size: 25 t=2,6 mm
	ANSI soldering ODF. Size: 1 1/8"

Figure 25 - Transcritical valve connection size (adapted from [16]).

The motorized aspect of this valve makes it perfect for this application, as it is able to precisely control the mass flow rate required by the system as well as, being able to adapt to different operating conditions of the refrigeration unit.

4.2.4 – Liquid Receiver

The liquid receiver is a tank that receives the humid vapour from the transcritical valve and then separates it into saturated liquid and saturated vapour. Both phases of the working fluid are stored in the tank: the liquid at the bottom and the vapour on top, due to the higher density of the liquid. The liquid is then redirected to the evaporators, creating the cooling effect. The vapour is compressed to the pressure of the gas cooler. This tank is also the refrigerant reservoir at times when the refrigeration system requires less operating flow.

One of the most important characteristics to define, when it comes to liquid receivers, is the operating pressure, because it determines the saturation temperature. In this project, the operating pressure is set at 40 bar (4 MPa), which corresponds to a saturation temperature of 5,3°C. This temperature is important as it needs to be high enough to prevent the formation of ice on the outside of the receiver, due to condensation on the surface, and be low enough to not exceed the maximum pressure of the receiver and activate the safety valve.

The liquid receiver is a pressure vessel, this means that it falls under the EU Pressure Equipment Directive (PED). This directive 2014/68/EU (formerly 97/23/EC) of the EU sets out the standards for the design and fabrication of pressure equipment ("pressure equipment" means steam boilers, pressure vessels, piping, safety valves and other components and assemblies subject to pressure loading) generally over one litre in volume and having a maximum pressure of more than 0,5 bar relative. It also sets the administrative procedures requirements for the "conformity assessment" of pressure equipment for the free placing on the European market, without local legislative barriers [22].

As a consequence of these directives, the liquid receiver must be CE certified, just like all other components, for the pressures and temperatures reached in the system. In the receiver, especially with CO₂, due to its high thermal expansion coefficient, a safety valve must be installed so as not to trap the fluid in the event of a pressure increase. The pressure limit set by the safety valve will be 60 bar (6 MPa), which allows the system to function normally and protects the equipment.

It is usual on liquid receivers to have a secondary safety valve set to 80 bar (8 MPa). During maintenance operations, when changing filter for example, all the connections to the receiver are closed. Maintenance operations are brief. However, the pressures inside the receiver usually rise above the 60 bar safety valve. For this reason, during these operations, the 60 bar valve is capped off, and the 80 bar safety valve sets the pressure limit for the receiver. The 80 bar limit allows operators more time to perform operations without risking releasing refrigerant to the atmosphere.

Liquid receivers exist in several different sizes, mounting positions, volumes, connection types and diameters. One supplier for these devices is *TECNAC*, which has a catalogue of CO₂ specific devices. For packaging reasons, a vertical receiver is best suited to the application and, according to [3], the liquid receiver volume should be between the following interval:

$$\dot{Q}_L [KW] * 1.3 < V [l] < \dot{Q}_L [KW] * 1.4 \quad (17)$$

The combined cooling load of the system is 130 KW. Therefore, according to equation (17), the volume of the receiver should be between 169 l and 182 l. The only receiver *TECNAC* has in this size is the RV180 which is 180 l. Moreover, *TECNAC* receivers have three different maximum pressure ratings of 60, 90 and 130 bar. Because the secondary safety valve is set to 80 bar, the receiver should have a maximum pressure higher than that. A 90 bar pressure was selected as the maximum pressure. The receiver selected is then the RV180 – 90bar – TR. Figure 26 presents the selected liquid receiver.



Figure 26 - Liquid receiver RV180 - 90bar - TR adapted from [23].

For this receiver L equals to 1646 mm and $\varnothing$ to 406 mm.

4.2.5 - Filters

Impurities, contaminants, and debris in a refrigeration system can affect its normal function, therefore, it is necessary to prevent them from free flowing in the system. These contaminants appear from poor cleaning at the time of building and installing the components, bad quality of refrigerants and oils, and decomposing oil in the system. Some of the effects of the impurities are as follows:

- **Solid contaminants:** cause damage and reduce compressor efficiency, obstruct the expansion valves, and accelerate the process of oil and refrigerant decomposition;

- **Organic contaminants:** obstruct the expansion valve orifice;

- **Non-condensable gases:** chemically reactive gases "attack" the various components of the system and may cause them to fail. Inert gases reduce the efficiency of condensation and accelerate chemical reactions, such as oil decomposition;

- **Acids:** are corrosive, mainly in the presence of water, damaging the metallic materials of the system. They also act as catalysts for chemical reactions and can cause a short-circuit in the compressor motor;

- **Moisture:** The presence of water in the system leads to the formation of ice on the expansion valves, promotes the appearance of acids from the oil and the refrigerant and can even burn the compressor motor [24].

The function of the filters is to prevent all the above effects from happening. In this refrigeration unit there will be installed three filters: one on each return line, from the LT and MT evaporators, and one on the feed line. This feed line is the liquid line that exits from the liquid receiver to the evaporators. There will be another filter, however, this one is an oil filter. It goes on the oiling system of the refrigeration unit and protects all the components that are lubricated.

The chosen manufacturer for the filters is *Carly Refrigeration Components Solutions*. They have a vast array of filters and other refrigeration components, as well as filters with replaceable cores. This is a huge advantage, because, as said before, all the connections on the unit are welded, which makes filter replacement impossible without cutting and rewelding a new one. Figure 27 presents the BCY filter from Carly and a cut out of the same filter.



Figure 27 - Carly BCY filters [25].

Replaceable cores also mean the usage of different filter types and materials. This is useful because the refrigeration unit is a closed system. Therefore, the first filters to be used have to filter most impurities collected in the equipment, during manufacturing and assembling. Afterwards, a finer mesh filter can be applied. This one captures the smallest contaminates in the unit. Both these filters are for temporary use only as they become saturated in little time, so, afterwards, a different type of filter is applied. This one is a permanent filter, which means it takes up to several month to reach saturation.

Figure 28 presents three different Carly CCY replaceable cores.



Figure 28 - Carly replaceable filter cores [25].

For this refrigeration unit, the selected filters for the return lines of the evaporators are from the range BCY. The first core is CCY N, which is a 50 micron cleaning core, composed of chemical agents and its main features are burnout decontamination, reinforced drying, reinforced acid neutralization and wax and resin binding. After 5 days of running, the CCY N can be replaced by the CCY F, which is a 10 microns felt filter used for reinforced filtration for the next 5 days of the unit functioning.

Afterwards, on the suction lines it is applied the CCY I core. This is a 140 micron stainless steel mesh cloth and screen filter, that can be used in a permanent faction, until saturation on these lines.

On the liquid line, there are only two filter cores to be applied. The first one, like in the suction lines, is the CCY N, which is replaced after 5 days by the PLATINIUM core. The later is also a 50 micron chemical agent that has optimum acid absorption and drying capabilities. Table 7 resumes filter cores used by the refrigeration unit.

Table 7 - Filter core applications.

Filter location	First 5 days	Following 5 days	Until saturation
Liquid line	CCY N	PLATINIUM	
MT return line	CCY N	CCY F	CCY I
LT return line	CCY N	CCY F	CCY I

4.2.6 – Heat Exchangers

Heat exchangers are used to transfer heat from one medium to another. These media may be a gas, liquid, or a combination of both. The media may be separated by a solid wall, to prevent mixing, or may be in direct contact. Heat exchangers can improve a system's energy efficiency by transferring heat from places where it is not needed to useful places.

The fluids within heat exchangers typically flow rapidly to facilitate the transfer of heat through forced convection. This rapid flow results in pressure losses in the fluids. The efficiency of heat exchangers refers to how well they transfer heat relative to the pressure loss they incur. Modern heat exchanger technology minimizes pressure losses, while maximizing heat transfer and meeting other design goals like withstanding high fluid pressures, resisting fouling and corrosion, and allowing for cleaning and repairs [26].

There are different types of heat exchangers, such as shell and tube, twisted tubes, tube inserts protruding, helical baffles and other. Nowadays, the most common and most efficient heat exchangers are the plate heat exchangers, also called plate and frame. Figure 29 presents an exploded view of a plate heat exchanger.

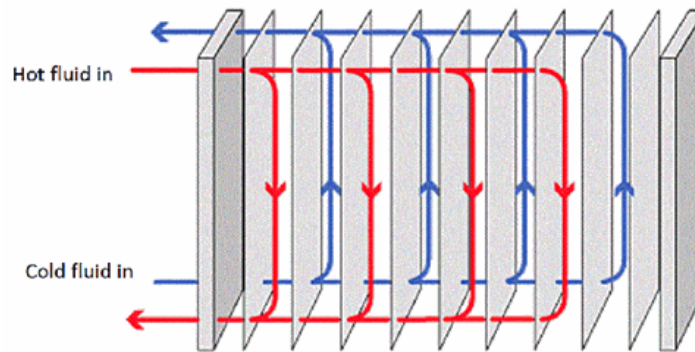


Figure 29 - Plate heat exchanger [26].

These heat exchangers are made from thin parallel plates, which contain a pattern of corrugations, and are stacked together to create a block. This has a major advantage over conventional heat exchangers in that the fluids are exposed to a much larger surface area and the corrugations promote turbulence within the fluid flow. This facilitates the transfer of heat and subsequently improves the transfer effectiveness. The most common design is to have the plates pressed against each other with gaskets in between. Such a design allows them to be opened and cleaned very easily, which will be appreciable in situations where high fouling is expected or if regular cleaning is required for hygiene reasons (food industry). Using gaskets will limit the operating pressure and temperature, so other designs with welded plates exist (braze plate heat exchangers). They can operate in more extreme conditions but cannot be open for inspection and cleaning [27].

This refrigeration unit employs four heat exchangers: three of them operate constantly and the last one is connected to the emergency group, only used on those occasions.

The first heat exchanger (heat exchanger 1) is located between the liquid line leaving the receiver and the return line from the LT evaporators. Its function is to raise

the temperature of the vapour from the evaporators up to 0°C before it enters the compressors, as well as, subcooling the liquid to prevent flashing from occurring in the line.

The second heat exchanger (heat exchanger 2) operates between the return line from the MT evaporators and the intermediate pressure of the dual-stage compressors. This heat exchanger main function is to cool the vapour between stages to take advantage of stage compression, as discussed in section 2.4. It also provides the required minimum vapour superheat to the MT compressors.

The third heat exchanger (heat exchanger 3) is located between the suction side of the parallel compressor and the return line from the gas cooler. Its function is to provide minimum required vapour superheat to the parallel compressor.

Determining the size and number of plates for each heat exchanger is an iterative process, which involves precise component dimensions that are not standard across different manufacturers. It is possible to calculate an approximate number of plates for the heat exchanger, however, in this case, the values of flow rate, pressure and temperature were supplied to the manufacturer. The manufacturer uses proprietary software, as well as its empiric knowledge to determine the size and number of plates of the heat exchangers.

One supplier of heat exchangers is Alfa Laval. They are an internationally recognized company with large experience and one of the most efficient technologies on the market. They were the manufacturer selected for the task of dimensioning the first three heat exchangers. The fourth heat exchanger is the one connected to the emergency group, and it is supplied with it. The model is the B4TH-20 from the manufacturer SWEP. Figure 30 presents an Alfa Laval heat exchanger.

Table 8 displays the heat exchangers dimensions and number of plates.

Table 8 - Heat exchangers model and size.

Heat exchanger	Model	Number of plates
Heat exchanger 1	CBXP52-20M-F	20
Heat exchanger 2	CBXP52-20M-F	20
Heat exchanger 3	AXP27-14L-F	14
Heat exchanger 4 (emergency)	B4TH-20	20



Figure 30 - Alfa Laval heat exchanger.

In appendix D there is the complete information about the manufacturers technical specifications of the heat exchangers.

4.2.7 – Flashgas Bypass Valve

The flashgas bypass valve has the same function as the parallel compressor, controlling the pressure inside the liquid receiver. However, as it was discussed in section 4.2.1 compressors have a minimum and maximum flow rate for each condition.

The flashgas bypass valve decompresses the vapour of the liquid receiver and injects it in between stages on the dual-stage compressors. This valve is an aid to the parallel compressor as it is used only at times when the flashgas flow rate necessary to remove from the liquid receiver is less or more than what the parallel compressor can handle.

The flashgas bypass valve at maximum capacity has to allow a mass flow rate of 500 kg/h of vapour to pass through from the liquid receiver to the intermediate pressure of the dual-stage compressors. As per the other valves on the refrigeration unit, to select the flashgas bypass valve, the Coolselector2 software was used. Figure 31 displays the software interface.

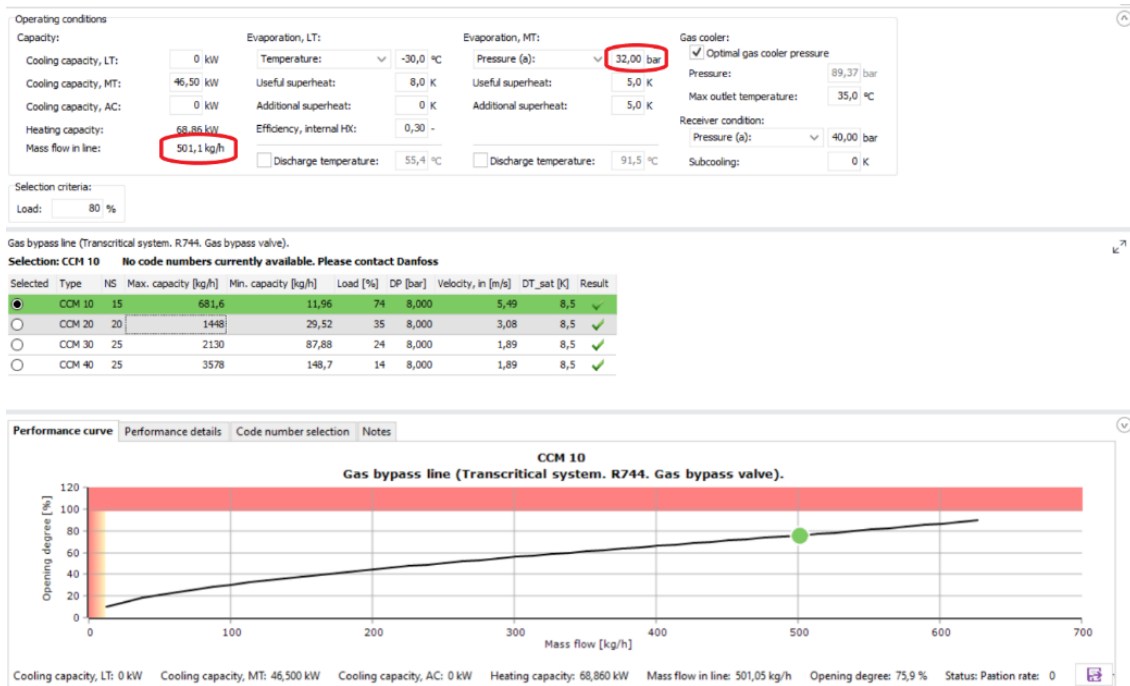


Figure 31 - Flashgas bypass valve selection (adapted from [16]).

As it is seen, the software input parameters do not correspond to the refrigeration unit's. This is done in order to obtain the necessary amounts of flashgas, cooling capacity, and MT suction pressure to match the dual-stage compressors intermediate pressure, when working with flashgas compression.

The family of valves selected is the CCM. They are electrically operated valves designed specifically for operation in CO₂ systems. They are one of two families of valves recommended for this function that have the desired connection type and are less expensive than the CCMT valves, as they do not handle as much pressure and temperature.

