

VŠB – Technical University of Ostrava  
Faculty of Mechanical Engineering  
Department of Power Engineering

**Diploma Thesis**  
**Design of Cooling System for Low-  
Temperature Drying Unit for Treatment of  
Raw Natural Gas**

Návrh systému chlazení pro nízkoteplotní  
sušící jednotku určenou k úpravě těžného  
zemního plynu

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## Diploma Thesis Assignment

Student: **Diego Emanuel García**

Study Programme: N2301 Mechanical Engineering

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Title: **Design of Cooling System for Low-temperature Drying Unit Designed for Treatment of Mined Natural Gas**  
**Návrh systému chlazení pro nízkoteplotní sušicí jednotku určenou k úpravě těžného zemního plynu**

The thesis language: English

### Description:

Create a compressor cooling unit design for treatment of mined natural gas. The cooling unit will be part of the low temperature drying system. Process the design for maximum and minimum natural gas flow, for specified refrigerants and for a specified cooling temperature control range. Determine the temperature of the natural gas at the inlet to the evaporator by calculation.

### Entered values:

- maximum natural gas flow 25 000 Nm<sup>3</sup>/day
- minimum natural gas flow 10 000 Nm<sup>3</sup>/day
- refrigerant R449A and refrigerant R404A
- temperature control range from -19°C to -13°C

### Content of the diploma thesis:

1. Capacitive and balance calculation of the system
2. Selection of compressor and expansion valve
3. Design of condenser
4. Design of evaporator
5. Diagram of cooling unit
6. Technical drawing of the evaporator

### References:

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MOKHATAB, Saeid, William A. POE a John Y. MAK, [2015]. Handbook of natural gas transmission and processing: principles and practices. Third edition. Amsterdam: Gulf Professional Publishing is an imprint of Elsevier. ISBN 0128014997.

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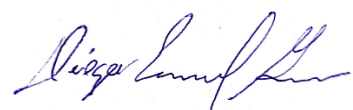


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In the thesis, I used internal information about the technical parameters of the gas treatment system obtained from the company MND a.s, the company agrees to their disclosure.

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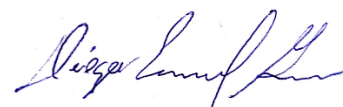
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## **ANNOTATION OF DIPLOMA THESIS**

GARCIA DIEGO EMANUEL. *Design of Cooling System for Low-Temperature Drying Unit for Treatment of Raw Natural Gas: Diploma Thesis*. Ostrava: VŠB – Technical University of Ostrava, Faculty of Mechanical Engineering, Department of Power Engineering, 2020, 93 p. Head of the diploma thesis: Ing. Petr Pavlík, Ph.D.

This master's thesis deals with the design of a cooling system for a low temperature drying unit for the treatment of natural gas. The theoretical part presents information about principles of natural gas, refrigeration cycles and heat exchangers. The practical part describes the methods and calculations to design the cooling system unit, selection of the compressor and thermal expansion valve, design of evaporator and design of condenser by comparing the use of refrigerant R404A and R449A. Lastly, it is shown the specified technical parameters and technical drawings.

**Keywords:** Natural gas, cooling system, compressor, evaporator, refrigeration cycle

## **ANOTACE DIPLOMOVÉ PRÁCE**

GARCIA DIEGO EMANUEL. *Návrh systému chlazení pro nízkoteplotní sušící jednotku určenou k úpravě těžného zemního plynu: Diplomová práce*. Ostrava: VŠB – Technická univerzita Ostrava, Fakulta strojní, Katedra energetiky, 2020, 93 s. Vedoucí práce: Ing. Petr Pavlík, Ph.D.

Tato diplomová práce se zabývá návrhem chladicího systému určeného pro nízkoteplotní sušící jednotku pro úpravu vlastností zemního plynu. Teoretická část se věnuje informacím ohledně principů úpravy zemního plynu, chladicím cyklům a výměníkům tepla. Praktická část se poté věnuje návrhům a výpočtům celého chladicího systému stejně tak jako jednotlivým částem, kterými jsou kompresor chladicího systému, expanzní ventil, výparník a kondenzátor. Návrhy a výpočty jsou zpracovány pro chladiva R404A a R449A. Práce obsahuje také specifikaci technických parametrů jednotlivých částí a výkresovou dokumentaci.

**Klíčová Slova:** Zemní plyn, chladicí systém, kompresor, výparník, chladicí cyklus.

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## List of used signs, symbols and Indices

Quantity	Name	Unit
$\mu_b$	Dynamic viscosity of the mixture at the bulk temperature	$kg \cdot m^{-1} \cdot s^{-1}$
$\mu_{pt}$	Velocity through the ports	$m \cdot s^{-1}$
$A$	Area	$m^2$
$B$	Nominal width	$m$
$b$	Spacing of plate	$m$
$Bo$	Boiling number	-
$C$	Specific energy consumption	$kWh.m^{-3}$
$C.E$	Cooling effect	$kJ.kg^{-1}$
$Co$	Convection number, Martinelli parameter	-
$COP$	Coefficient of Performance	-
$C_p$	Specific heat capacity	$kJ.kg^{-1}K^{-1}$
$D_c$	Equivalent diameter	$m$
$d_{pt}$	Port diameter	$m$
$e_c$	Specific kinetic energy	$kJ.kg^{-1}$
$e_p$	Specific potential energy	$kJ.kg^{-1}$
$h$	Specific enthalpy energy	$kJ.kg^{-1}$
$Ha$	Altitude of location	$m$
$j_f$	Friction factor for turbulent flow	-
$k$	Conductivity	$W \cdot m^{-1} \cdot K^{-1}$
$L$	Effective length of flow	$m$
$LMTD$	Log-Mean Temperature Difference	$^{\circ}C$
$L_w$	Distance	$m$
$\dot{m}$	Mass flow	$Kg.s^{-1}$
$m_f$	Molar fraction	%
$N$	Number	-
$Nu$	Nusselt number	-
$P$	Pressure	$KPa, bar$
$P_{comp}$	Technical power required by the compressor	$kW$
$P_{el}$	Electrical power	$kW$
$Pr$	Prandtl number	-
$Q$	Heat	$kJ/kg$
$\dot{Q}$	Heat transfer rate or power	$kW$
$Q_{NB}$	Boiling heat transfer value	$kW \cdot m^{-2}$
$R$	Universal gas constant	$J.mol^{-1}K^{-1}$
$R_c$	Pressure ratio	-
$Re$	Reynolds number	-
$R_f$	Fouling factor	$m^2 \cdot ^{\circ}C \cdot W^{-1}$
$S$	Specific entropy	$kJ.K^{-1}.kg^{-1}$
$tk$	Thickness of the plate	$m$
$F_s$	Fluid surface parameter	-
$T$	Temperature	$^{\circ}C, K$
$U$	Overall heat transfer coefficient	$W \cdot m^{-2} \cdot ^{\circ}C^{-1}$
$V$	Volume	$m^3$
$\nu$	Kinematic viscosity	$m^2 \cdot s^{-1}$
$\dot{V}$	Volumetric flow	$m^3 \cdot h^{-1}$

$V_{ch}$	Channel velocity	$m \cdot s^{-1}$
$\dot{W}$	Work energy	$kW$
$x$	Quality of the refrigeration process	-
$z$	Compressibility factor	-
$\alpha$	Heat transfer coefficient	$W \cdot m^{-2} \cdot K^{-1}$
$\Delta h$	Difference of enthalpy	-
$\Delta p$	Pressure drop	$kPa$
$\eta$	Efficiency	%
$\lambda$	Thermal conductivity of the plate	$W \cdot m^{-1} \cdot ^\circ C^{-1}$
$\rho$	Density	$kg \cdot m^{-3}$
$\Phi$	Relative humidity	%

## Indices

Index	Name
$b$	bulk temperature
$air$	air
$CB$	convective Boiling Regime
$ch$	channels
$ch$	cross-sectional area
$cold$	cold fluid
$comp$	compressor
$cond$	condensation
$dew$	dew point
$eff$	effective plate area
$evap$	evaporation
$hot$	Hot fluid
$in$	inlet
$ise$	Isentropic
$l$	liquid state
$max$	maximum
$min$	minimum
$NB$	nucleate boiling regime
$out$	outlet
$p$	plates
$pass$	passes
$prom$	mean
$pt$	ports
$ref$	refrigerant
$tot$	total
$TP$	two phase
$v$	vapor phase
$w$	wall temperature

## Abbreviations

Abbreviation	Importance
AF	Altitude Factor
AHRI	Air-Conditioning Heating and Refrigeration Institute.
ASHRAE	American Society of Heating, Refrigerating and Air-Conditioning Engineers.
COP	Coefficient of Performance
CS2	Cool Selector 2
DT	Difference of temperature
EES	Engineering Equation Solver
EN	European standard
EU	European Union
GWP	Global Warming Potential
HCFC	Hydrochlorofluorocarbons
HFC	Hydrofluorocarbons
HFO	Hydrofluoro-olefin
LMTD	Logarithmic mean temperature difference
LTS	Low Temperature Separator
MND	Moravské naftové doly
NG	Natural Gas
ODF	Outside Diameter Female
SAE	Society of Automotive Engineer
SPDT	Single Pole Double Throw
TEV	Thermal Expansion Valve
THR	Total Heat Rejection
TR	Ton of Refrigeration

# Introduction

As the demand for natural gas increases, the development of engineering has been involved in the need to return to the study of the processes of separation and dehydration of natural gas, to meet the necessary requirements of quality standards for the commercialization of natural gas. Besides, the natural gas industry faces the growing challenges of bringing sustainability between energy efficiency in cost operations and accountability for international environmental standards.

This work was developed in partnership with the company Moravské Naftové Doly (MND), which represents an important figure of the industry of natural gas of this country. Relying on low-temperature separation as one of the principal processes for treatment of natural gas, the purpose of the thesis is to design a low temperature drying unit, with the condition that at the exit, the dispatch pipeline, and the properties of the natural gas at the outlet shall be guaranteed the requirement described further.

The low temperature separation is described as a process primarily to remove moisture and to extract gas condensate. These units could use refrigeration technology or Joule-Thomson effect. It has been proofed that refrigeration technology is more efficient when gas conditions are too low to provoke a Joules Thomson effect though. Nowadays refrigeration systems are on the trend to look for solutions in the use of more refrigerants less contaminant and less affecting the environment, which has been imposed by regulations. Moving from the use of one refrigerant to another represents a good opportunity to also look for the optimization of production and the lowering of energy demand.

To summarize the low-temperature separation process, in the first step, a heat exchange occurs in the recuperator to make sure that temperature gas on the outlet meets the requirement. The water is separated from the gas mixed previously with ethanol and finally occurs the cooling down of the gas mixture with an auxiliary refrigeration system. However, the aim of this thesis is the design of the refrigeration system.

The thesis is divided in two parts, the theoretical part and the practical. Chapter 1 presents a background of principles of natural gas, its properties, and processes. Chapter 2 shows a description of the refrigeration cycles, steps, and main components. Chapter 3 present the analysis of the evaporator and the condenser as a heat exchanger. Chapter 4 shows an introduction of the practical parts and its data, to enter then the process of selection and designing of each component, shown in the later chapters 5,6,7,8 as the analysis of compressor, thermal expansion valve, evaporator, and condenser respectively, using refrigerant R404A. Then chapter 9 shows the design and selection of equipment using R449A. Finally it is present some recommendation, conclusion and technical drawing of the system designed.

# 1 Principle of natural gas

## 1.1 Natural gas

In order to start with the design, first, we should know what is natural gas? Natural gas can be defined as a mixture of gases, of variable composition (with methane as a predominant component) found at relatively high pressures in geological formations of the Earth's crust. [1]

The main use of natural gas is fuel. Therefore heaviest hydrocarbons, such as butane and propane, are extracted before using natural gas as fuel. Natural gas consists of low boiling hydrocarbons; the main component, approximately 85 %, is methane. In smaller proportions, ethane is present, approximately 10 % and propane that can reach percentages of up to 3 %. In even smaller quantities, butane, pentane, hexane, heptane, and octane appear. As impurities are common carbon dioxide, nitrogen, helium, and hydrogen sulfide. [2]

A gas consisting of less heavy hydrocarbons (methane for the most part) is so-called dry gas and is distributed to domestic and industrial users for use as fuel. The remaining gas fraction, composed of higher molecular weight hydrocarbons, is the so-called wet gas.[2]

For the designing production facilities, is an essential need to know two things.

- Accurate knowledge of hydrocarbon fluid phase behavior.
- How the gas fluid will behave under a wide range of pressure and temperature conditions.

## 1.2 Properties of gases

Given the fact that fluid for the system design is a gas, in both cases the refrigerant and the natural gas, it should be discussed certain gas properties such as gas density, viscosity, the factor of compressibility.

- **General Law of Gases**

An ideal gas is a hypothetical gas whose pressure, volume, and temperature behavior can be fully described with the ideal gas equation (1.1):

$$PV = nRT \tag{1.1}$$

Although there is no ideal gas in nature, discrepancies in the behavior of real gases in reasonable temperature and pressure margins do not substantially alter the calculations. Therefore, we can safely use the ideal gas equation to solve many gas problems. [3]

Even the circumstances mention before there is a thermodynamic property useful for modifying the law of gases ideal to adjust to the behavior of a real gas

The compressibility factor ( $Z$ ), also known as the compression factor, is the reason for the molar volume of a gas relative to the molar volume of an ideal gas at the same temperature and pressure [3]. So due to this case, the equation of real gas can be described as:

$$PV = ZnRT \tag{1.2}$$

- **Gas density**

Gas density changes with pressure and temperature, so the use of standardized volumes when referring to quantities of gas is mandatory. The gas density is defined as mass per unit volume and so can also be derived and calculated from the real gas law [3]. A normal cubic meter of gas corresponds to 1 [m<sup>-3</sup>] at 15 [°C] at 101.325 [kPa]. Equation (1.3) is described below as:

$$\rho_g = \frac{\dot{m}}{V} = \frac{PM}{ZRT} \quad (1.3)$$

For this analysis, we need to work with changes in pressure, volume, and temperature. When conditions change, we must use a modified form of the ideal gas equation that takes into account the initial and final conditions. It means we are going to change from standard conditions to actual process conditions using this equation (1.4):

$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2} \quad (1.4)$$

As the process involve heat transfer between to medium; the heating value of natural gas and refrigeration is an important factor for the thermodynamic properties of ideal gases and the calculation of power. Although this heating value in natural gas could be variable and depends on its accumulations which are influenced by the amount and types of gases they contain.

So the next question should be: Under which conditions are desired to be the output of the process? As was said before, natural gas has to be delivered to the distributor line and the assurance of quality is important.

It means that the gas has to undergo a series of energy processes in which going to be based on our theoretical study: Natural gas dehydration.

### **1.3 Natural gas dehydration**

The most common reason for dehydration in natural gas is to prevent the formation of carbohydrates in pipelines. Natural gas hydrates are crystalline solid compounds formed by the combination of natural gas and water under pressure at temperatures considerably above the freezing point of the water. As I mentioned before, the properties of natural gas have to undergo a series of changes to reach the desired properties and in the presence of free water, carbohydrates will form when the temperature is below the point called hydrate temperature, for this reason, the need to prevent the formation of carbohydrates is obvious. Consequently, the easiest way to remove carbohydrates is to remove substantially natural gas flow water. [4]

#### **1.3.1 Problems with water in the gas**

But what is the main problem of water in the gas? Basically what we have to know is described in the book Natural Gas: Extraction to End Use in chapter 1.2 Problems with water in the gas.