The selected valve is the CCM 10 as it is the smallest one that can ensure the required refrigerant flow, as well as the medium temperature and pressure. Figure 32 presents the valve, connection type, as well as other characteristics of the selected device.

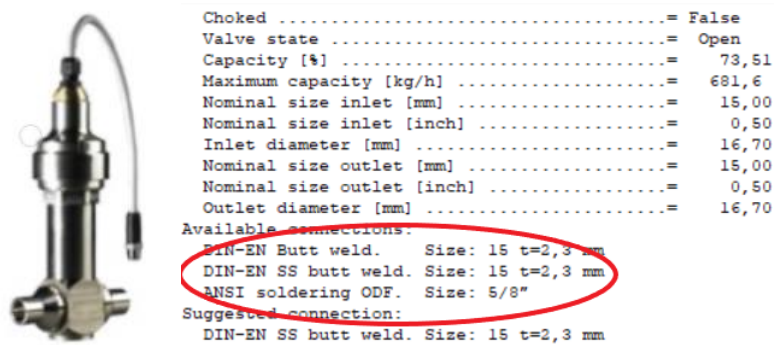


Figure 32 - Flashgas bypass valve characteristics (adapted from [16]).

4.2.8 – Liquid Injection Valve

The liquid injection valve is an expansion valve located between the liquid receiver and the first stage compressor outlet in the two stage compressors. Its function is the same as the heat exchanger number 2 (section 4.2.6), which is cooling the refrigerant between stages in the 2 stage compressors. It cools the refrigerant by expanding liquid from the liquid receiver into the low-pressure discharge line of the dual-stage compressors. This valve activates when the refrigerant flow on the MT return line is not enough to create the necessary cooling between stages.

To select the valve, it is necessary to calculate the cooling load required. The cooling load is calculated assuming the valve replaces the heat exchanger completely. As discussed on the previous section, the refrigerant intake temperature on the high-pressure side of the compressor is 20°C. According to the *Dorin software* [19], in intercooler mode, the first stage discharge enthalpy (LT) is 528,0 kJ/kg (69.1°C), and the second stage intake enthalpy (HP) is 477,4 kJ/kg (20°C), as can be represented in Figure 33.

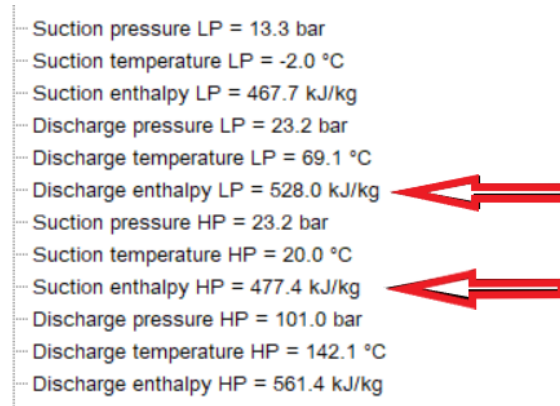


Figure 33 - Refrigerant characteristics between compression stages (adapted from [19]).

Combining the enthalpy difference and the refrigerant flow on the low pressure side of the compressors $\dot{m}_{LP}$, the cooling load is calculated using equation (18):

$$\dot{Q}_{Liquid\ Injection} = \frac{(h_{LP} - h_{HP}) * \dot{m}_{LP}}{3600} [kW] \quad (18)$$

Which results in 6,68 kW assuming the maximum refrigerant flow on the low-pressure side of the system (475 kg/h). As in the selection of the high pressure transcritical valve, the coolselector2 software was used.

As it is observed in Figure 34, the cooling load, as well as intake and discharge pressures were used as input parameters.

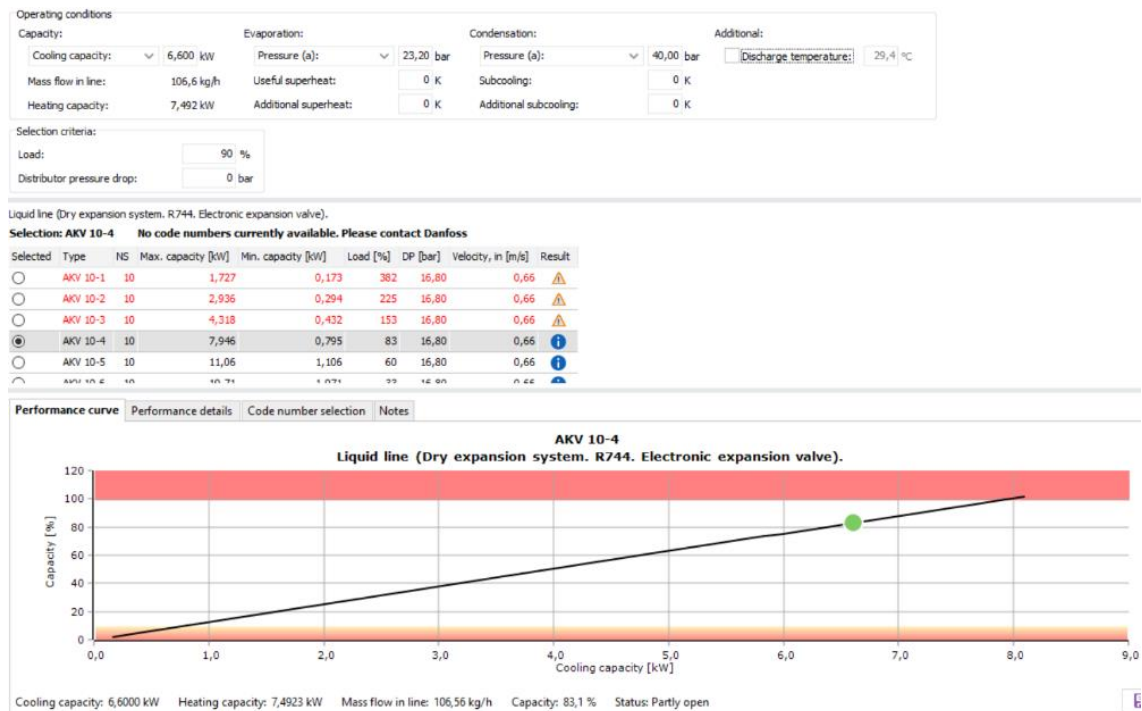


Figure 34 - Liquid injection valve selection (adapted from [16]).

The selected valve is from the AKV family of valves. They are a normally closed pulse width modulated type valve. This allows for precision control of the refrigerant flow, and, therefore, the cooling load, maintaining a constant temperature on the intake of the high-pressure side of the two stage compressors. The selected valve is the AKV 10-4, which is the smallest valve that can ensure the cooling load necessary for the system to function. Figure 35 presents the connections type, as well as other characteristics of the selected valve.

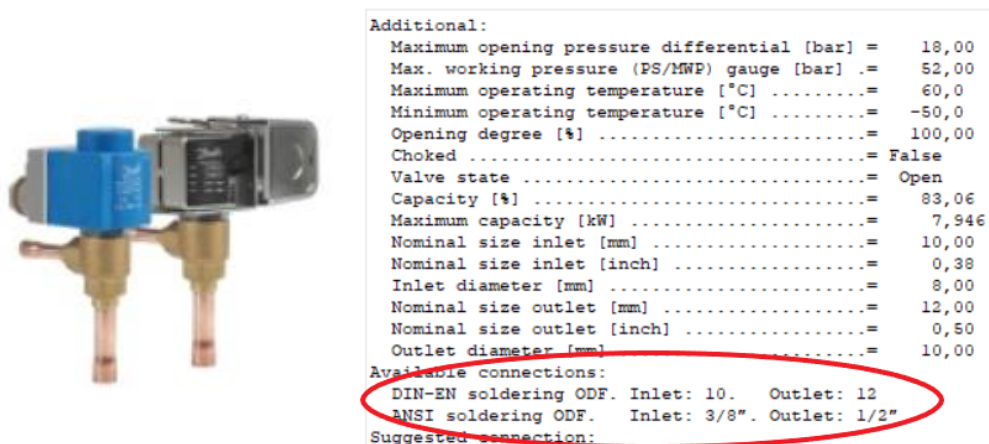


Figure 35 - Additional characteristics AKV 10-4 valve (adapted from [16]).

4.2.9 – Vapour Injection Valve

Much like the liquid injection valve, the vapour injection valve function is to replace or aid a heat exchanger. As discussed in section 4.2.1, compressors require a minimum vapour superheat in order to maintain correct operation. The function of the vapour injection valve is to ensure that the vapour on the suction side of the MT compressors has the minimum superheat required of 10 K.

In the case of the MT side of the refrigeration unit, 5 K of superheat is useful, i.e., they happen on the evaporators, and the other 5 K of superheat result from the heat exchanger 2 under normal operating conditions. In case that the LT compressors are not working, or the heat exchanger cannot provide the needed superheat, it is required that the vapour injection valve provides enough vapour from the hot gas cooler line, to raise the temperature of the fluid by the required 5 K.

In order to calculate the mass flow rate through the referred valve it is necessary to know the enthalpy of the fluid in different states.

According to [21], at MT evaporator pressure of 2,65 MPa (26,5 bar), the enthalpies are:

- $h_{\text{saturated}} = 435,14 \text{ kJ/kg}$;
- $h_{5 \text{ K Superheated}} = 442,10 \text{ kJ/kg}$;
- $h_{10 \text{ K superheated}} = 449,06 \text{ kJ/kg}$.

Also, the vapour leaving the MT and parallel compressors have an average enthalpy of:

- $h_{\text{to gas cooler}} = 510,31 \text{ kJ/kg}$.

*This refers to the average vapour enthalpy going to the gas cooler at moments when the LT compressors are not working. This state was selected, because when LT compressors are working, heat exchanger 2 provides the superheat necessary, and therefore, operating this valve becomes unnecessary.

The mass flow rate is then calculated according to the first law of thermodynamics, using equation (18).

$$\dot{m}_1 * h_1 + \dot{m}_2 * h_2 = \dot{m}_3 * h_3 \quad (19)$$

Where:

- $\dot{m}_1$ is the mass flow rate from the MT evaporators;
- h_1 is the enthalpy 5 K superheated;
- $\dot{m}_2$ is the mass flow rate of vapour passing on the injection valve;
- h_2 is the enthalpy of the vapour going to the gas cooler;
- $\dot{m}_3$ is the sum of the previous mass flow rates = $\dot{m}_1 + \dot{m}_2$;
- h_3 is the required final enthalpy = $h_{10 \text{ K superheated}}$.

This equation returns a value for $\dot{m}_2 = 181,8 \text{ kg/h}$. The vapour density before entering the valve according to [21] is $\rho = 165 \text{ kg/m}^3$. Using equation (16) results in $\dot{V}_2 = 1,09 \text{ m}^3/\text{h} = 0,00030 \text{ m}^3/\text{s}$.

As in section 4.2.2 and according to [3] the average velocity in this line is 10 m/s. By applying the equations (10) and (11) it results in a valve orifice of 6,21 mm in diameter.

The orifice diameter of a valve is a function of its Kv value. The Kv value is a measure of the flow rate through a valve for a given medium and pressure drop. The larger this value, the higher the flow rate through the valve will be at a given pressure drop.

The position that this valve occupies in the refrigeration unit is a very particular one. It needs to handle a high pressure of 12 MPa (120 bar), as well as a high temperature of 130°C.

The selected manufacturer for this valve is *Castel Italian technology*, and the family of valves is the 14-E series. These valves series are specifically designed to be suitable for CO₂ transcritical and subcritical systems with a maximum working pressure up to 140 bar, and a maximum working temperature above 130°C.

The 14-E family of valves has different models with different Kv values for each valve. However, there is no available correlation between Kv and orifice diameter. To obtain this correlation, the *GSR Ventiltechnik* catalogue was used. Figure 36 displays part of this catalogue.

TECHNICAL FEATURES



Seat Ø mm	Kv-value m³/h	Standard type	max. pressure for coils NC			ATEX NC	
			.182	.032	.012	.178	.148
1,0	0,06	.5240/..02/	0-50	0-90	0-90	0-20	0-90
1,5	0,09	.5241/..02/	0-30	0-85	0-90	0-16	0-90
2,0	0,13	.5242/..02/	0-15	0-40	0-90	0-10	0-55
2,5	0,16	.5243/..02/	0-8	0-22	0-45	0-6	0-28
3,0	0,20	.5244/..02/	-	0-15	0-30	-	0-20
4,0	0,35	.5245/..02/	-	0-8	0-16	-	0-10
5,0	0,50	.5246/..02/	-	0-5	0-10	-	0-6
6,0	0,75	.5247/..02/	-	0-4	0-8	-	0-5

Figure 36 – GSR Valve technical data sheet (adapted from [28]).

As it pointed out in Figure 36, a 6 mm diameter orifice/seat implies a Kv value of 0,75. Figure 37 presents Castel's catalogue of the 14-E family of valves.

Operating principle	Drawing	Part Number	Coil Type	Connections ODS Ø		Kv [m³/h]	PS [bar]	TS [°C]		Package Pcs.			
			(1)	[in.]	[mm]			Min	Max				
Direct acting		1418E/M6S05	HF3	-	6	0,065	140	-40 (2)	+130 (2)	14			
		1418E/M10S05		-	10						0,15		
Piston pilot operated		1428E/M6S020		-	6	0,32						140	-40
		1428E/M10S020		-	10								
		1428E/M6S030		-	6	0,80							
		1428E/M10S030		-	10								
		1438E/M10S050		-	10	1,10							
		1438E/M12S050		-	12								
		1438E/M10S070		-	10								
		1438E/M12S070		-	12								

Figure 37 - Castel's catalogue of the 14-E valves (adapted from [17]).

As it is seen, Castel does not offer a valve with a Kv value of 0,75, it does, however, offer one with a 0,80 value. The selected valve is the 1438E/M10S050, as it is the smallest one with the required capabilities. This valve is piston pilot operated and has stainless steel connections to match the refrigeration unit pipes. Figure 38 presents the selected valve.



Figure 38 - Vapour injection valve (adapted from [17]).

The selected valve based on the Kv value is not from the same manufacturer of the determined value. Even so, the Kv factor/value is a standardized key figure for the achievable flow rate of a fluid through a valve and has been in use since the 1950s. The Kv factor is calculated according to standard DIN EN 60 534, wherein the value is defined according to VDE/VDI 2173 guidelines by the measurement of water at a pressure drop of approximately 1 bar and a temperature of 5-30°C. The unit of the result is given in m³/h [29].

4.2.10 – Oil System

Oil Separator

As mentioned before, the mechanical components of a compressor need to be lubricated. Some of this oil inevitably enters the refrigerant line, which creates some problems. One of the problems is the fact that the oil does not create a cooling effect and increases the mass flow through the system. Another problem, which can be even more important, is that the compressors can run out of oil in their sump, if enough oil enters the refrigerant stream.

To prevent this from happening, the refrigeration unit is equipped with an oil system. This system is composed of an oil separator, an oil accumulator, and an oil filter and each compressor have its own oil level regulator.

The oil separator is located after the HP compressors on the exit of the refrigeration unit to the gas cooler. This device must be able to withstand the high pressures and temperatures of the working fluid. *Temprite* is a supplier of these products, and they have an online software [30] to select their devices. Figure 39 shows the results from the manufacturer software that was used to select the device.