*“If the temperature of pipeline walls or storage tanks decreases below the Tdew of the water vapors present in the gas, the water starts to condense on those cold surfaces, and the following problems can appear.*

- *NG in combination with liquid water can form methane hydrate. Methane hydrate is a solid in which a large amount of methane is trapped within the crystal structure of water, forming a solid similar to ice. The methane hydrate production from a unit amount of water is higher than the ice formation. The methane hydrates formed by cooling may plug the valves, the fittings or even pipelines.*
- *NG dissolved in condensed water is corrosive, especially when it contains CO<sub>2</sub> or H<sub>2</sub>S.*
- *Condensed water in the pipeline causes slug flow and erosion.*
- *Water vapor increases the volume and decreases the heating value of the gas.*
- *NG with the presence of water vapor cannot be operated on cryogenic plants". [4]*

### 1.3.2 Dew point control

The dew point of the hydrocarbon is the temperature at which the hydrocarbon begins to become counted when the natural gas is cooled at constant pressure and consequently ceases to be completely gaseous.

It is, however, an important quality parameter, stipulated in contractual specifications through the supply network, from producers to end consumers.

When gas is transported in pipelines, it should be considered also the control of the formation of hydrocarbon liquids in the piping system. Liquid condensation is a problem in measurement, pressure drop, and safe operation. It can also be a major problem when it becomes a design as a two-phase flow and the accumulation of liquid. To avoid the formation of liquids in the system, it is necessary to control the dew point of the hydrocarbon below the pipeline operating conditions. In the following figure (1.1) is shown an example of a hydrocarbon dew point curve for a typical natural gas mixture. The red arrow shows also the Cricondentherm, which is the maximum temperature above which liquid can't be formed regardless of the pressure.

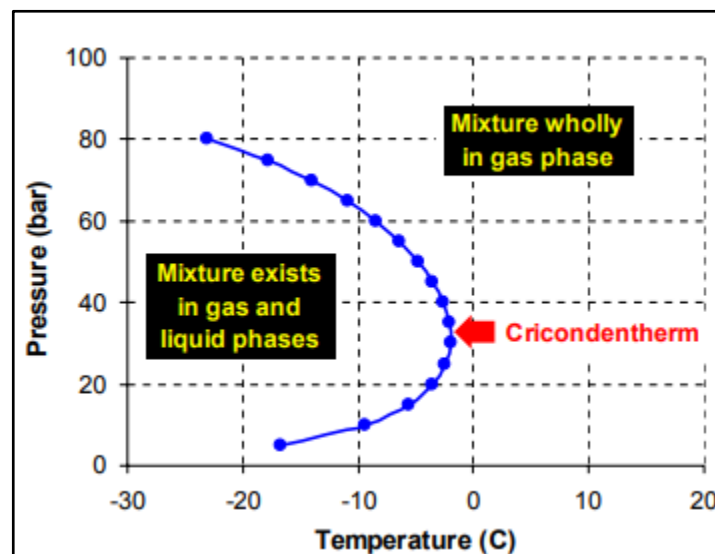


Figure 1.1 – Typical retrograde phase envelope and dew point curve for natural gas [5]

These dew point curves show that as the pressure decreases, liquid forms. The heavier the hydrocarbon, the more the dew point temperature increases as the pressure decreases.

### 1.3.3 Low-temperature separation

Several methods can be used to reduce the dew point of hydrocarbons.

If sufficient pressure is available, disposal can be achieved by expansion cooling on an LTS unit (low separation temperature).

The Joule-Thomson effect is often used to reduce the temperature of the gas during expansion. This reduction in temperature results not only in the condensation of hydrocarbon liquids but also in water condensation.

Water is usually removed as carbohydrates in this process, melted and removed. Therefore, the process can actually achieve control of the dew point of both water and hydrocarbon in a single unit. Nevertheless, as I mentioned before, the role of the Joule-Thomson effect could be not the best option, if we are talking about low pressure, because it is not feasible to apply it for these conditions.

### 1.3.4 Low-temperature separation with refrigeration

An alternative to the expansion cooling system is to use a mechanical cooling system to remove heavy hydrocarbon components, reduce the spray point, and separate water vapor as it is shown in the figure (1.2).

The method of dehydration that is intended to be used in this system is a refrigeration process that mixes ethanol with the gas and cools the gas to very low temperatures, it means is pumped heat from one medium to another. In this way, glycol helps ensure that hydrate formation does not block upstream process equipment from the LTS separator.

The refrigeration process is used at many different temperature levels to condense or cool gases, vapors, or liquids. Refrigeration is necessary when the process requires cooling to a temperature not reliably available from the usual water service or another coolant source, including Joule-Thomson, or polytropic expansion of natural gas or process system vapors [6].

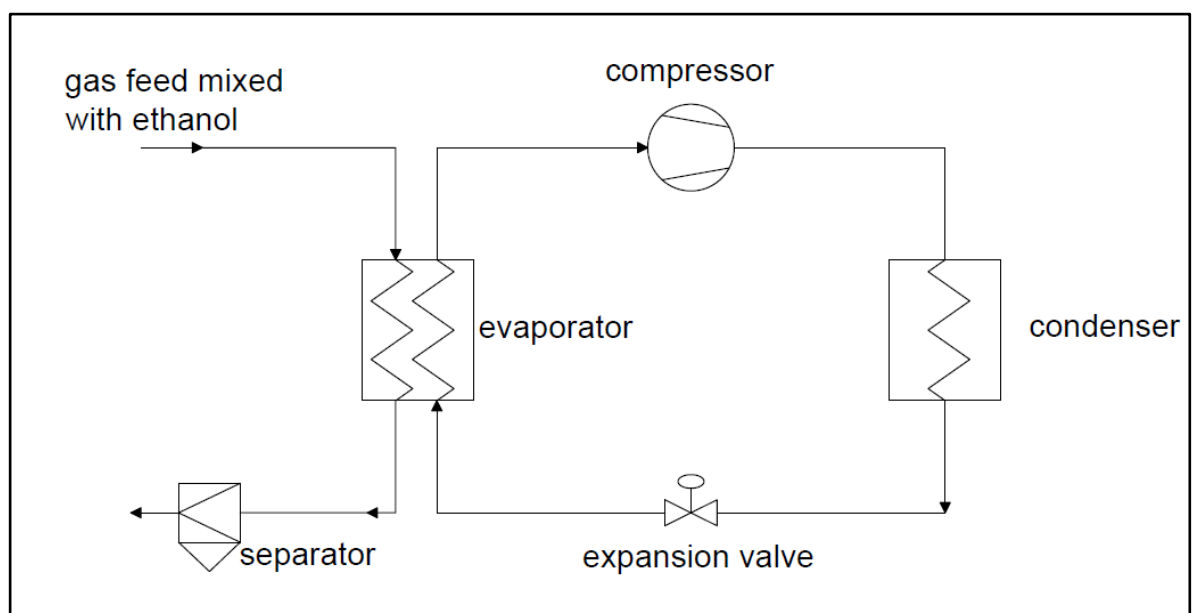


Figure 1.2 – Common scheme of low-temperature separation with refrigeration [author]



## 2 Principles of refrigeration cycles

In the previous chapter, my purpose was to relate the interaction between the refrigeration system and the low-temperature separation system. For low-temperature processes is necessary a heat rejection to the refrigeration system. As a result, further understanding of refrigeration theory is requisite.

Cooling involves transferring the energy from the body that we intend to cool to another, taking advantage of its thermodynamic properties. It generally has large-scale industrial use, for example in ice manufacturing and gas dehydration, as is the case in which we are seeing in this thesis. Applications in the oil and gas industry also include the purification of lubricating oils, reactions at low temperatures and the separation of volatile hydrocarbons.

Cooling systems have a range of classifications, ranging from use to the different means by which heat transfer usually occurs.

This study focuses on compression cooling systems. So I will expand the topic further below.

### 2.1 Compression cooling system

Mechanical or steam compression cooling is the most common mode of refrigeration. This method divided the circuit into two zones of high and low pressure, in which the refrigerant passes by, thanks to the work of compression applied by the compressor. In the ideal case, all the flow is frictionless, except the one passing through the expansion valve, and all processes except the condenser and evaporator are adiabatic. However in the real cycle, there are a large number of factors that produce irreversibility in the cooling cycle, so the cycle theoretical undergoes some deviations. The production of cold by mechanical systems is based on physical laws that regulate the evaporation and condensation of a fluid. That means the refrigerant has to return to its initial state and be ready to repeat the cycle through the main components of the system are evaporator, condenser, compressor and expansion valve [7]. The system is shown in detail in figure (2.1).

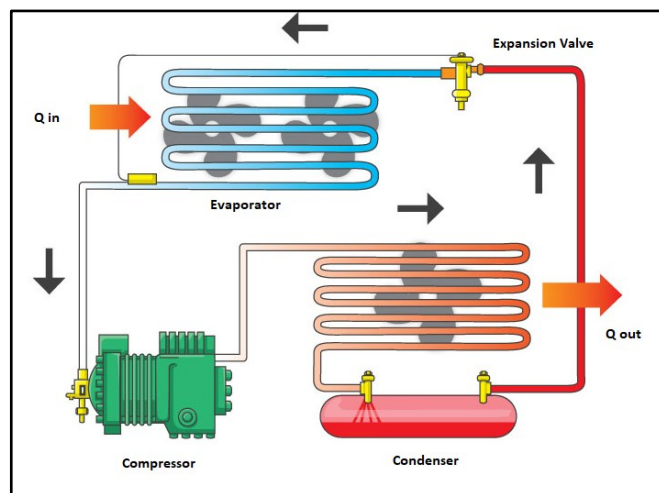


Figure 2.1 – Scheme of the refrigeration cycle and its main components [8]

## 2.2 Ideal cooling cycle

An ideal steam compression cycle presented in the figure (2.2) refers to an inverted cycle of Carnot, and it is characterized due to that the refrigerant evaporates completely before being compressed and replaces the turbine with a strangulation device. [7]

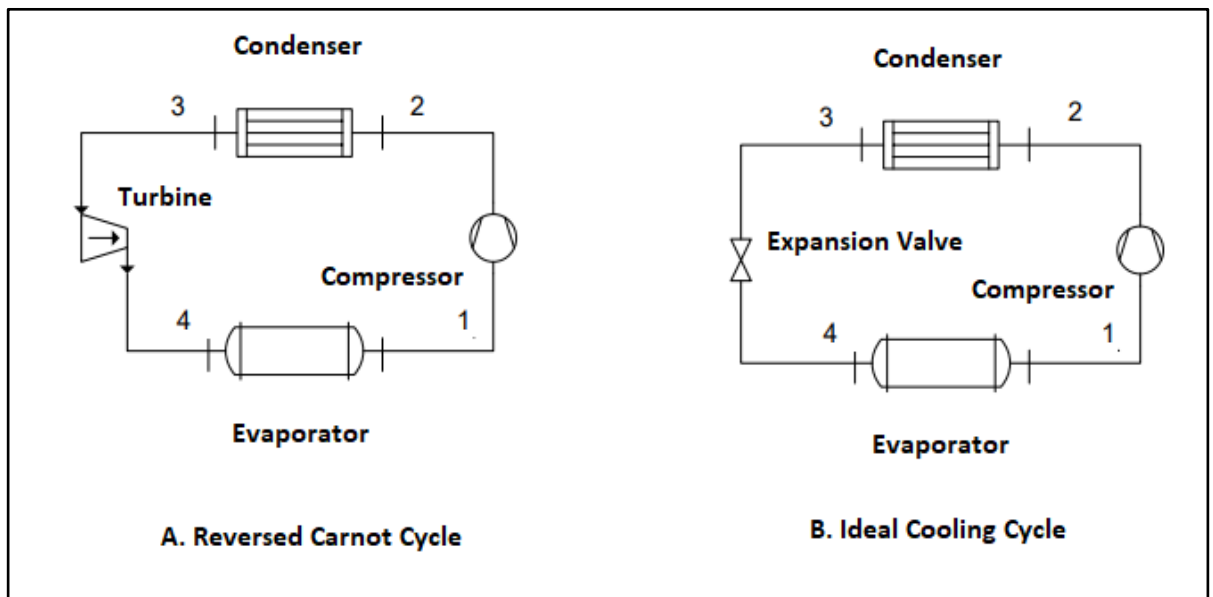


Figure 2.2 – Comparison of reversed Carnot and ideal cooling cycle [author]

In an ideal cycle of compression cooling, the processes are the following:

1-2 The refrigerant enters the compressor in state 1 as saturated steam and is compressed isentropically up to the condenser pressure. The temperature of the refrigerant increases during the isentropic compression process, up to a fairly higher than the temperature of the surrounding environment.

2-3 Then the refrigerant enters the condenser as superheated steam in state 2 and comes out as saturated liquid in state 3, as a result of heat rejection around. The refrigerant temperature in this state will be maintained above the surrounding temperature.

3-4 Saturated liquid coolant in state 3 is strangled to pressure evaporator when passed through an expansion valve or capillary tube.

4-1 Represent temperature drops below the temperature of the refrigerant cooled space during this process. Coolant enters evaporator in state 4 as low-quality wet steam and evaporates completely absorbing heat from the refrigerated space. Coolant comes out of the evaporator saturated steam and re-enters the compressor, completing the cycle.[7]

The process is shown in the following T-s Diagram, figure (2.3).

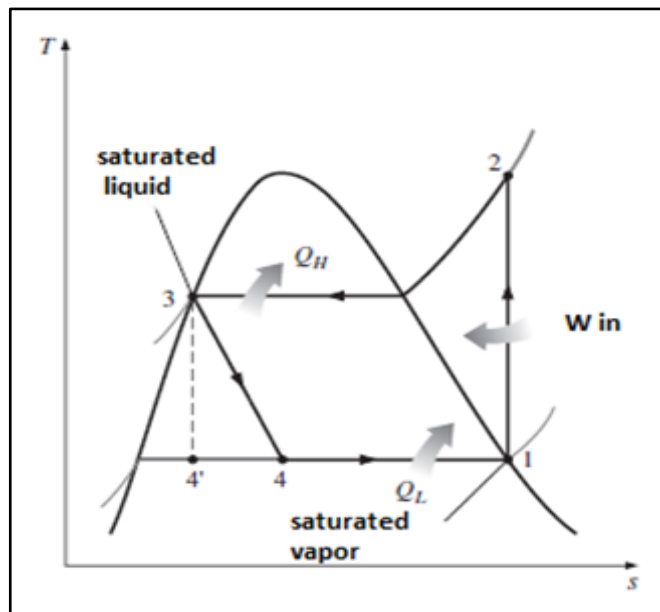


Figure 2.3 – T-s diagram of an ideal cooling system [7]

## 2.3 Equipment and main components of a compression cooling system

The basic components of any steam compression cooling system are evaporator, compressor, expansion valve, and condenser. However, the complete system has many other additional components (e.g., piping, controls, and regulations) that will not be explored here. Despite this, the general scheme representing all steam compression cooling systems is composed of its 4 main components, which, in the same way, make up the system analyzed in this thesis.

### 2.3.1 Compressor

The main function of the compressor is to circulate the coolant through the system and increases the pressure of the refrigerant vapor to create the pressure differential between the condenser and evaporator.

Depending on manufacture, technology or operation there are several types of compressors used for air conditioning and cooling units. Some of the main types of compressors are the following: positive displacement and dynamic.

Positive displacement compressors increase the pressure of the refrigerant vapor by reducing the volume of the compression chamber through the work applied to the compressor mechanism. Positive displacement compressors include many types of compressors currently used, such as reciprocating, rotary (rolling piston, rotary paddles, single screw, double screw), and orbital. [9]

Dynamic compressors increase the pressure of the refrigerant vapor, by continuous transfer of kinetic energy, from the rotating element to the steam, followed by the conversion of this energy in the form of increased pressure. [9]

To avoid leakage of refrigerant to the environment, the compressor must be completely sealed and also for the purpose to avoid contamination of the refrigerant by air or moisture.