Temprite

Home Products Our Company Resources Contact Search

Company Name:

Project:

Date: 04-07-2022

Refrigerant: R744tc

Units: SI

Type of Compressor: Reciprocating

Type of Separator: Conventional/Coalescent, Hermetic/Accessible, NO Reservoir

SST and Capacity: -10°C, 130 kW

Discharge Header Size: All Sizes

Gas Cooler Outlet Temperature: 40°C

Results:

Model 139A: Accessible
Capacity at 26%
Connections: 2 in

Model 138A: Accessible
Capacity at 34%
Connections: 1 1/2 in

Model 137A: Accessible
Capacity at 49%
Connections: 1 1/4 in

Figure 39 - Software results for an oil separator (adapted from [30]).

The software recommends three different oil separators. According to manufacturer recommendations, the oil separator pipe connections cannot be smaller than the pipe that it is fitted to. This parameter is the deciding factor when selecting one of these separators, since all of them are below 80% capacity which is the maximum according to [3].

As it was selected in section 4.2.2, the pipe that directs the refrigerant to the Gas Cooler is DIN-EN SS 25, which is equivalent to 1 inch. This means all the above separators are suitable for this refrigeration unit, therefore, the selected separator is Model 137A. This is the smallest one and consequently the less expensive.

Oil Accumulator

Connected to the oil separator is the oil accumulator. It stores the oil retrieved from the separator and ensures a constant feed to the compressors. Furthermore, in the oil accumulator, any refrigerant trapped within the oil is boiled-off and the oil is cooled.

An oil accumulator must be used when there are compressors running in parallel that can be started and stopped individually. One of the biggest problems with this type of installation is ensuring the correct oil level for the different operating conditions. This is because when stopped, the oil level in the compressor crankcase can drop, preventing the compressor from starting when needed.

The size of the oil accumulator required is determined by the number of compressors connected in the unit, compressor oil charge, system refrigeration charge and operating conditions. In Figure 40, it is presented a selection chart of oil accumulators based on *ESK Schultze*'s experience.

Ölsammler OSA-CDH → P_{smax}: 130 bar

Oil reservoirs OSA-CDH → P_{smax}: 130 bar

Technische Spezifikation

Max. zulässiger Betriebsüberdruck (P_{smax})

im Temperaturbereich

[1] Zul. Betriebstemperatur: 100 ... -10 °C → P_{s1} = 130 bar

[2] Zul. Betriebstemperatur: -10 ... -40 °C → P_{s2} = 97,5 bar

Technical specification

Max. allowable operating pressure (P_s max)

according to the temp. range

[1] Allow. operating temperature: 100 ... -10 °C → P_{s1} = 130 bar

[2] Allow. operating temperature: -10 ... -40 °C → P_{s2} = 97.5 bar

Auslegung Ölsammler

Selection of oil reservoirs

Ölsammler Oil reservoir	Verdichter-Anzahl und Ölfüllung pro Verdichter [Liter] Number of compressors and oil charge per compressor [litres]									
Typ / Type	Stck./pcs l		Stck./pcs l		Stck./pcs l		Stck./pcs l		Stck./pcs l	
OSA-5-CDH	3	2	4	1,5						
OSA-12-CDH	3	4	4	3,0	5	2,5	6	1,6	7	1,4
OSA-23-CDH	3	6,8	4	5,1	5	4,1	6	3,4	7	2,9
									8	2,5

Technische Daten

Technical data

Ölsammler Oil reservoir	Inhalt Volume				Abmessungen Dimensions						Gewicht Weight
Typ Type	Vt l	V1 l	V2 l	V3 l	ØD mm	H mm	E mm	D mm	C mm	F mm	kg
OSA-5-CDH	5,0	0,8	2,5	4,5	114	782	166	391	636	166	16,4
OSA-12-CDH	12,0	1,5	6,0	10,9	159	949	182	482	802	182	38,6
OSA-23-CDH	23,0	2,5	11,5	20,5	219	928	168	476	783	198	68

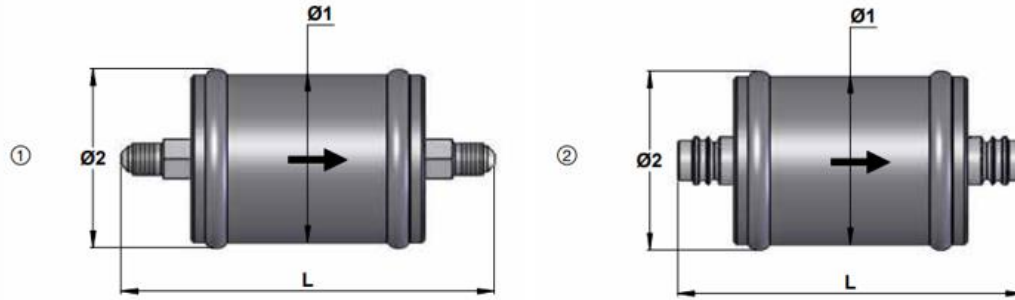
Vt = Volumen gesamt / Total volume

Figure 40 - Oil accumulator selection chart (adapted from [31]).

These oil accumulators have a maximum allowable pressure of 130 bar. This is required because the safety valve on the high-pressure side of the system (gas cooler pressure) is set to 120 bar. The refrigeration unit is composed of 5 separate compressors that all have the same oil charge of 2,5l. According to the selection chart, for 5 compressors with an average oil charge of 2,5l it is recommended to use the OSA-12-CDH. As is seen in Figure 40, this accumulator has a total volume of 12L and weighs 38,6kg.

Oil Filter

The oil filter is installed between the oil accumulator and the compressors oil inlet. It is intended to prevent solid particles from entering the compressors. It is important that this filter withstands high pressure and that it is suitable for the type of oil and operating fluid in question. Taking these factors into consideration, the selected filter from the *Carly* catalogue is the HCYF-P14 53. Figure 41 displays the manufacturer catalogue of oil filters.



CARLY references	Volume	Maximal working pressure	Working pressure ⁽¹⁾	Maximal working temperature	Minimal working temperature	Working temperature ⁽¹⁾	CE Category ⁽²⁾
	V L	PS bar	PS BT bar	TS maxi °C	TS mini °C	TS BT °C	
HCYF-P14 52	0,11	140	15	120	-40	-30	Art4§3
HCYF-P14 52 S/MMS	0,11	140	15	120	-40	-30	Art4§3
HCYF-P14 53	0,11	140	15	120	-40	-30	Art4§3
HCYF-P14 53 S/MMS	0,11	140	15	120	-40	-30	Art4§3

Figure 41 – Carly catalogue of oil filters and characteristics (adapted from [25]).

The main difference between the filters displayed in Figure 41 is the connection types. The HCYF-P14 53 was selected as it has threaded connections corresponding to ①.

Oil Level Regulators

In multiple-compressor parallel systems, oil level regulators are installed on the oil inlet of each compressor to ensure they maintain an adequate oil level. The selected oil level regulators are from the same manufacturer as the oil accumulator *ESK Schultze*, and the selected model is the ERM 6-CDH-OC. These regulators have solenoid valves that can withstand pressure drops of up to 100 bar. Their maximum pressure permissible is 130 bar, as represented in Figure 42, from the product catalogue of the manufacturer.

Technische Daten			Technical data	
Elektronischer Ölspiegelregulator Electronic Oil level regulator	Zul. Arbeitsdruckdifferenz Ölrückführung Allow. working pressure difference oil return	Max. zul. Öldruck in der Ölrückföhrleitung Max. allowable pressure in the oil return line	Max. zulässiger Betriebsdruck Max. allowable working pressure	Verdichteranschluss Ausführung Compressor connection version
Abb./Typ Fig./Type	bar	bar	bar	
a ERM 6-0-BC b ERM 6-0-BC-L	1,5 ... 100*	130	60	3/4-Loch-Flansch / 3/4-bolt flange 3/4-Loch-Flansch lang / 3/4-bolt flange long
c ERM 6-OC d ERM 6-OC-B				Gewinde / Thread: 1.1/8"-18 UNEF
e ERM 6-CDH-OC f ERM 6-CDH-OC-B			130	Gewinde: 1.1/8"-18 UNEF (Mit Gewinde G1" oder M36 auf Anfrage) Thread: 1.1/8"-18 UNEF (with threaded connection G1" or M36 on request)
a ERM 6-0-BC-R717 c ERM 6-OC-R717	31*	31	31	3/4-Loch-Flansch / 3/4-bolt flange Gewinde / Thread: 1.1/8"-18 UNEF

Figure 42 - Electronic oil regulator catalogue from ESK Schultze (adapted from [31]).

These oil level regulators are able to shut down their corresponding compressor in case of insufficient oil. This is an indispensable safety feature to prolong the operating life of the compressor.

Oil Solenoid Valve

The oil solenoid valve is located between the oil separator and the accumulator. Its function is to prevent refrigerant from entering the accumulator, separating the two devices. When the bottom of the separator has enough oil volume, the valve opens, allowing it to enter the accumulator, and then, closes, before removing all the oil from the separator. This ensures that the bottom of the separator maintains a small volume of oil, preventing the ingress of refrigerant into the oil system.

The valve selected for this task is from the manufacturer ESK and it is the model ESK CO2 MV-11W-1-CDH-P. This valve was selected as it is able to withstand both pressure and temperature required. Figure 43 shows the selected oil solenoid valve.

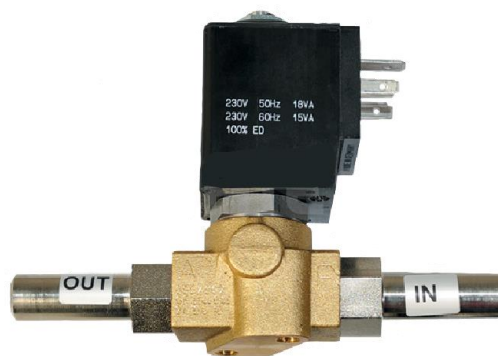


Figure 43 - Oil Solenoid Valve [31].

4.2.11 – Thermal Insulation

Thermal insulation minimizes heat transfer between objects in thermal contact or in the range of radiative influence. It consists in low thermal conductivity materials combined to achieve an even lower system thermal conductivity. Thermal insulation can be achieved with specially engineered methods or processes, as well as with suitable object shapes and materials.

Insulation is typically used on the suction lines to prevent condensation and to limit unwanted parasitic heat gains. Any heat gain to these suction lines will increase the amount of work required at the compressor. Hot gas lines on refrigeration systems generally do not require insulation since the objective is to reject heat to the environment [32].

Thermal Insulation is not only applied to return lines from the evaporator, but also to any component or line that is below ambient temperature. In the case of this refrigeration unit, the liquid receiver, the heat exchangers, the liquid line, and the aspiration line for the parallel compressor and flashgas bypass valve are also insulated.

A supplier of thermal insulation is *Armacell*. They focus on insulation materials for technical equipment, high-performance foams for high-tech and lightweight applications. The product selected for insulating the refrigeration components is their AF/ArmaFlex (Figure 44). This engineered foam is, not only a thermal insulator, but also it is the optimal combination of a very low thermal conductivity with extremely high resistance to water vapour transmission. This prevents long-term energy losses and water vapour ingress, which reduces the risk of under insulation corrosion. The built-in Microban® antimicrobial protection and the excellent fire performance make the product especially suitable for use in public buildings and process industries [33].



Figure 44 - AF/Armaflex insulation products [33].

Different lines have different temperatures, which implies different insulation thicknesses. Thermal insulation, as said before, prevents heat absorption but also condensation on the surface and, therefore, in the selection of the insulation thickness, the manufacturer software was used.

Figure 45 presented the software window for the input parameters that were used to calculate the insulation thickness for the liquid line.

CALCULATION OPTIONS   

Calculation Type: Condensation

Calculation Options: Thickness to be calculated

Object type: Pipe

Material: Stainless steel

Nominal size (DN): 25 Outer diameter: 28.0 [mm]

Wall thickness: 1.20 [mm]

Object orientation: Horizontal

Length: 1.00 [m]

Medium: Carbon dioxide (CO2)

Insulation Material: -- AF/ArmaFlex -- Sheets

Surface Emissivity: Flexible Elastomeric Foam (E= 0.930)

Calculation Standard: ISO 12241

Relative humidity: 85.00 [%] Pipe / Ducts - humid [85]

Ambient temperature: 38.00 [°C] ---Please select---

Line temperature: 5.00 [°C]

Indoor/Outdoor: Indoors

Figure 45 - Input parameters for calculating thermal insulation thickness (adapted from [34]).

As is seen in Figure 45, this software calculates the insulation thickness according to the ISO 12241 Standard. This standard gives rules for the calculation of heat-transfer-related properties of building equipment and industrial installations, predominantly under steady-state conditions, which refrigeration systems and units must follow. Figure 46 presents the result provided by the software.

CALCULATION OPTIONS  

THE FLEXIBLE INSULATION SYSTEM FOR RELIABLE CONDENSATION CONTROL, TRUSTED SINCE 40 YEARS
AF/ARMAFLEX® – the high-performance, energy-efficient professional insulation for use in air-conditioning, refrigeration and process facilities.

Thermal transmittance: 0.182[W/m·K]

Min. required insulation thickness [mm]	Selected insulation	Mean insulation thickness [mm]	Outer surface temperature [°C]	Heat Flow [W/m]
25.00	If necessary please contact us.		35.4	6.00

Figure 46 – Thermal insulation calculation results [34].

The same principle was applied to every line on the system, as well as the liquid receiver. On the heat exchangers, the thickness of thermal insulation applied was determined according to [3].

Table 9 presents the insulation thickness applied to different components.

Table 9 -Thermal insulation thickness of different components.

Position/line	Temperature [°C]	Insulation thickness [mm]
MT suction line	-5	32
LT suction line	-25	40
Liquid line	5,3	25
Liquid receiver	5,3	32
Heat exchangers	-	32
Parallel compressor/Flashgas bypass aspiration line	5,3	25

4.2.12 – Emergency Group

In the hypothetical case of a power failure, all the electrical components on the refrigeration unit would stop working. This relates to compressors, fans on the gas cooler and evaporators, electrical expansion valves and the refrigeration unit control system. This also stops the refrigerant flow through the whole system.

Despite the thermal insulation discussed in section 4.2.11, the liquid receiver and liquid line, as well as the return lines from the evaporators, due to the temperature difference to their surroundings, will absorb heat. Heat absorption on the return lines from the evaporators is not a big concern, as the refrigerant density does not decrease drastically. However, the actual concern is the heat absorption in the liquid line and receiver. At liquid receiver pressure, the saturated liquid density is 7,5 time higher than the saturated vapour one. This difference in density, in a confine space, implies a drastic pressure increase, which would activate the safety valves in the unit and release the refrigerant to the atmosphere.

To prevent this from happening, the refrigeration unit is equipped with an emergency group. The emergency group is a remote condensing unit that includes a compressor, a condenser and an expansion valve fitted to the unit's main frame, and it is connected to a diesel backup generator.

Remote condensing units come in several different cooling capacities. Therefore, it is necessary to select one, based on the actual cooling necessities. To determine the cooling necessities, a calculator was developed. This calculator takes in to account the dimensions of the liquid receiver and liquid line and it also factors in the thermal insulation applied to each of these surfaces.

Refrigeration units are placed in ever changing environments. Their engine rooms have slow-moving air or are ventilated, and the surrounding walls have very different temperatures depending on if they are shaded or not. In order to avoid errors in convection and radiation calculations, this calculator assumes the exterior temperature as the surface temperature of the insulation.

Figure 47 displays part of the interface of the calculator.

Liquid Receiver	
Liquid receiver:	RV180-130bar-TR
Volume [l]	RV90-130bar-TR (355)
L [m]	RV120-130bar-TR
Ø [m]	RV150-130bar-TR
Top area w/ insulation [m ²]	RV180-130bar-TR
Side area w/ insulation [m ²]	RV250-130bar-TR
	RV300-130bar-TR
	RV400-130bar-TR
	RV500-130bar-TR
Conditions	
Exterior Temp. [°C]	38
Receiver pressure [bar]	42
Interior Temp. [°C]	7,22
Receiver insulation	
Thickness [mm]	32
Thickness [m]	0,032
Insulation conductivity [w/mK]	0,033
Polypropylene base conductivity [w/mK]	0,4
Polypropylene base thickness [m]	0,015

Figure 47 - Emergency group calculator receiver conditions.