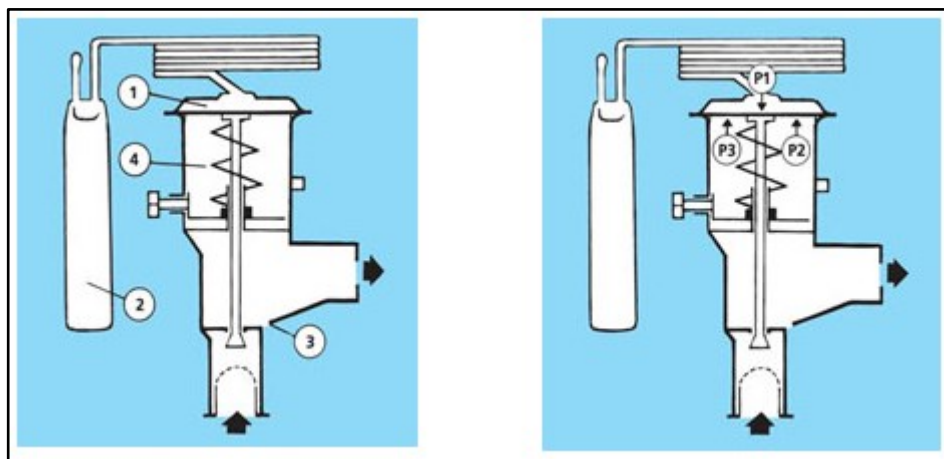
### 2.3.2 Condenser

The condenser in a cooling system is a heat exchanger that rejects all heat from the system. The refrigerant vapor enters the condenser after being compressed by the compressor at a high pressure and high temperature, allowing the exchange of heat with the working fluid, in this case, natural gas. This heat is composed of the heat absorbed by the evaporator plus the heat produced by the degradation of some of the mechanical energy delivered to the compressor. The cold coolant is then condensed back into the liquid state and drained from the condenser to continue in the cooling cycle.[9]

Some of the most common types of condensers, according to their operation and their materials, are for example the air-cooled, water-cooled, concentric tube, shell, and tubes. In this case, the design suggests that the method of heat transfer is going to be by natural convection with air.

### 2.3.3 Expansion valve

The valves are the expansive unit that generates a change in fluid pressure, from high temperatures and pressures in the condenser to low at the inlet of the evaporator.[9]



**Figure 2.4 – Parts and functioning of a thermal expansion valve [10]**

Basically, the operation of the expansion valve, as is shown on figure (2.4), consists of a thermostatic element (1) separated from the body using a membrane, the thermostatic element is in contact with the bulb (2) through a capillary tube, a body with a seat and hole (3) and a spring or spring [10]

Functioning:

P-1: the pressure of the bulb that acts at the top of the membrane and in the direction of the opening of the valve.

P-2: the pressure of the evaporator, which influences the lower part of the membrane and the closing direction of the valve.

P-3: the force of the spring, which influences the lower part of the membrane and the only variable that is controllable by the technician.[10]

For the selection of the expansion valve, it is also important to take in count its: Pressure drop through the valve, internal or external pressure equilibrium, refrigerant, evaporator capacity, evaporation pressure, condensation pressure.

### 2.3.4 Evaporator

The evaporator is the component of the refrigeration system where the heat dissipated from a hot medium is removed. Just like condensers, it is also known as a heat exchanger, normally consisting of a series of tubes arranged in a metal container (housing). To this device comes the liquid refrigerant, partially vaporized and then boiling, at low pressure. In this study, the evaporator is designed to be in direct contact with the heat-dissipating unit. [11]

To generate the movement of the refrigerant inside the evaporator, all the steam that leaves the equipment is sucked by the compressor, increasing the pressure necessary to start the condensation process.

Some of the main requirements for the optimal operation of an evaporator are: maintain a constant exchange volume, allow coolant flow with minimal pressure drop, have an appropriate design (with suitable materials) that allows heat flow to the refrigerant in an easy and fast way.[9]

## 2.4 Refrigerant

The refrigerant acts as a means of transport to move heat from the evaporator to the condenser, where it is discharged into the atmosphere or cooling water, in the case of water-cooled systems. A change in a liquid state to steam, and vice versa, allows the refrigerant to absorb and discharge large amounts of heat efficiently.[12]

Despite exists a whole range of refrigerant used in the cooling industry, for the study of this thesis, refrigerant R449A and R404A are considered parameters of the design.

R449A and R404A are both mixtures of common hydrofluorocarbons (HFC), which in fact are organic compounds composed of Hydrogen, Fluorine, and Carbon. In general, HFCs are relatively non-flammable, chemically stable, and nonreactive Unlike R404A, R449 contains a new hydrofluoro-olefin molecule R1234yf (HFO). For a better understanding, table (2.1), show below, the percentage of composition of each refrigerant.

**Table 2.1 – Main composition of Refrigerant R404A and R449A [13]**

Composition	R404A	R449A
R143a	52 %	0 %
R32	0 %	24 %
R125	44 %	25 %
R134a	4 %	26 %
R1234yf	0 %	25 %

## 2.4.1 Refrigerant properties

These refrigerants share common properties that characterized it as suitable for cooling effect. Some of the common physical properties are described below, and others are shown in Table (2.2).

- It should have a low freezing point than normal operating conditions.
- The critical pressure of the refrigerant should be higher than the condenser pressures.
- High heat transfer coefficient.
- High Evaporator Pressure.
- Low condensing pressure.[14]

R404A is a quasi-azeotrope refrigerant blend, which means it condenses and boils at nearly the same temperature for a given pressure. Saturated pressure curves play an important role in the analysis of the behavior of two phases. As we are talking about refrigerant, this knowledge is going to be important. This relation is shown in figure (2.5).

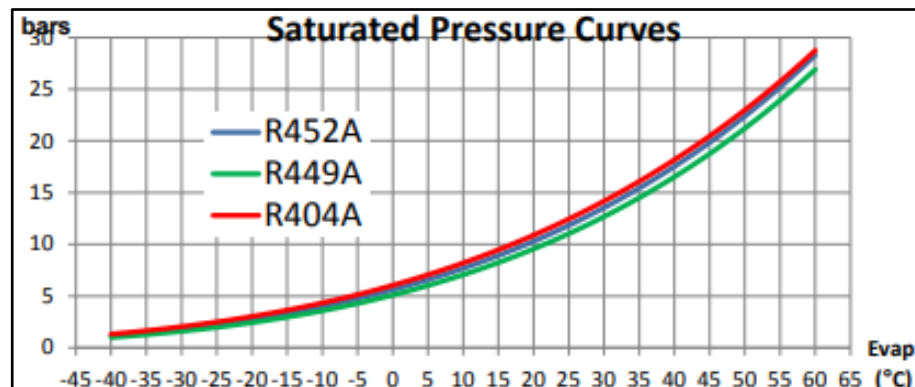


Figure 2.5 – Saturated pressure curves of R452A, R449A, and R404A [13]

Each type of refrigerant has particular advantages and disadvantages and the choice of the combination between system and refrigerant that is imposed will depend on environmental, economic, and energy parameters, and the approved regulation in each country.

Table 2.2 – Main properties of Refrigerant R404A and R449A [13]

Refrigerant Properties	R404A	R449A
Critical temperature [°C]	72.1	80.1
Boiling point at 1 bar [°C]	-46.5	-46
Critical pressure [bar]	37.3	44.1
Liquid density at 32 [°C]	1,010	1,061
Vapor density at -30 [°C]	10.7	7.3

## 2.4.2 Environmental concerns and regulations

In recent years, climate change and ozone depletion have arisen an intensified interest in protecting the environment from harmful effects associated with the emission of gases into the atmosphere. First, removal of refrigerants, HFCs and HCFCs were established in the Montreal Protocol, because they were considered contributors to the destruction of the ozone layer.

Subsequently, in the Kyoto Protocol, the basis and steps for reducing atmospheric warming were negotiated in which HFCs (among others) were identified as fluids that cause this phenomenon. Thus in Europe, Regulation (EU) No. 517/2014 set limits in various areas of cooling for HFCs. Limits are quantified using the GWP (Global Warming Potential) parameter, which quantifies the effect of refrigerants when emitted directly into the atmosphere, with carbon dioxide as the basis for comparison.[15]

As Danfoss's article explains on its website; the global warming potential (GWP) of R404A of 3922, is already affected by EU refrigerant disposal rules and is well above the 2500 level imposed by the F-Gas regulation for stationary cooling equipment for temperatures above -50 [°C]. Therefore supply has been reduced by quotas, meaning prices have fluctuated a lot in recent months. As a response to this issue starting, January 1, 2020, maintenance with recycled R404A will only be permitted, and can no longer be used in any new refrigeration and cooling equipment. [16]

On the other hand, R449A is considered a stable replacement or alternative for R404A with a 50-65 % reduction in the GWP. The following figure (2.6) is a representative comparison.