This calculator also takes in consideration the incapacity of insulating the bottom of the liquid receiver the same way as the rest of it, as that is where the mounting surface is. In other to avoid direct conduction between the liquid receiver and the refrigeration unit structure, a layer of polypropylene is used between the two, and the air gap in the bottom of the receiver is also factored in.

According to [3], a commercial refrigeration system of this size typically has liquid and return lines from the evaporators of around 50 m. This value is also taken into account by the calculator. Lastly, a safety coefficient is applied by the user, in this case, the chosen value was 1,3.

Figure 48 displays the complete interface of the calculator and the result provided by it.

Emergency group calculator					
Liquid Receiver		Liquid Line			
Liquid receiver:	RV180-130bar-TR	Exterior diameter [mm]	33,7	Receiver heat absorption [w]:	100,7
Volume [l]	180	Exterior diameter [m]	0,0337	Liquid line heat absorption [w]:	350,7
L [m]	1,646	Length [m]	50	Total heat absorption [w]:	451,4
Ø [m]	0,406	Insulation thickness [mm]	25	Safety coefficient:	1,3
Top area w/ insulation [m ²]	0,1735	Insulation thickness [m]	0,025	Emergency group cooling P. [w]:	586,8
Side area w/ insulation [m ²]	2,4304	Insulation conductivity [w/mK]	0,033		
Conditions		R Insulation	0,0878		
Exterior Temp. [°C]	38				
Receiver pressure [bar]	42				
Interior Temp. [°C]	7,22				
Receiver insulation					
Thickness [mm]	32				
Thickness [m]	0,032				
Insulation conductivity [w/mK]	0,033				
Polypropylene base conductivity [w/mK]	0,4				
Polypropylene base thickness [m]	0,015				
Intermediate results					
R side insulation	0,41285				
Q side [w]	74,55				
R top insulation	5,58921				
Q top [w]	5,51				
R polypropylene base	0,21615				
R air gap	1,2769				
Q base [w]	20,61				
Air properties	0,04514				
Air gap thickness [m]	0,01				

Figure 48 – Emergency group calculator results.

The cooling load required by this refrigeration unit is 586,8 W. Therefore, the remote condensing unit must be able to provide this cooling load at these operating conditions, i.e., 38°C of ambient temperature and 7°C of refrigerant temperature.

One supplier of remote condensing units is Danfoss, which means it is possible to select one from their software [16].

Figure 49 presents the first characteristics of the remote condensing unit.

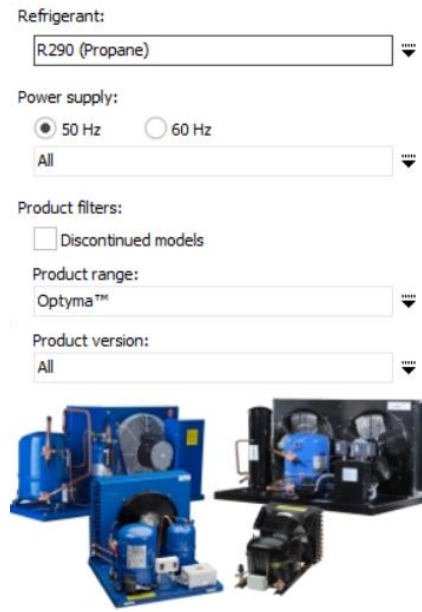


Figure 49 - Remote condensing unit (adapted from [16]).

The selected refrigerant for the emergency group is the R290 (propane) which, like the CO₂, has a low GWP of 3 [35]. Nonetheless, unlike the CO₂, it does not have as high of a coefficient of expansion, and hence, can be used in a partial time refrigeration system as the emergency group is.

For selecting the actual model of the emergency group, it is necessary to provide the software with operating conditions to select a remote condensing unit. Figure 50 displays the inputs provided to the software.

Figure 50 - Operating conditions for the emergency group (adapted from [16]).

As is shown in Figure 50, the cooling capacity as well as the ambient temperature were provided. The evaporation temperature was selected at 5°C, which provides a 2°C difference on the heat exchanger.

Figure 51 presents the software selection for a remote condensing unit based on the provided operating conditions.

Selected	Model	Code number	Compressor model	Product range	Product version	Refrigerant	Cooling [kW]	COP cooling [W/W]	Total power [kW]	Total current [A]	Frequency [Hz]	Power supply	Tc [°C]	Match
☐	OP-MCNC004NYA09G	114F1205	NLY45Rab	Optyma™	A09	R290	0,494	2,08	0,237	1,316	50	220 - 240 V 1 ph	54,9	84%
☐	OP-MCNC006NYA11G	114F1204	NLY45Rab	Optyma™	A11	R290	0,494	2,08	0,237	1,316	50	220 - 240 V 1 ph	54,9	84%
☒	OP-MCNC006NYA10G	114F1309	NLY60Rab	Optyma™	A10	R290	0,676	2,13	0,317	1,628	50	220 - 240 V 1 ph	54,4	115%
☐	OP-MCNC006NYA09G	114F1308	NLY60Rab	Optyma™	A09	R290	0,676	2,13	0,317	1,628	50	220 - 240 V 1 ph	54,4	115%
☐	OP-MCNC008NYA10G	114F1412	NLY80Rab	Optyma™	A10	R290	0,966	2,43	0,398	2,231	50	220 - 240 V 1 ph	50,3	165%

Figure 51 - Emergency group selection (adapted from [16]).

The best match for this task is the OP-MCNC006NYA10G. This is a remote condensing unit that has 115% of the required capacity. However, it is the smallest device that provides the required cooling load, so, it is the selected device.

Chapter 5

Project Analysis

At this stage, with the refrigeration unit diagram and all the components selected, it is possible to do a project analysis of the energy consumption of the unit. It should be noted that, due to the fact that the project is not yet implemented, it is not possible to obtain real values for the energy consumption of the system. Therefore, it was necessary to resort to calculation software's.

In addition to the energy consumption of the refrigeration unit, it is also important to know the initial cost associated with the implementation of the project.

Throughout this analysis, a traditional CO₂ booster refrigeration system with parallel compression will be used as a reference. This is due to the fact that this is the refrigeration unit system that is intended to be replaced with this project, assessing whether there are any economic advantages in using the system designed in this dissertation.

5.1 - Initial Cost Analysis

Starting by analysing the reference refrigeration unit, which is a Booster CO₂ with parallel compression, it can be observed on Table 10 the price of most components necessary for the correct operation of a similar power refrigeration unit. There are several components that are the same in both the reference and the actual refrigeration unit, which means, the price is the same in both cases. There are, however, some prices that can differ, such as the frame, for instance. This will not be the exact same one for both units, but, in these cases, an approximate price is used for a better comparison between them.

Table 10 presents components, quantity, and price for a reference Booster CO₂.

Table 10 – Cost of components for a reference Booster CO₂ with parallel compression [3].

Component	Quantity	Price per unit [€]	Total price [€]
MT Compressor	3	5923,40	17770,20
LT Compressor	2	3184,70	6369,40
Parallel Compressor	1	5753,40	5753,40
MT Compressor	1	5310,40	5310,40
Liquid receiver	1	3680,00	3680,00
Heat exchanger	1	358,00	358,00
Filters and Cores	3	295,26	885,78
Oil separator	1	1850,00	1850,00
Oil accumulator	1	1760,00	1760,00
Oil level regulator	7	145,00	1015,00
Oil filter	1	309,47	309,47
Transcritical valve	1	1246,00	1246,00
Flash gas bypass valve	1	1246,00	1246,00
Emergency group	1	601,00	601,00
Frame	1	8000,00	8000,00
Safety valves	1	1400,00	1400,00
Cut-off valves	57	55,22	3147,54
		Total [€]	60702,19

The reference refrigeration unit is composed of seven separate compressors: two of them are LT compressors, four are MT compressors and there is a parallel compressor as well. In order to approximate the reference unit with the projected refrigeration unit, the parallel compressor is the same for both units. The LT stage of compressors uses two CDS401B compressors. These were selected as they are the smallest ones that could create the required cooling load on the respective evaporator, according to the software. On the MT stage of compression, it was necessary to use three CD 2500H compressors, as well as an extra CD 1400H equipped with an inverter in order to ensure the correct operation of the unit. The number of compressors on the reference refrigeration unit is not the same as in the projected one. This number of compressors was selected based on existing refrigeration units already installed and operating of similar size according to [3].

In Table 11, it is possible to observe the total cost of components of the projected refrigeration unit. This refrigeration integrates extra components that could also exist in a reference Booster CO₂ system. However, because of the total separations of evaporation stages that this unit provides, they are an absolute necessity.

Table 11 – Cost of components of the projected CO₂ refrigeration unit.

Component	Quantity	Price per unit [€]	Total price [€]
Dual-stage Compressor	2	6786,20	13572,40
Parallel Compressor	1	5753,40	5753,40
MT Compressor	1	5515,40	5515,40
MT Compressor	1	6369,40	6369,40
Liquid receiver	1	3680,00	3680,00
Heat exchangers	3	815,34	2446,02
Filters and Cores	3	295,26	885,78
Oil separator	1	1850,00	1850,00
Oil accumulator	1	1760,00	1760,00
Oil level regulator	5	145,00	725,00
Oil filter	1	309,47	309,47
Transcritical valve	1	1246,00	1246,00
Flash gas bypass valve	1	1246,00	1246,00
Injection valves	2	467,16+191,00	658,16
Emergency group	1	601,00	601,00
Frame	1	8000,00	8000,00
Safety valves	1	1400,00	1400,00
Cut-off valves	57	55,22	3147,54
		Total [€]	59165,57

Comparing both units in terms of the initial component cost it can be concluded that the price difference between the two is insignificant, around 2,5%. The main difference is the number of compressors on the projected refrigeration unit. The dual-stage compressors effectively act as four compressors, while not being twice as expensive.

Equipment cost is not the complete cost of the system. There are other initial costs, for example, the control system, the refrigerant piping, the labour for the assembly of the components and the refrigerant initial charge. Even so, all these costs are considered the same for both systems. Another parameter that can be considered equal is the rate of leakage, since both systems have the same cooling load and various other components, the leakage should be the same at around 10% to 15% per year. However, this is not an initial cost for the unit, it is only a maintenance cost over the lifetime of the unit.

5.2 – Performance Analysis

In this section is presented a performance analysis of both the reference refrigeration unit and the projected one. This analysis presents the COP and the flash gas operating mode of the refrigeration unit for each of the operating modes that were determined in Chapter 3. In the case of the reference refrigeration unit, the COP values for each operating mode were obtained using the Dorin Software [19], calculating directly in the BOOSTER (CO₂) mode.

Selecting the flash gas mode depends on the parallel compressor inverter frequency. In the case it being below 30 Hz the system is changed into flash gas bypass, instead of parallel compression, as this is the minimum frequency that guarantees a reliable operation of the compressors according to [3]. Table 12 presents the resulting COP and flash gas mode for each operating mode of the reference refrigeration unit.

Table 12 - COP and flash gas mode of a reference Booster CO₂.

Cooling load	MT= 100 kW and LT= 30 kW		
Gas cooler outlet temperature [°C]	41	33	15
Opt. Gas cooler pressure [bar]	101	81,5	51
Opt. Liquid receiver pressure [bar]	42	42	35
COP	1,31	1,86	3,58
Flash gas mode	Parallel comp.	Parallel comp.	FG bypass
Cooling load	MT= 0 kW and LT= 30 kW		
Gas cooler outlet temperature [°C]	41	33	15
Opt. Gas cooler pressure [bar]	101	81,5	51
Opt. Liquid receiver pressure	35	35	35
COP	0,95	1,22	2,11
Flash gas mode	FG bypass	FG bypass	FG bypass
Cooling load	MT= 100 kW and LT= 0 kW		
Gas cooler outlet temperature [°C]	41	33	15
Opt. Gas cooler pressure [bar]	101	81,5	51
Opt. Liquid receiver pressure [bar]	45	42	40
COP	1,66	2,27	4,94
Flash gas mode	Parallel comp.	Parallel comp.	FG bypass
Cooling load	MT= 50 kW and LT= 15 kW		
Gas cooler outlet temperature [°C]	41	33	15
Opt. Gas cooler pressure [bar]	101	81,5	51
Opt. Liquid receiver pressure [bar]	45	37	35
COP	1,47	1,92	3,86
Flash gas mode	Parallel comp.	Parallel comp.	FG bypass

The extra operating modes determined in Chapter 3 establish different cooling loads and also establish different gas cooler outlet temperatures. However, as is seen in Table 12, there are two other parameters displayed, which are the pressure of the gas cooler and of the liquid receiver. Both these parameter influences the COP along with the operation of the refrigeration unit. The gas cooler pressure differs, depending on the outlet temperature, and the selected one for each temperature is the one that maximizes the COP. The same can be said about the liquid receiver pressure, as its pressure also influences the COP. This is because of the percentage of flash gas that has to be compressed by the parallel compressor. Each operating mode was tested with a variety of liquid receiver pressures, and the one selected is the one that maximizes the COP. Both these parameters are important to establish, so that the comparison between both systems is as close as possible and generates useful data. The complete outputs from the software for the project conditions of the reference refrigeration unit are presented in Appendix E. Table 13, presents the COP as well as flash gas mode of the projected refrigeration unit.

Table 13 - COP and flash gas mode of the projected refrigeration unit.

Cooling load	MT= 100 kW and LT= 30 kW		
Gas cooler outlet temperature [°C]	41	33	15
Opt. Gas cooler pressure [bar]	101	81,5	51
Opt. Liquid receiver pressure [bar]	42	42	35
COP	1,37	1,90	3,65
Flash gas mode	Parallel comp.	Parallel comp.	FG bypass
Cooling load	MT= 0 kW and LT= 30 kW		
Gas cooler outlet temperature [°C]	41	33	15
Opt. Gas cooler pressure [bar]	101	81,5	51
Opt. Liquid receiver pressure [bar]	40*	35	35
COP	1,09	1,32	1,91
Flash gas mode	FG bypass	FG bypass	FG bypass
Cooling load	MT= 100 kW and LT= 0 kW		
Gas cooler outlet temperature [°C]	41	33	15
Opt. Gas cooler pressure [bar]	101	81,5	51
Opt. Liquid receiver pressure [bar]	45	42	40
COP	1,65	2,28	5,02
Flash gas mode	Parallel comp.	Parallel comp.	FG bypass
Cooling load	MT= 50 kW and LT= 15 kW		
Gas cooler outlet temperature [°C]	41	33	15
Opt. Gas cooler pressure [bar]	101	81,5	51
Opt. Liquid receiver pressure [bar]	45	37	35
COP	1,46	1,72	3,47
Flash gas mode	Parallel comp.	Parallel comp.	FG bypass

* These values have been altered from the reference table, as on this particular system they provide an efficiency advantage, increasing the COP.

As discussed before, the manufacturers software does not have a calculation mode that integrates the dual-stage compressors on a BOOSTER CO₂ refrigeration system. Therefore, it was not possible to obtain the COP and Flash gas mode directly from the software. For the operating modes that include just one of the evaporator stages, it was possible to use the software directly. For the MT evaporators without LT, the calculation mode selected is the MT transcritical systems (CO₂), which allows changes on the Flash gas mode as well. In the case of having just the LT stage of evaporators, the selected calculation mode was the Semi-Hermetic Compressors Double Stage CD_2S (CO₂). This calculation mode has a flash gas bypass option that compresses it on the high-pressure stage of the compressor, so this option was selected.

For the other operating modes, it was necessary to use both these calculation options on the software, and finally calculate the COP separately. The software outputs an “absorbed power” value for each calculation, depending on the compressor working conditions. This value was added from both calculations option in order to calculate the COP. In the cases where the parallel compressor can compress the entirety of the flash gas, the LT compressors calculation was changed into Intercooler configuration. Then the outlet temperature on the gas cooler was raised in the MT calculation, so that the total mass flow rate of flash gas was equal to the sum of flash gas created from the complete system. By doing this, it was possible to obtain a real absorbed power value that included the total work of the parallel compressor.

On the other hand, if the parallel compressor frequency was too low to maintain continuous operation, the total flash gas from the entire system was added to the LT calculation in a bypass configuration. Afterwards, the absorbed power of the parallel compressor on the MT calculation was subtracted from the total absorbed power, calculating the correct COP. If the parallel compressor was not enough, the excess flash gas vapour was added to the LT calculation and removed from the MT calculation, once again, allowing a correct COP calculation.

Appendix F contains the software reports used for calculating the COP and flash gas mode of the projected refrigeration unit for project conditions.

Figure 52 presents a graph which condenses the information of Tables 12 and 13, and allows for a final comparison.

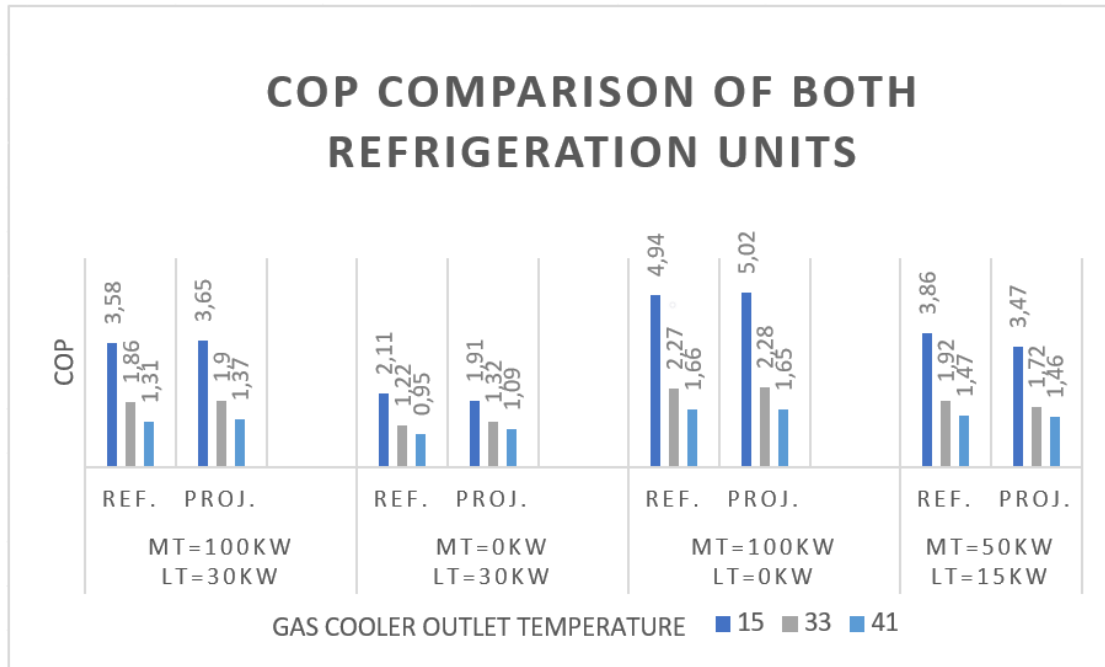


Figure 52 - COP comparison of both refrigeration units.

As one can see, in the first three operating modes, MT=100kW and LT=30kW, the projected refrigeration unit has a higher COP than the reference one, at an average of 3,2%, which is not significant.

In the following three operating modes, MT=0kW and LT=30kW, in the case of gas cooler outlet temperature of 41°C, the projected refrigeration unit has a COP 14,7% higher than the reference unit. At 33°C it has 8,2% higher COP, though, it presents a disadvantage of 10,5%, when the temperatures are cooler.

When the LT and dual-stage compressors are not operating, MT=100kW and LT=0kW, the refrigerating unit behave the same in terms of energy efficiency, since the COP differs an average of 0,6% across the three operating modes.

Finally, in the last operating modes, MT=50kW and LT=15kW, the projected refrigeration unit presents an inferior COP of 7,6% on average, when compared to the reference system.

Chapter 6

Conclusions and Future Work

Conclusions

This project was developed throughout one semester between March and July 2022. It was possible to accomplish a refrigeration unit project that met all the proposed objectives. The main objective was a refrigeration unit diagram, which integrates dual-stage compressors on a parallel compression Booster CO₂ system, while selecting all the necessary components. The total cost of this refrigeration unit, as well as its efficiency, were compared with a reference refrigeration unit, allowing for a greater understanding of commercial refrigeration.

With this project it was possible to observe that a system of this nature has a lot of variables. For instance, the location influences the outside temperature, which is one of the most important parameters, as well as cooling load and the working fluid itself. On another hand, the economical factors are also present in the component selection, as they can favour some manufacturers over others.

The refrigeration unit is not assembled and operating, which makes it impossible to obtain actual power consumption data. Despite this obstacle, a study was carried out on the basis of simulations using the Dorin Software [19]. These simulations allow for an estimation and comparison of the COP of this refrigeration unit. Pricing of various components were obtained directly from the suppliers, which made it possible to assess the initial cost of the unit.

In spite of integrating dual-stage compressors, as well as extra components, the price of this refrigeration unit is similar to the price of a standard Booster CO₂. The same can be concluded about the energy consumption and efficiency. In most operating modes, the projected refrigeration unit has a similar theoretical COP as the conventional Booster CO₂.

The main advantage of the projected refrigeration unit is the fewer number of compressors, since que dual-stage compressors act as two compressors, without requiring the same space as two separate compressors. This can be an important advantage, depending on the final location of the refrigeration system and the space available for it. The ability of shutting down compressor stages does not increase the efficiency of the system at the studied operating modes, however, it reduces the number of compressors operating, which subsequently reduces ware and maintenance required.

It is also important to state that the interest in refrigeration units that operate exclusively with natural refrigerants, such as CO₂, is more than just environmental, but also economic and legal, due to the restrictions imposed by the European Commission. The F-GAS regulation proves that despite all the environmental concern in which the world is involved, economic factors are the ones that drive the market and companies.

Finally, it can be concluded that on a Booster refrigeration system integrating dual-stage compressors do not provide an advantage in performance, since the projected refrigeration unit only had considerable higher COP in freezer operating modes i.e., the dual-stage compressors are operating by themselves. Therefore, these range of compressors are more suitable for refrigeration units that only require one set of LT evaporators.

Future Work

In order to complete the full project of the refrigeration unit, as the mechanical systems are defined, it is now important to design the control system for the unit. This control system will allow it to operate in ideal conditions maximizing the COP.

Afterwards, with the unit completed and operating, it will be interesting to verify the energy consumption of the system, as well as the economic and environmental performances. These data will allow for a better comparison of actual energy consumption when compared with the traditional CO₂ refrigeration system.

Finally, studying the integration of different technologies in these refrigeration systems, for example using flooded evaporators or multi ejectors as these devices operating modes, in theory, would increase the energy efficiency of the refrigeration units, which should result in lower operating costs.

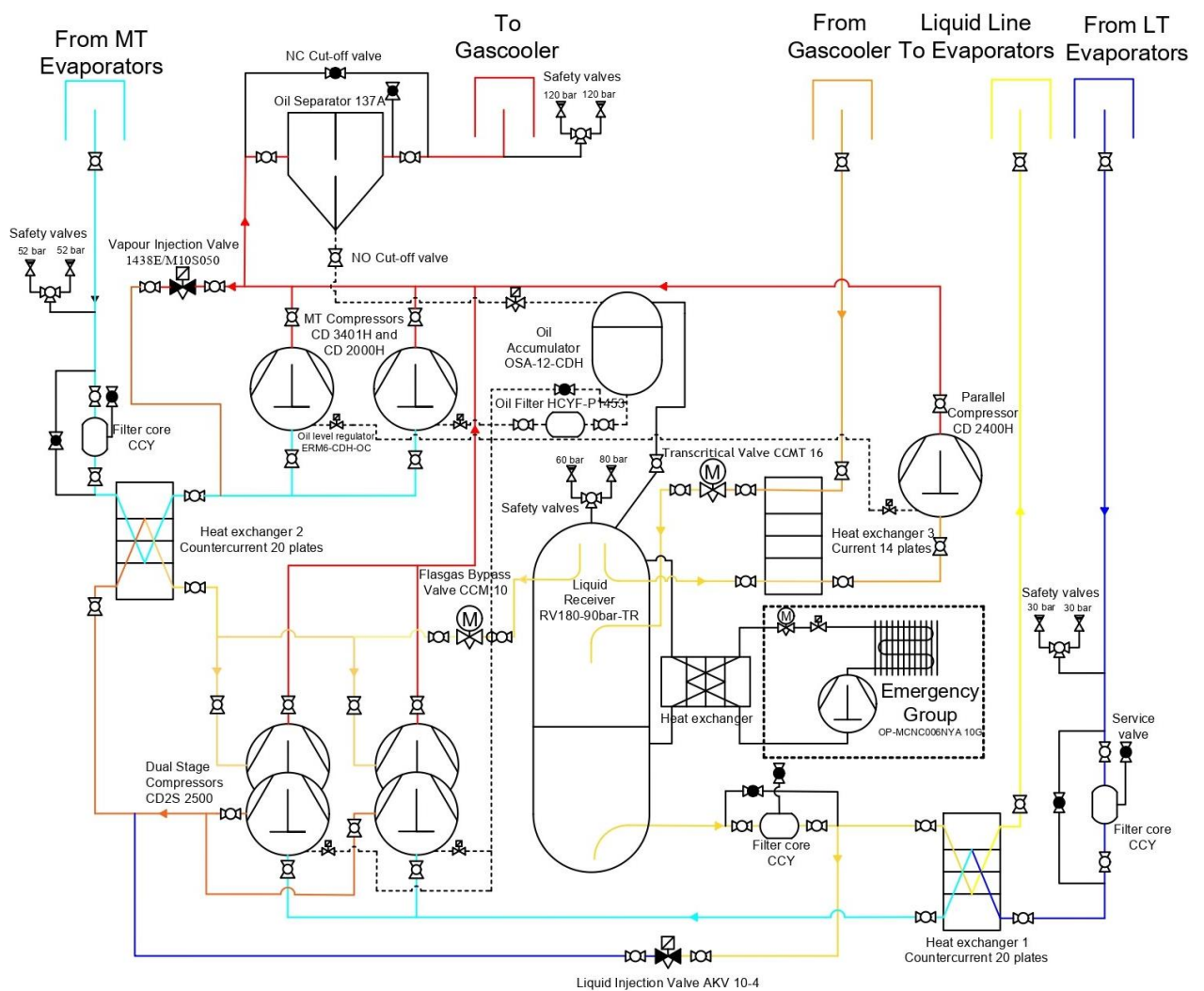
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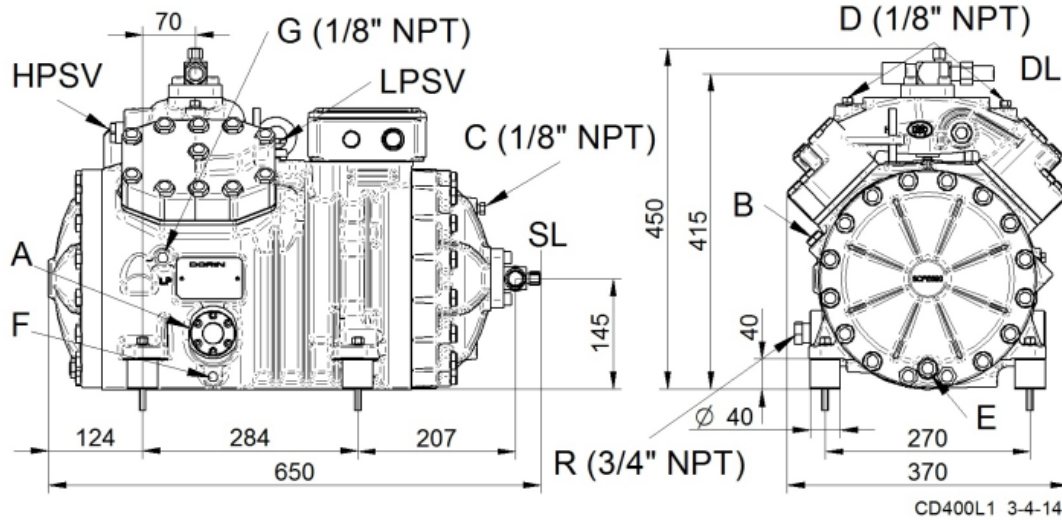
Appendix

Appendix A: Refrigeration Unit Diagram:



Appendix B: Selected Compressors Overall Dimensions

Overall dimensions [mm]: CD 2000H



A - Oil sight glass

B - Oil charge plug

C - Low pressure connection

D - High pressure connection

E - Oil drain plug

F - Crankcase heater

G - Oil return plug

R - Oil level connection

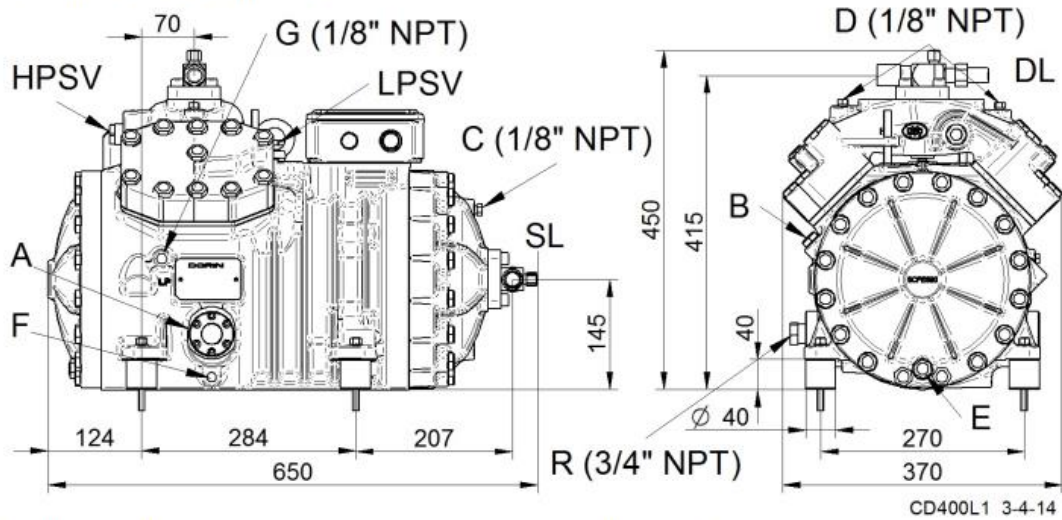
DL - Discharge service valve

SL - Suction service valve

LPSV - LP safety valve

HPSV - HP safety valve

Overall dimensions [mm]: CD 2400H



A - Oil sight glass

B - Oil charge plug

C - Low pressure connection

D - High pressure connection

E - Oil drain plug

F - Crankcase heater

G - Oil return plug

R - Oil level connection

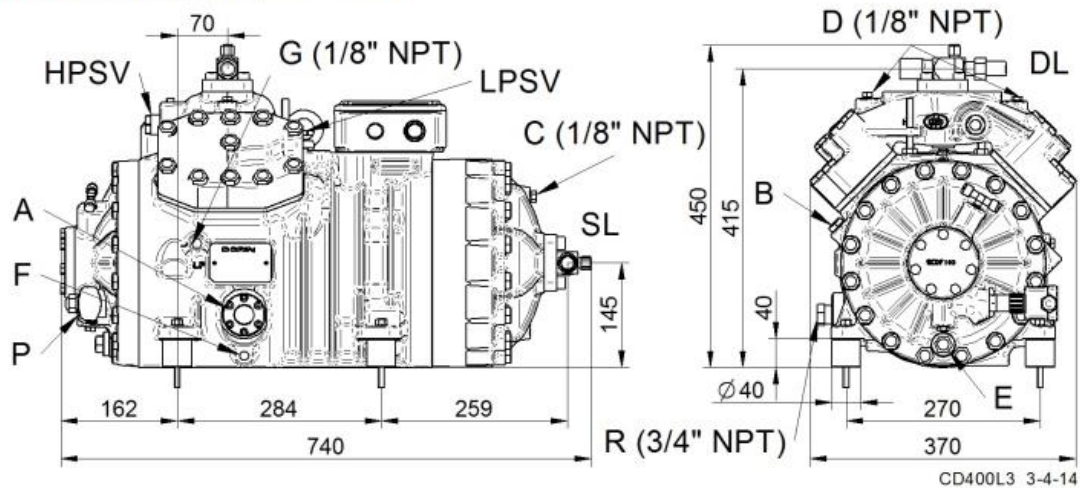
DL - Discharge service valve

SL - Suction service valve

LPSV - LP safety valve

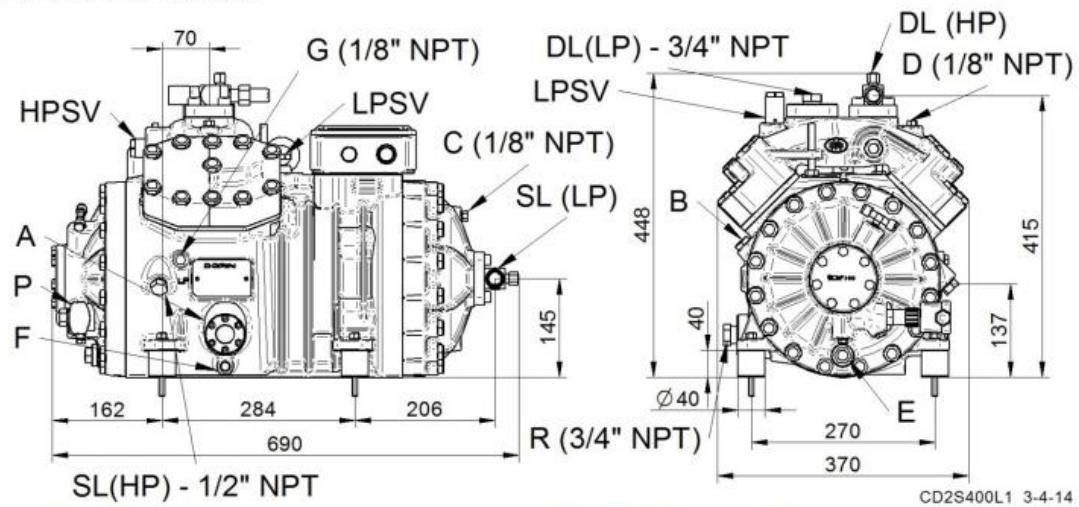
HPSV - HP safety valve

Overall dimensions [mm]: CD 3401H



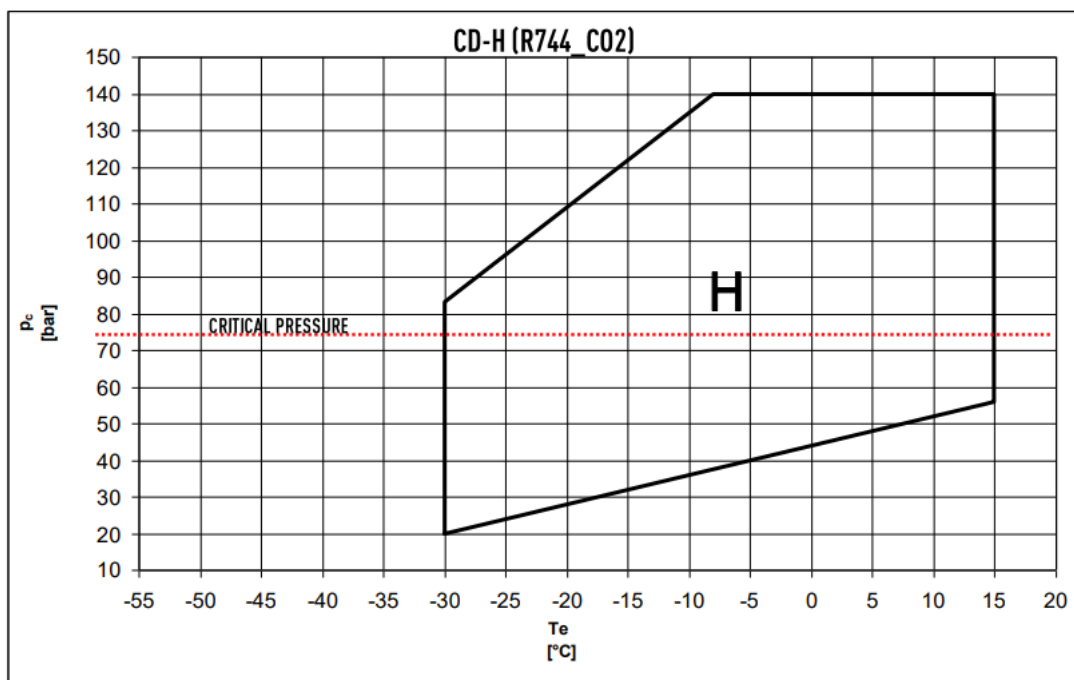
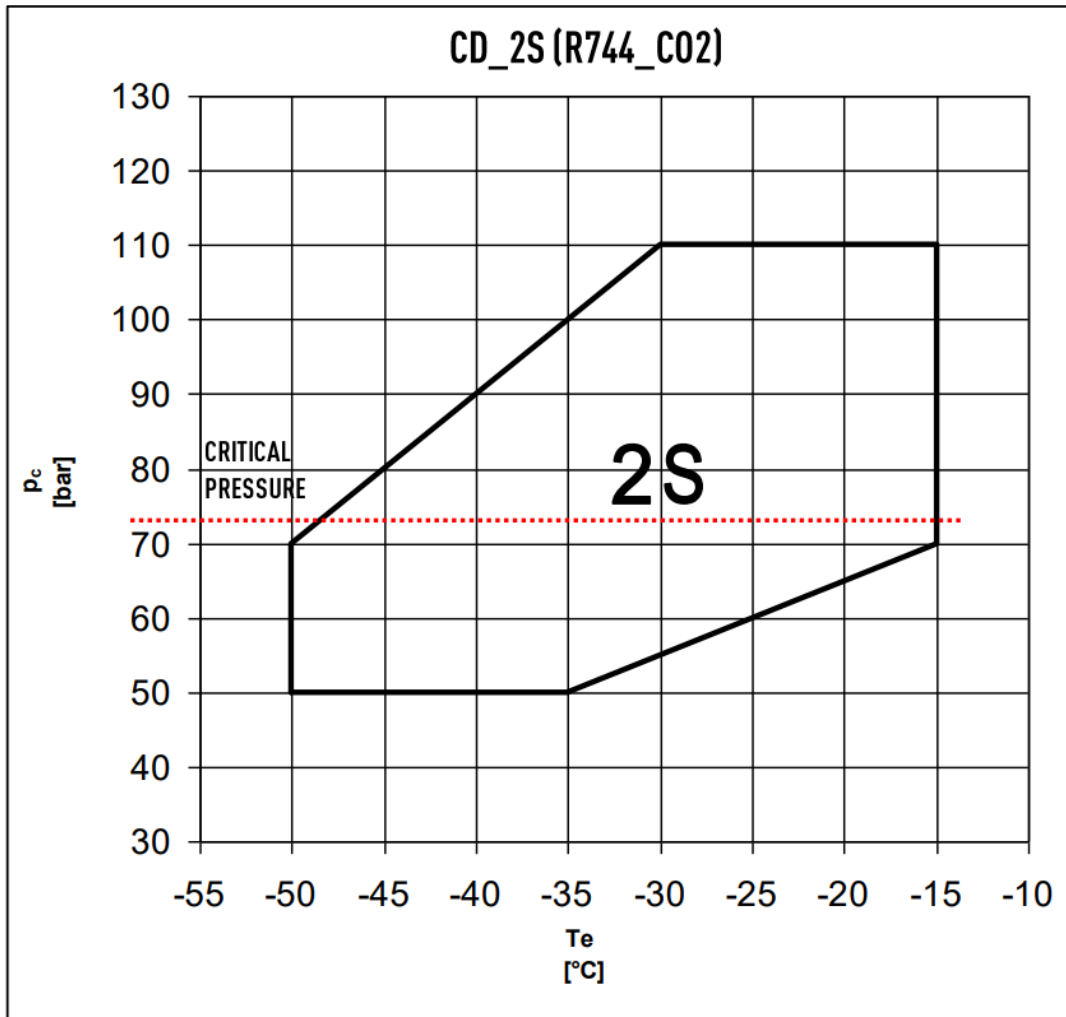
- A - Oil sight glass
- B - Oil charge plug
- C - Low pressure connection
- D - High pressure connection
- E - Oil drain plug
- F - Crankcase heater
- G - Oil return plug
- P - Oil differential pressure switch
- R - Oil level connection
- DL - Discharge service valve
- SL - Suction service valve
- LPSV - LP safety valve
- HPSV - HP safety valve

Overall dimensions [mm] CD2S 2500



- A - Oil sight glass
- B - Oil charge plug
- C - Low pressure connection
- D - High pressure connection
- E - Oil drain plug
- F - Crankcase heater
- G - Oil return plug
- P - Oil differential pressure switch
- R - Oil level connection
- DL(LP) - 1° stage discharge tap
- SL(LP) - 1° stage suction service valve
- DL(HP) - 2° stage discharge service valve
- SL(HP) - 2° stage suction tap
- LPSV - LP safety valve
- HPSV - HP safety valve

Appendix C: Selected Compressors Application Envelope.



Appendix D: Heat Exchangers Technical Specifications.

Heat exchanger 1

Technical specification

Brazed Plate Heat Exchanger



Project ref: AL22095RB - Permutadores CO2 - 03-06-2022
 Line ref: Permutador 1
 Model: **CBXP52-20M-F**
 Item Id: 3075013307
 No of units: 1

Page: 1(2)
 Date: 03-06-2022

	Hot side S1 -> S2	Cold Side S3 -> S4
Process data		
Capacity:	kW 4.4	
Fluid:	Carbon dioxide	Carbon dioxide
Duty type:	Liquid cooling	Gas heating
Mass flow rate:	kg/h 2,075	475
Inlet temperature:	°C 5,5	-25,0
Inlet pressure:	bara	52,00
Outlet temperature:	°C 3,1	-2,0
Total pressure drop calculated (allowed)	kPa 21,2 (200,00)	6,9 (200,00)
Velocity connections:	m/s 1,17	1,57
Specified fouling resistance*10000:	m²-K/W 20,9	

Heat exchanger specification	
Relative directions of fluids:	Countercurrent
Number of plates:	20
Channel volume:	dm³ 1,0 0,9
Number of circuits:	1 1
Design pressure at -196 °C	bar 90 90
Design pressure at 225 °C	bar 75 75
Design temperature (min/max):	°C -196 / 225
Pressure vessel code:	PED
Material Channel plates / Sealing:	ALLOY 316 / Cu
Connection S1 (Hot-In):	Soldering 1 1/8" (H21XP) ALLOY 304
Connection S2 (Hot-Out):	Soldering 1 1/8" (H21XP) ALLOY 304
Connection S3 (Cold-In):	Soldering 1 1/8" (H21XP) ALLOY 304
Connection S4 (Cold-Out):	Soldering 1 1/8" (H21XP) ALLOY 304
Unit dimensions (length x width x height):	mm 106 x 111 x 526
Net weight, empty / operating:	kg 7,5 / 8,50
Packed length x width x height:	mm 0 x 0 x 0
Packed weight:	kg 7,5

Heat exchanger 1

Technical specification

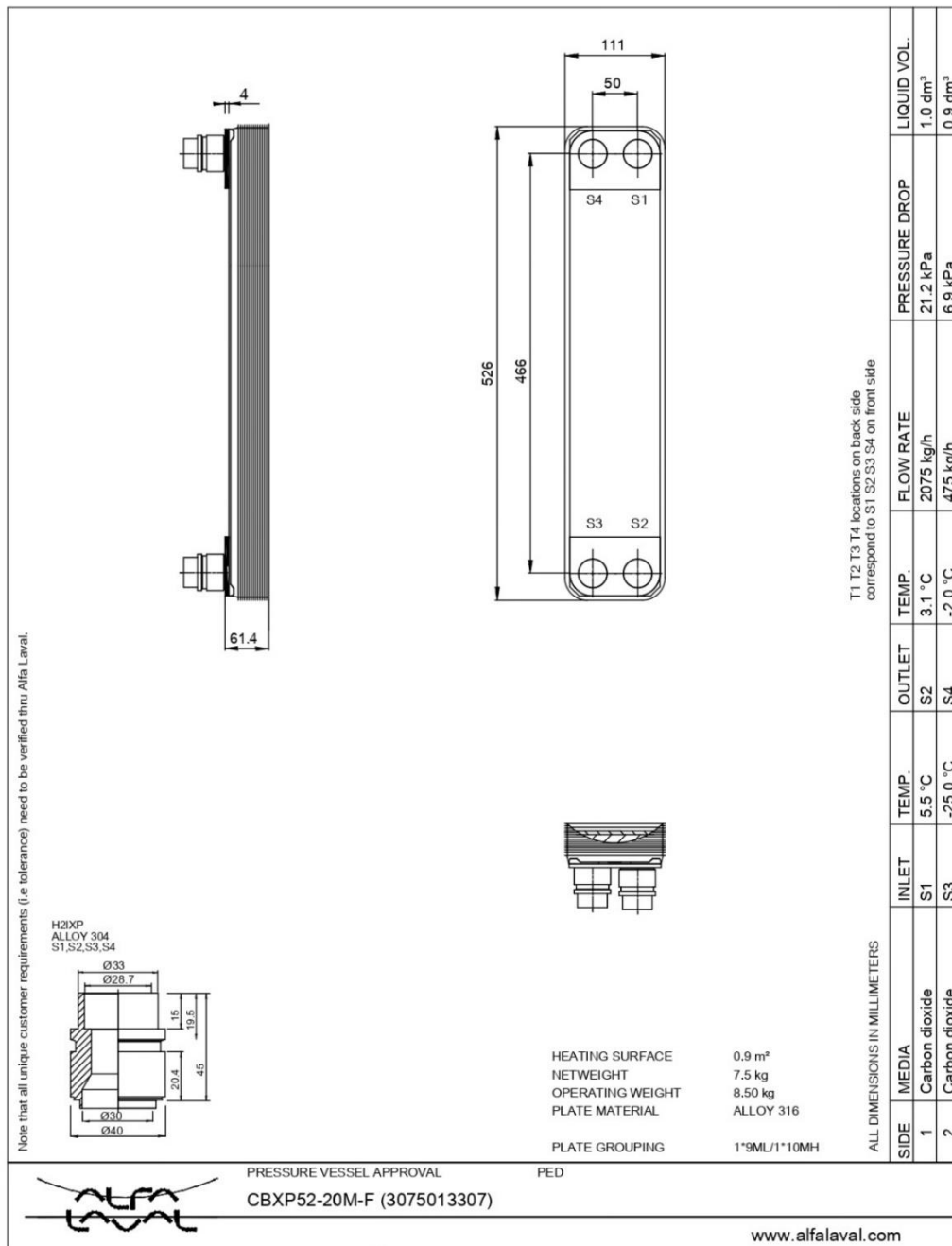
Brazen Plate Heat Exchanger



Project ref: AL22095RB - Permutadores CO2 - 03-06-2022
 Line ref: Permutador 1
 Model: **CBXP52-20M-F**
 Item Id: 3075013307
 No of units: 1