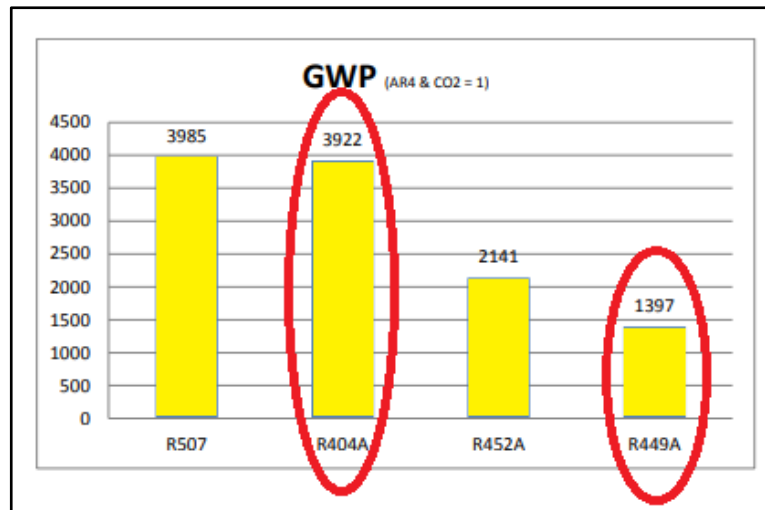


Figure 2.6 – Global warming potential comparison of some refrigerant [13]

R449A is also non-flammable and it has been assigned an A1 safety classification, which means that the substance is classified as “nondangerous”, under ASHRAE 34 and EN 378 standards. Another fact is that chlorine is not present in R449A, having zero Ozone Depletion Potential.

However, R449A contains mildly flammable components R32 and R1234yf, and it should not be mixed with air to check for system leaks as these mixtures can become combustible.

### 2.4.3 Selection of refrigerant

Notwithstanding, R404A and R449A are going to be the object of analysis for this cooling system. It is important to know that in each of the different cold production methods, there are one or more appropriate refrigerants for certain operating conditions, which guarantee optimum efficiency and safety, concerning their chemical and physical properties. But some certain

minimum conditions and properties must be satisfied, that it certainly aids select the best refrigerant. These conditions are listed below.

- The refrigerant must not be combined or reacted with the materials used for the construction of the cooling system,
- Chemical stability within the domain of operating temperatures and pressures.
- Absence of toxicity.
- It must not be explosive or flammable.
- Easy leak detection. For safety, operation, and economic aspects, it is necessary that the circulation of refrigerant is carried out in hermetic ducts and that leaks should occur.
- should be immediately detected, preferring those refrigerants that have a smell penetrating
- No effect on the lubricant. Some of the cooling system equipment used lubrication, and it should not cause any effect on the lubricant.
- In the case of vaporization cooling, the evaporation pressure of the refrigerant should be within the possible, something higher than atmospheric pressure. In this way prevents air from entering the system.
- The generation of high condensation pressures requires structures that support this pressure, increasing the cost. It is suggested to work the refrigerant under operation not close to the critical point, in order to more easily perform the condensation.
- The higher its cooling capacity or power, the lower a cooling power will be the required amount of refrigerant in circulation for a given cooling power.
- Cost and availability.[17]

## 2.5 Coefficient of performance

As previously mentioned, the goal of a mechanical cooling system is to transfer heat from one low-pressure environment to another, at a higher temperature. The cooling effect or amount of heat removed from the environment at low temperature is much greater than the calorific equivalent of the work required to produce this effect. Thus the effectiveness of a cooling system is measured as well as for an engine such as the relationship between the useful cooling effect and the work required to produce that effect. This ratio is called the coefficient of performance and indicates the effectiveness of the system [18]. The equation is the following (1.5).

$$COP = \frac{\text{Cooling effect}}{\text{Work}} \quad (2.1)$$

Moreover, the enthalpy difference of the refrigerant between the inlet and the outlet locations of an evaporator is called the refrigeration effect or cooling\_effect. And the Work is represented by the heat equivalence of the energy supplied to the compressor.

In a first analysis, we can say that a larger COP, indicates that you get a greater cooling effect, with less mechanical work of the compressor, while a lower COP, indicates that you get a less cooling effect with more work of the compressor.



## 2.6 General analysis of compression cooling system

Previously in chapter 2.2, the ideal compression cooling cycle was described by the Temperature-entropy diagram, providing us a better comprehension of the standard process. Charts or diagrams have been used extensively in the literature to present the thermodynamic properties of refrigerants. However, another commonly used charts depict enthalpy and pressure values as shown in figure (2.7).

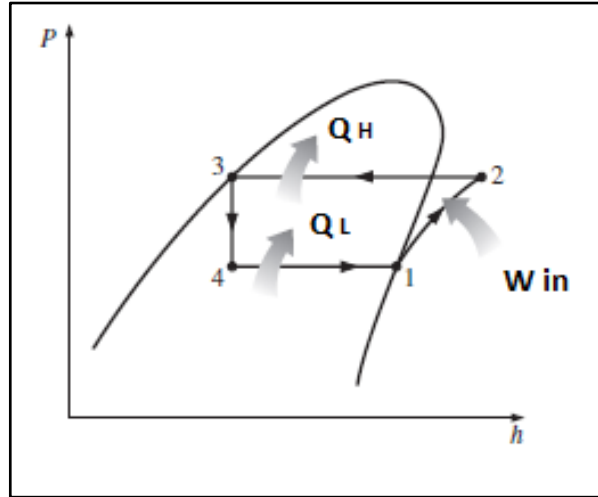


Figure 2.7 – P-h diagram of an ideal compression cooling cycle [7]

This P-h diagram follows also the same number of the process as it is described on the temperature- enthalpy diagram. Though pressure–enthalpy charts are useful in obtaining values of H 1, H 2, and H 3, it also brings us ideas about energy balance. During the compression step (from 1-2), work is done by the compressor, resulting in an increase in both, enthalpy and pressure. The expansion valve is a constant-enthalpy process (from 3-4) that allows the liquid refrigerant under high pressure to pass at a controlled rate into the low-pressure section of the refrigeration system.[18]

The four components associated with the steam compression cooling cycle are stationary flow devices, so the four processes that make up the cycle can be analyzed as stationary flow processes.

$$\left[ \dot{Q}_{in} + \dot{W}_{in} + \sum_{in} \dot{m}(h + e_c + e_p) \right] - \left[ \dot{Q}_{out} + \dot{W}_{out} + \sum_{out} \dot{m}(h + e_c + e_p) \right] = \frac{d}{dt} \left[ m_{vc} \sum (u + e_c + e_p)_{vc} \right] \quad [kW] \quad (2.2)$$

Changes in kinetic energy and refrigerant potential can be ignored due to that they are often small relative to the terms of work and heat transfer [7]. Then the equation 1.6 could be rewritten as equation (1.7).

$$(q_{in} - q_{out}) + (w_{in} - w_{out}) = h_{out} - h_{in} \quad (2.3)$$

And the equation (1.5) about Coefficient of performance could be relevant to equation (1.8), according to the P-h diagram shown before in figure 2.7:

$$COP = \frac{h_1 - h_4}{h_2 - h_1} \quad [-] \quad (2.4)$$

The change in enthalpy per unit of mass is described based on the specific temperature and heat called  $C_{p,prom}$ , this being an average specific heat at constant pressure for the working temperature range. Besides,  $T_1$  represents the fluid temperature and  $T_4$  the outlet temperature of the evaporator, as in equation (1.9) shows:

$$\Delta h = C_{p,prom} (T_1 - T_4) [\text{kJ} \cdot \text{kg}^{-1}] \quad (2.5)$$

That means the rate of heat accepted by the refrigerant as it undergoes evaporation process in the evaporator is given by:

$$\begin{aligned} Q_{\text{evap}} &= \Delta h_{41} [\text{kJ} \cdot \text{kg}^{-1}] \\ Q_{\text{evap}} &= C_{p,prom} (T_1 - T_4) [\text{kJ} \cdot \text{kg}^{-1}] \end{aligned} \quad (2.6)$$

And the heat rejected to the environment can be expressed as:

$$\begin{aligned} Q_{\text{cond}} &= \Delta h_{23} [\text{kJ} \cdot \text{kg}^{-1}] \\ Q_{\text{cond}} &= C_{p,prom} (T_2 - T_3) [\text{kJ} \cdot \text{kg}^{-1}] \end{aligned} \quad (2.7)$$

A real steam compression cooling cycle differs from one ideal in several respects, mainly due to the irreversibility that occurs in various components.