Page: 2(2)
 Date: 03-06-2022

Fluid properties		Hot side	Cold Side
Density (in/out):	kg/m ³	879,05/895,04	150,50/187,75
Specific heat capacity:	kJ/(kg·K)	3,18	1,49
Thermal conductivity:	W/(m·K)	0,104	0,015
Viscosity (in/out):	cP	0,0977/0,1019	0,0132/0,0165



Heat exchanger 2

Technical specification

Braze Plate Heat Exchanger



Project ref: AL22095RB - Permutadores CO2 - 03-06-2022
 Line ref: Permutador 2
 Model: **CBXP52-20M-F**
 Item Id: 3075013307
 No of units: 1

Page: 1(2)
 Date: 03-06-2022

		Hot side S1 -> S2	Cold Side S3 -> S4
Process data			
Capacity:	kW	10,2	
Fluid:		Carbon dioxide	Carbon dioxide
Duty type:		Gas cooling	Gas heating
Mass flow rate:	kg/h	475	1.600
Inlet temperature:	°C	70,0	-17,0
Inlet pressure:	bara	52,00	52,00
Outlet temperature:	°C	20,0	-2,0
Total pressure drop calculated (allowed)	kPa	8,2 (200,00)	69,3 (200,00)
Velocity connections:	m/s	2,39	4,97
Specified fouling resistance*10000:	m ² -K/W	31,8	

Heat exchanger specification			
Relative directions of fluids:		Countercurrent	
Number of plates:		20	
Channel volume:	dm ³	1,0	0,9
Number of circuits:		1	1
Design pressure at -196 °C	bar	90	90
Design pressure at 225 °C	bar	75	75
Design temperature (min/max):	°C	-196 / 225	
Pressure vessel code:		PED	
Material Channel plates / Sealing:		ALLOY 316 / Cu	
Connection S1 (Hot-In):		Soldering 1 1/8" (H21XP) ALLOY 304	
Connection S2 (Hot-Out):		Soldering 1 1/8" (H21XP) ALLOY 304	
Connection S3 (Cold-In):		Soldering 1 1/8" (H21XP) ALLOY 304	
Connection S4 (Cold-Out):		Soldering 1 1/8" (H21XP) ALLOY 304	
Unit dimensions (length x width x height):	mm	106 x 111 x 526	
Net weight, empty / operating:	kg	7,5 / 7,77	
Packed length x width x height:	mm	0 x 0 x 0	
Packed weight:	kg	7,5	

Heat exchanger 2

Technical specification

Braze Plate Heat Exchanger

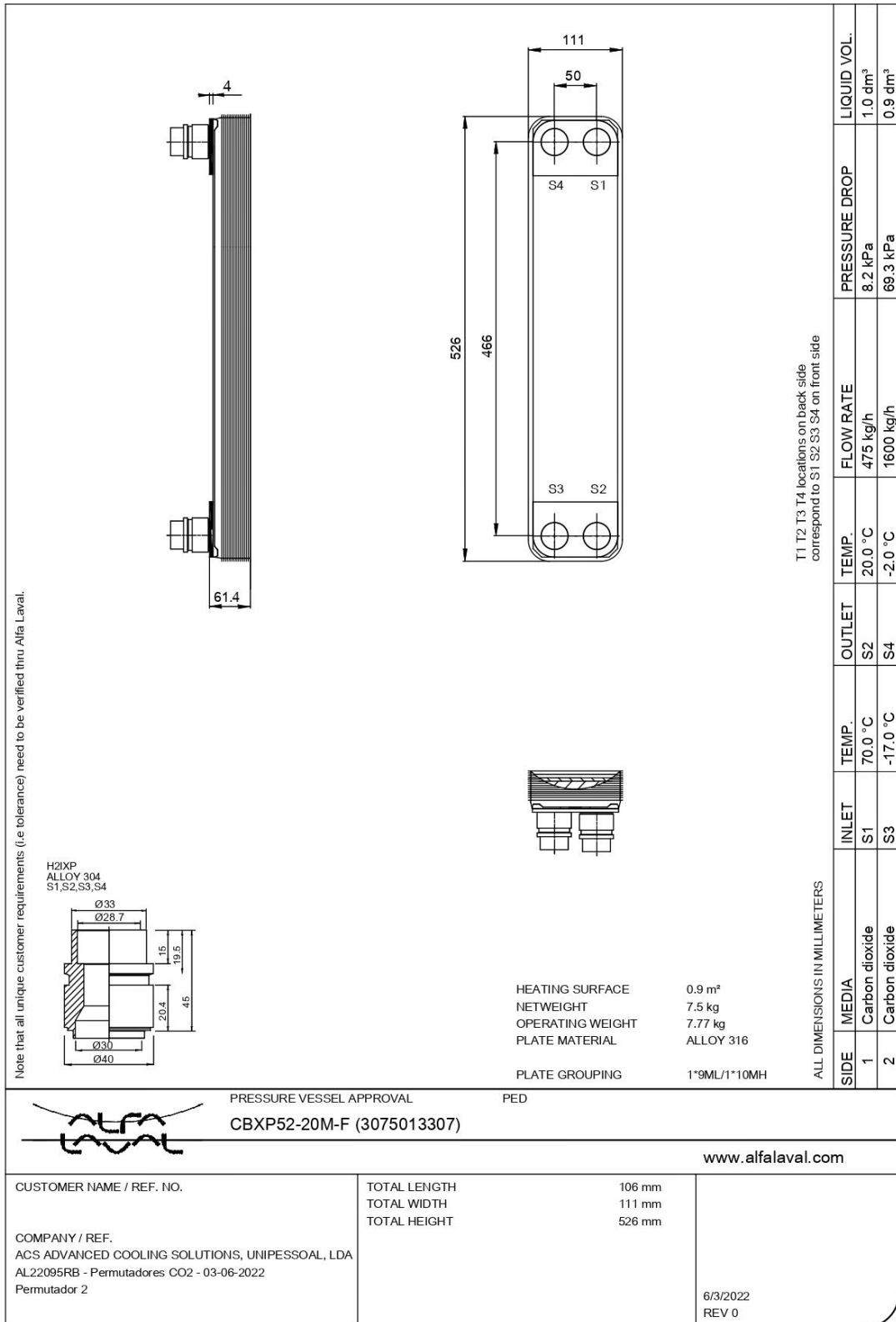


Project ref: AL22095RB - Permutadores CO2 - 03-06-2022
 Line ref: Permutador 2
 Model: **CBXP52-20M-F**
 Item Id: 3075013307
 No of units: 1

Page: 2(2)
 Date: 03-06-2022

Fluid properties		Hot side	Cold Side
Density (in/out):	kg/m ³	98,49/146,32	159,86/184,71
Specific heat capacity:	kJ/(kg·K)	1,86	1,56
Thermal conductivity:	W/(m·K)	0,025	0,016
Viscosity (in/out):	cP	0,0363/0,0210	0,0142/0,0165

Heat exchanger 2



Heat exchanger 3

Technical specification



Project ref: AL22095RB - Permutadores CO2 - 03-06-2022
 Line ref: Permutador 3
 Model: **AXP27-14L-F**
 Item Id: 3287174684
 No of units: 1

Page: 1(2)
 Date: 03-06-2022

		Hot side S1 -> S2	Cold Side S3 -> S4
Process data			
Capacity:	kW	16,5	
Fluid:		Carbon dioxide	Carbon dioxide
Duty type:		Gas cooling	Gas heating
Mass flow rate:	kg/h	4,375	2,300
Inlet temperature:	°C	41,0	5,0
Inlet pressure:	bara	100,00	50,00
Outlet temperature:	°C	38,6	15,0
Total pressure drop calculated (allowed)	kPa	53,9 (200,00)	33,6 (200,00)
Velocity connections:	m/s	8,37	6,47
Specified fouling resistance*10000:	m ² -K/W	0,5	

Heat exchanger specification			
Relative directions of fluids:		Countercurrent	
Number of plates:		14	
Channel volume:	dm ³	0,4	0,3
Number of circuits:		1	1
Design pressure at -20 °C	bar	130	130
Design pressure at 150 °C	bar	130	130
Design temperature (min/max):	°C	-20 / 150	
Pressure vessel code:		PED	
Material Channel plates / Sealing:		ALLOY 316 / Cu	
Connection S1 (Hot-In):		Soldering 1 1/8" (H21XP) ALLOY 304	
Connection S2 (Hot-Out):		Soldering 1 1/8" (H21XP) ALLOY 304	
Connection S3 (Cold-In):		Soldering 1 1/8" (H21XP) ALLOY 304	
Connection S4 (Cold-Out):		Soldering 1 1/8" (H21XP) ALLOY 304	
Unit dimensions (length x width x height):	mm	150 x 160 x 362	
Net weight, empty / operating:	kg	22,2 / 22,35	
Packed length x width x height:	mm	0 x 0 x 0	
Packed weight:	kg	22,2	

Heat exchanger 3

Technical specification

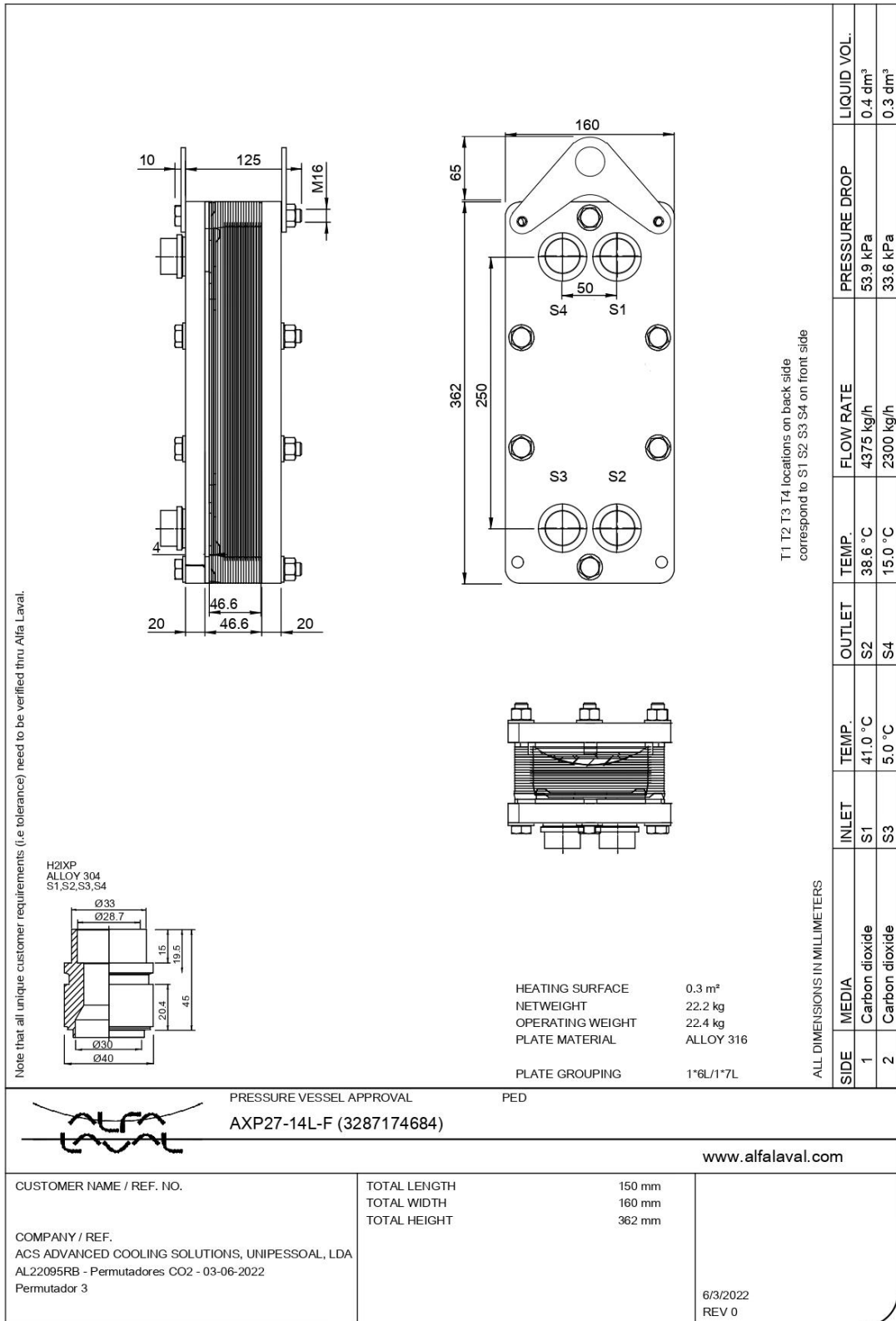


Project ref: AL22095RB - Permutadores CO2 - 03-06-2022
 Line ref: Permutador 3
 Model: **AXP27-14L-F**
 Item Id: 3287174684
 No of units: 1

Page: 2(2)
 Date: 03-06-2022

Fluid properties		Hot side	Cold Side
Density (in/out):	kg/m ³	259,44/253,54	176,31/145,99
Specific heat capacity:	kJ/(kg·K)	5,61	2,54
Thermal conductivity:	W/(m·K)	0,023	0,018
Viscosity (in/out):	cP	0,0266/0,0259	0,0178/0,0198

Heat exchanger 3



Appendix E: Software Report of the Reference Refrigeration Unit for Project Conditions.