Also, we have to consider that, in the ideal cycle, the refrigerant leaves the evaporator and enters the compressor as saturated steam. The compression process in the ideal cycle is internally reversible and adiabatic and therefore isentropic. And last, ideally, the coolant is assumed to leave the condenser as a saturated liquid at the compressor output pressure.

### 3 Principle of heat exchangers

Two of the major components of the refrigeration system: Evaporator and Condenser, included in the design are related or classified as heat exchangers equipment. As long as we know we are going to deal with heat exchangers, we should also know some basic parameters and features about it. Heat exchangers are used in a wide range of applications, and therefore they can be classified by different configurations and types.

Although, as I stated previously, this whole design is a composition of a two-part thesis; in the first part, the main focus is about heat exchangers, hence further information is going to be available there. For the study that corresponds to this part of the thesis, it is only necessary to understand parameters that will be relevant to the design of the compression cooling system.

The simplest type of heat exchanger consists of two concentric tubes of different diameters, as it is shown in figure (3.1) below. As it is stated in the parameters of the design, the condenser as Air-cooled type. Air-cooled condensers can be either a tube-and-fin type or plate type. It can present some advantages and disadvantages between them, like for example fins on tubes allow a large heat transfer area in a compact case. Regardless, the plate condensers have no fins, so they require considerably larger surface areas. The plate condensers are also cheaper to construct and require little maintenance [18].

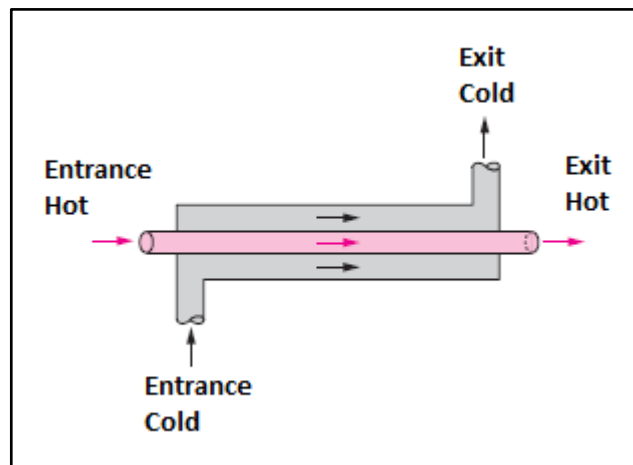


Figure 3.1 – Parallel flux of two concentric tube heat exchanger simples type [19]

In contrast, evaporators are not that easy to define a classification for this type of application. However, the best choice is relay on previous experience of the company and in refrigeration literature. For the natural gas industry, if we assume the gas as two phases, due to the content of water; the typical evaporators that we are going to find are those with internal circulation, in which the fluid is going to flow through, separately form the refrigerant by walls or tubes.

Commonly, evaporators and condensers are types of heat exchangers that one of the fluids undergoes a phase change process; in this case the refrigerant. The reason for the heat transfer is shown in figure (3.2) for a condenser and figure (3.3) for an evaporator, respectively.

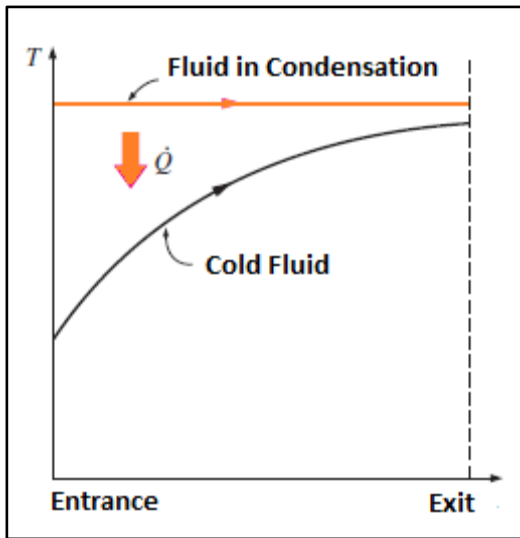


Figure 3.2 – Variation of temperature in a condenser [19]

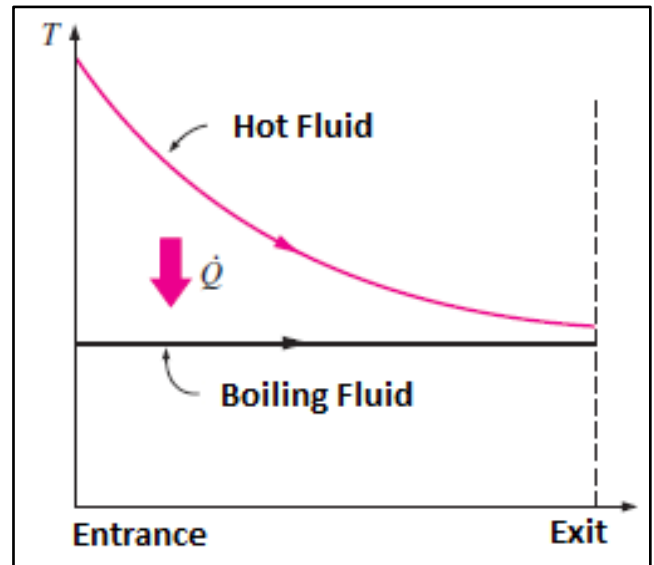


Figure 3.3 – Variation of temperature in an evaporator [19]

### 3.1 Characteristics for Evaporator and Condenser Design

- Temperature difference between the fluids.

The difference in temperatures between the refrigerant fluid, evaporating inside the heat exchanger, and that of the fluid to be cooled is called thermal jump, as shown in figure(3.4)[20]. In heat exchangers, it is necessary to consider design for small temperature differences with a stable boiling process. One of the main reasons is for reducing the power requirement in the compressors.

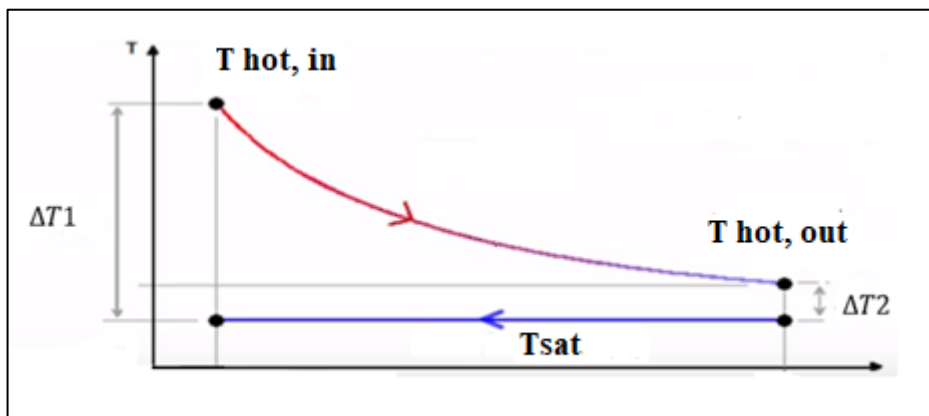


Figure 3.4 - Consideration of temperature difference between the refrigerant and the natural gas [author]

If we analyze a real temperature profile of the refrigerant in the evaporator, we are going to see that in reality, the evaporation temperature is never constant through the evaporator. The reason is that inside an evaporator, the increased velocity of the liquid/gas refrigerant mixture will induce a pressure drop, which thus reduces the saturation temperature [21]. According to

the Refrigeration Manual of Juan Franco Lijo, for practical design and indicative purposes, we will consider the following conditions for thermal jumps:

a. If the fluid to be cooled is liquid, the temperature difference is in the order of 5 °C.

b. If the fluid to be cooled is gas, the temperature difference is in the order of 5 to 8 °C for forced evaporators and 8 to 12 °C for natural circulation evaporators. [20]

- **Influence of Superheating on the Evaporation Process**

Superheating is the process in the evaporation in which no liquid refrigerant is present, therefore the temperature of the vapor increase and becomes superheated.