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PAGE 1

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All data subject to change

Calculation - 21-BOOSTER

Input

Configuration	Parallel compression
MT	
Refrigerant	R744
Required capacity (evaporator)	100.0 kW
Evaporating temperature	-10.0 °C
Suction line outlet superheat	5.0 K
Evaporator outlet superheat	5.0 K
Gas cooler outlet temperature	41.0 °C
Gas cooler pressure	101.0 bar
Receiver pressure	42.0 bar
• Compressor MT 1	
Model	CD 2500H
N°	3
Voltage / phases / frequency	380-420 V / 3 / 50 Hz STAR
Inverter	False
Voltage / phases / frequency (Inverter)	380-420 V / 3 / 50 Hz DOL
• Compressor MT 2	
Model	CD 1400H
N°	1
Voltage / phases / frequency	380-420 V / 3 / 50 Hz STAR
Inverter	True
Voltage / phases / frequency (Inverter)	380-420 V / 3 / 50 Hz DOL
• Parallel compressor	
Model	CD 2400H
N°	1
Inverter	True
Voltage / phases / frequency	380-420 V / 3 / 50 Hz STAR
Sizing	90 %
Voltage / phases / frequency (Inverter)	380-420 V / 3 / 50 Hz DOL
LT	
Refrigerant	R744



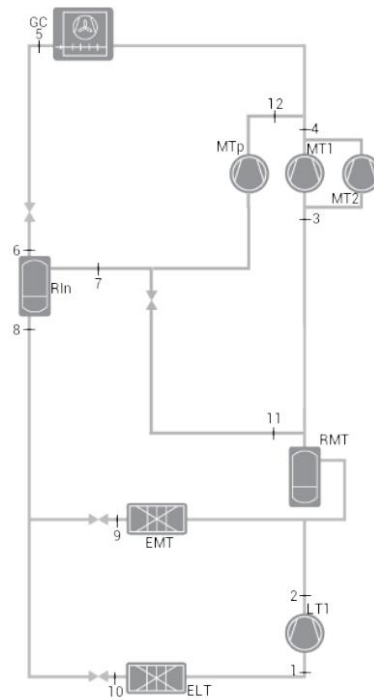
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PAGE 2

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All data subject to change

Required capacity (evaporator)	30.0 kW
Evaporating temperature	-32.0 °C
Suction line outlet superheat	25.0 K
Evaporator outlet superheat	5.0 K
• Compressor LT 1	
Model	CDS401B
N°	2
Voltage / phases / frequency	380-420 V / 3 / 50 Hz STAR
Inverter	True
Voltage / phases / frequency (Inverter)	



GC: Gas cooler (Heat rejection air or water heat exchanger)
MT2: Compressor MT 2
MT1: Compressor MT 1 (Compresses vapour from MT to high pressure side)
MTp: Parallel compressor (Compresses vapour from intermediate to high pressure side)
Rln: Intermediate pressure receiver (Refrigerant charge storage tank)
RMT: Receiver MT (MT booster receiver)
EMT: Evaporator MT (MT display cabinets and cold rooms)
LT1: Compressor LT 1 (Compresses vapour from LT to MT pressure side)
ELT: Evaporator LT (LT display cabinets and cold rooms)

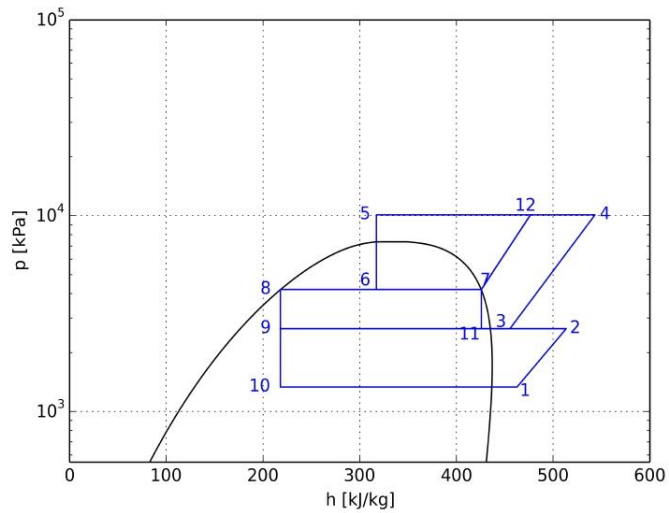


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PAGE 3

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All data subject to change



Performance calculation

System

Cooling capacity (MT)	100.00 kW
Cooling capacity (LT)	30.00 kW
Absorbed power	99.30 kW
COP	1.31
Reserve capacity MT	16.6 %
Reserve capacity LT	2.2 %
Reserve capacity parallel	-12.7 %

Compressors (MT)

Cooling capacity	100.00 kW
Cooling capacity at compressor	139.56 kW
Absorbed power	65.52 kW
Mass flow (LP)	2328.5 kg/h
Suction pressure	26.5 bar
Suction temperature	4.9 °C



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 INNOVATION

PAGE 4

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All data subject to change

Suction enthalpy	455.4 kJ/kg
Discharge pressure	101.0 bar
Discharge temperature	128.2 °C
Discharge enthalpy	543.3 kJ/kg
MT 1	
Compressor 1	ON (100.0 %)
Compressor 2	ON (100.0 %)
Compressor 3	ON (100.0 %)
MT 2	
Compressor 1	ON (Inverter: 19.9 Hz)
Parallel compressors (MT)	
Absorbed power	26.22 kW
Mass flow (MP)	1664.9 kg/h
Suction pressure	42.0 bar
Suction temperature	7.3 °C
Suction enthalpy	425.8 kJ/kg
Discharge pressure	101.0 bar
Discharge temperature	83.7 °C
Discharge enthalpy	476.6 kJ/kg
MT prl	
Compressor 1	ON (Inverter: 65.7 Hz)
@ Evaporator (MT)	
Cooling capacity	100.00 kW
Mass flow	1604.5 kg/h
Inlet pressure	26.5 bar
Inlet temperature	-10.0 °C
Inlet enthalpy	218.0 kJ/kg
Inlet quality	0.16
Outlet pressure	26.5 bar
Outlet temperature	-5.0 °C
Outlet enthalpy	442.4 kJ/kg
@ Intermediate pressure receiver (MT)	
Inlet enthalpy	317.2 kJ/kg



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All data subject to change

Input mass flow	3993.4 kg/h
Liquid outlet enthalpy	218.0 kJ/kg
Output mass flow	2086.4 kg/h
Vapour outlet enthalpy	425.8 kJ/kg
Output vapour mass flow	1907.0 kg/h
Mass flow (Parallel compressor)	1664.9 kg/h
Mass flow (Flash gas by-pass)	242.1 kg/h

@ Gas cooler (MT)

Heating capacity	219.92 kW
Mass flow (HP)	3993.4 kg/h
Inlet pressure	101.0 bar
Inlet temperature	108.2 °C
Inlet enthalpy	515.5 kJ/kg
Outlet pressure	101.0 bar
Outlet temperature	41.0 °C
Outlet enthalpy	317.2 kJ/kg

Air conditioning (MT)

Air conditioning load	0.00 kW
Air conditioning mass flow	0.0 kg/h

Compressors (LT)

Cooling capacity	30.00 kW
Cooling capacity at compressor	32.76 kW
Absorbed power	7.55 kW
Mass flow (LP)	481.9 kg/h
Suction pressure	13.3 bar
Suction temperature	-7.0 °C
Suction enthalpy	462.7 kJ/kg
Discharge pressure	26.5 bar
Discharge temperature	57.5 °C
Discharge enthalpy	513.5 kJ/kg

LT 1

Compressor 1	ON (100.0 %)
Compressor 2	ON (Inverter: 67.3 Hz)

@ Evaporator (LT)



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All data subject to change

Cooling capacity	30.00 kW
Mass flow	481.9 kg/h
Inlet pressure	13.3 bar
Inlet temperature	-32.0 °C
Inlet enthalpy	218.0 kJ/kg
Inlet quality	0.29
Outlet pressure	13.3 bar
Outlet temperature	-27.0 °C
Outlet enthalpy	442.1 kJ/kg

Appendix F: Software Reports used to Calculate the COP of the Projected Refrigeration unit for Project Conditions.



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INNOVATION

Model: CD2S2500 - PAGE 1

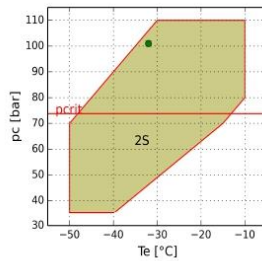
02/07/2022 - Dorin Software v. 21.10

All data subject to change

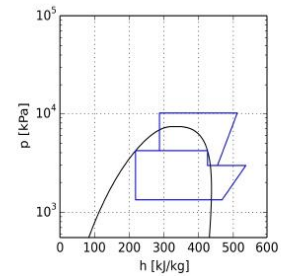
Calculation - CD_2S

Input

Model	CD2S2500
Refrigerant	R744_CO2
Configuration	Open flash tank
Operating mode	Transcritical
Evaporating temperature	-32.0 °C
Gas cooler pressure	101.0 bar
Superheat	30.0 K
Evap. outlet superheat	5.0 K
Gas cooler outlet temperature	34.5 °C
Voltage / phases / frequency	380-420 V / 3 / 50 Hz
Intercooler outlet temperature	20.0 °C
Intermediate pressure at receiver	42.0 bar



2S = Standard application
- = Critical pressure 73,77 bar
• = Dew temperature





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INNOVATION

Model: CD2S2500 - PAGE 2

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All data subject to change

Performance calculation

Cooling capacity	17.24 kW
Cooling capacity at compressor	19.21 kW
Absorbed power	15.67 kW
Absorbed current	30.4 A
Gas cooler capacity	25.91 kW
COP	1.10
Mass flow LP	277.0 kg/h
Mass flow MP	138.4 kg/h
Mass flow HP	415.4 kg/h
Suction pressure LP	13.3 bar
Suction temperature LP	-2.0 °C
Suction enthalpy LP	467.7 kJ/kg
Discharge pressure LP	29.7 bar
Discharge temperature LP	81.9 °C
Discharge enthalpy LP	536.3 kJ/kg
Suction pressure HP	29.7 bar
Suction temperature HP	8.5 °C
Suction enthalpy HP	454.4 kJ/kg
Discharge pressure HP	101.0 bar
Discharge temperature HP	105.4 °C
Discharge enthalpy HP	511.5 kJ/kg
Intercooler capacity	5.19 kW



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Model: CD2S2500 - PAGE 3

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All data subject to change

Technical data

No. cylinders LP	2.0	
No. cylinders HP	2.0	
Displacement LP @ 50Hz	11.65	[m³/h]
Displacement HP @ 50 Hz	6.92	[m³/h]
Displacement LP @ 60 Hz	13.98	[m³/h]
Displacement HP @ 60 Hz	8.3	[m³/h]
Volume ratio LP / HP	1.68	
Suction valve (socket welding)	22.0	[mm]
Suction valve (butt welding)	28.0	[mm]
Discharge valve (socket welding)	22.0	[mm]
Discharge valve (butt welding)	28.0	[mm]
Oil charge	2.5	[L]
Net weight	175.0	[kg]



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PAGE 1

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All data subject to change

Calculation - 00-MTTRANS

Input

Configuration	Parallel compression
MT	
Refrigerant	R744
Required capacity (evaporator)	100.0 kW
Evaporating temperature	-10.0 °C
Suction line outlet superheat	5.0 K
Evaporator outlet superheat	5.0 K
Gas cooler outlet temperature	41.0 °C
Gas cooler pressure	101.0 bar
Receiver pressure	42.0 bar
• Compressor MT 1	
Model	CD 3401H
N°	1
Voltage / phases / frequency	380-420 V / 3 / 50 Hz STAR
Inverter	False
Voltage / phases / frequency (Inverter)	380-420 V / 3 / 50 Hz DOL
• Compressor MT 2	
Model	CD 2000H
N°	1
Voltage / phases / frequency	380-420 V / 3 / 50 Hz STAR
Inverter	True
Voltage / phases / frequency (Inverter)	380-420 V / 3 / 50 Hz DOL
• Parallel compressor	
Model	CD 2400H
N°	1
Inverter	True
Voltage / phases / frequency	380-420 V / 3 / 50 Hz STAR
Sizing	90 %
Voltage / phases / frequency (Inverter)	380-420 V / 3 / 50 Hz DOL

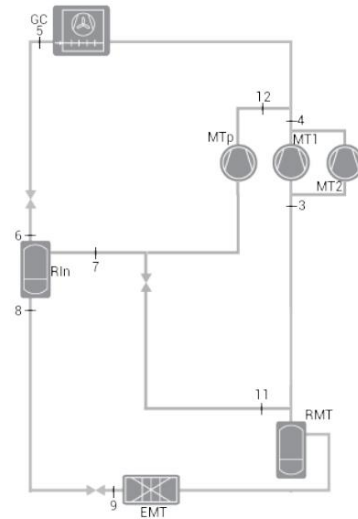


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All data subject to change



GC: Gas cooler (Heat rejection air or water heat exchanger)

MT2: Compressor MT 2

MT1: Compressor MT 1 (Compresses vapour from MT to high pressure side)

MTp: Parallel compressor (Compresses vapour from intermediate to high pressure side)

Rln: Intermediate pressure receiver (Refrigerant charge storage tank)

RMT: Receiver MT (MT booster receiver)

EMT: Evaporator MT (MT display cabinets and cold rooms)

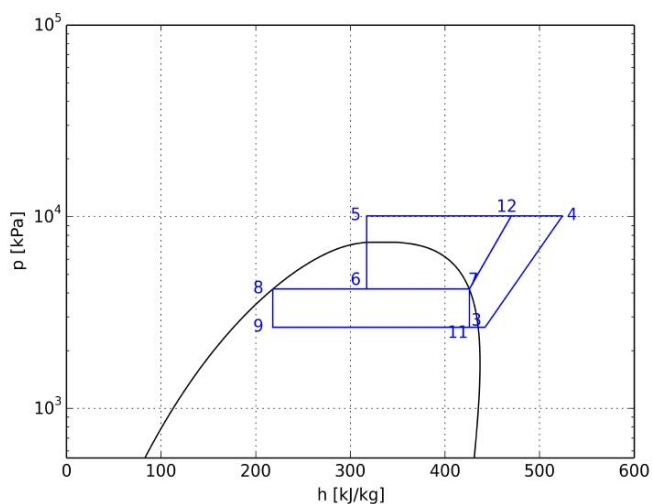


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PAGE 3

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All data subject to change



Performance calculation

System

Cooling capacity (MT)	100.00 kW
Absorbed power	61.62 kW
COP	1.62
Reserve capacity MT	12.1 %
Reserve capacity parallel	12.1 %

Compressors (MT)

Cooling capacity	100.00 kW
Cooling capacity at compressor	100.00 kW
Absorbed power	41.47 kW
Mass flow (LP)	1604.5 kg/h
Suction pressure	26.5 bar
Suction temperature	-5.0 °C
Suction enthalpy	442.4 kJ/kg
Discharge pressure	101.0 bar



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All data subject to change

Discharge temperature	114.2 °C
Discharge enthalpy	524.2 kJ/kg
MT 1	
Compressor 1	ON (100.0 %)
MT 2	
Compressor 1	ON (Inverter: 50.2 Hz)
Parallel compressors (MT)	
Absorbed power	20.15 kW
Mass flow (MP)	1466.6 kg/h
Suction pressure	42.0 bar
Suction temperature	7.3 °C
Suction enthalpy	425.8 kJ/kg
Discharge pressure	101.0 bar
Discharge temperature	80.2 °C
Discharge enthalpy	470.2 kJ/kg
MT prl	
Compressor 1	ON (Inverter: 53.9 Hz)
@ Evaporator (MT)	
Cooling capacity	100.00 kW
Mass flow	1604.5 kg/h
Inlet pressure	26.5 bar
Inlet temperature	-10.0 °C
Inlet enthalpy	218.0 kJ/kg
Inlet quality	0.16
Outlet pressure	26.5 bar
Outlet temperature	-5.0 °C
Outlet enthalpy	442.4 kJ/kg
@ Intermediate pressure receiver (MT)	
Inlet enthalpy	317.2 kJ/kg
Input mass flow	3071.1 kg/h
Liquid outlet enthalpy	218.0 kJ/kg
Output mass flow	1604.5 kg/h
Vapour outlet enthalpy	425.8 kJ/kg



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All data subject to change

Output vapour mass flow	1466.6 kg/h
Mass flow (Parallel compressor)	1466.6 kg/h
Mass flow (Flash gas by-pass)	0.0 kg/h
@ Gas cooler (MT)	
Heating capacity	154.53 kW
Mass flow (HP)	3071.1 kg/h
Inlet pressure	101.0 bar
Inlet temperature	96.8 °C
Inlet enthalpy	498.4 kJ/kg
Outlet pressure	101.0 bar
Outlet temperature	41.0 °C
Outlet enthalpy	317.2 kJ/kg
Air conditioning (MT)	
Air conditioning load	0.00 kW
Air conditioning mass flow	0.0 kg/h