Normally superheating only accounts for approximately 5% of the total heat of absorption, and the driving force for heat transfer is gradually reduced because the temperature of the vapor increases. That means that a heat exchanger designed with a very high level of superheating will require a proportionately very large heat transfer area for vapor heating. On the other way, there could be also a decrease in evaporation temperature if levels of superheating are not well considered related to the heat transfer area, which could lead to reduced system efficiency and capacity.[22]

- **Type of refrigerant. Azeotropic and Zeotropic Refrigerants**

As I mentioned previously, the design of the refrigeration system is going to be based according to the properties of the refrigerant R404A and R449A. It is important to dedicate a section, before the analysis of the design, to a parameter called Glide, which is a special characteristic of the zeotropic refrigerant mixture. Glide, or temperature glide, is the difference between the bubble point and the dew point and is significant for evaporators and condenser design.

Azeotropic refrigerant is a chemical mixture that never has the same vapor phase and liquid phase composition at the vapor-liquid equilibrium state [23]. In other words, a zeotropic mixture will boil between across a range of temperatures at that same single pressure, and because of that, the phenomenon of glide may occur. As a mixture, during the boiling process in the evaporator, the vapor formed first will be rich in the refrigerant with the lowest boiling temperature. Consequently, the saturation temperature will increase and reach a maximum, just before the superheating.

Meanwhile, an azeotropic refrigerant is a mixture of components that boil at the same temperature. This indicates that the components in this type of refrigerant will evaporate and condense together as one because there is no such factor called glide. Therefore the pressure drop will result in the inlet saturation temperature being higher than the outlet evaporation temperature. [24]

In our case, let's focus our analysis on R404A and R449A. What types of refrigerant are them? For a better understanding, we can take a look at the table (3.1) below, which describes its behavior at a certain dew point.

**Table 3.1 - Effective Temperature Glide for refrigerant R404A and R449A [13]**

	Effective Temperature Glide (K)			
	Evaporator			Condenser
Fluid - $T_{dew}$	-35 °C	-10 °C	0 °C	45 °C
R404A	0.3	0.3	0.4	0.3
R449A	3.0	3.5	3.8	4.5

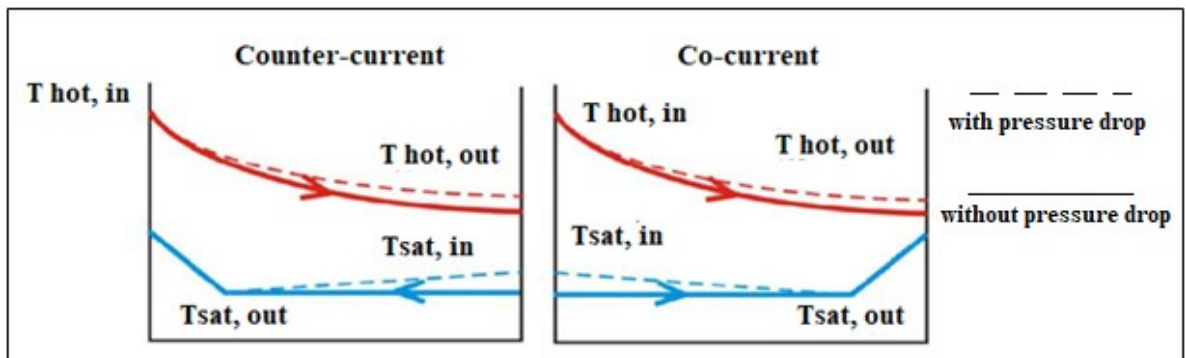
For R404A, we can see that it shows a temperature glide lower than 0.5K. Consequently, it condenses and boils at nearly the same temperature for a given pressure. R404A is characterized by being a quasi-azeotrope refrigerant blend.

On the other hand, R449 is a zeotropic refrigerant blend containing R32, R125, R1234yf, and R134a, and it is shown on the table, that depending on the operating conditions R449A exhibit 1.5 to 5K of temperature glide.[13]

These considerations are important when we talk about the design of the evaporator and condenser, which results in more influence in the pressure drop that undergoes the refrigerant during the boiling or condensation process. Normally, the temperature at which boiling starts is called the bubble point. Meanwhile, the point where all liquid has evaporated is called the dew point. The dew point temperature corresponds to the outlet evaporation temperature of a pure refrigerant.

- **Heat exchanger configuration: counter-current or current arrangement**

Heat exchanger configuration is often another parameter that is a reason for discussion. To make the best proper selection it is important to analyze, the temperature difference between the evaporation media and the secondary fluid, as it is shown in figure (3.5) below.



**Figure 3.5 - Evaporator with superheating [author]**

We can see that both temperature profiles include not only the temperature of saturation but also the part of superheating. The graph at the right shows a co-current configuration. In the co-current configuration, the two temperature profiles tend to the same value, meaning that it has a negative influence on the temperature profile. Thus heat exchanger would have a part with low heat transfer efficiency characterized by low heat transfer coefficients and low-temperature differences, as we can see on the graph.

The behavior of the counter-current configuration differently shows how to compensate low values of heat transfer coefficient with high values of temperature difference. Another factor that is a plus for non-azeotropic refrigerants is that the temperature glide has a positive effect on the mean temperature difference for counter-current evaporators.

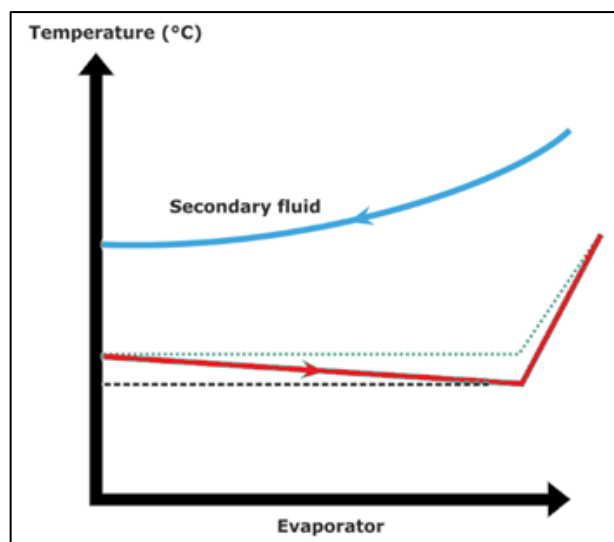
In conclusion, it is convenient to select the counter-flow configuration, consequently, the evaporator should operate close to ideal conditions.

- **Pressure drop effect on the heat exchanger**

Pressure drop is not only a phenomenon with both positive and negative consequences for the heat transfer process, but it is very important, as it affects the power and energy consumption in the compressor that drives the fluids. Also, we have to consider that pressure drop can be influenced by many causes, and the following criteria could be taken into account:

- Pressure drop due to friction,
- pressure drop due to speed changes,
- pressure drop due to changes in direction,
- pressure drop due to the type of refrigerant.

As it is the result of many causes, the pressure drop is an important step to close the refrigeration cycle. Although it is true, the expansion of the refrigerant before entering the evaporator is obtained by the pressure drop exerted by the expansion valve. Then the refrigerant flows through the channels of the evaporator, changes of speed occur and turbulence is created, thus result also a pressure drop. The pressure drop is proportional to the velocity squared, so a higher vapor content in the refrigerant will result in a larger pressure drop. However, this pressure drop results in positive due that turbulence is desirable in heat exchangers because it improves heat transfer. Below, it is shown a graph of the effects of pressure drop on the saturation temperature.



**Figure 3.6 -The effect of pressure drop on the saturation temperature [22]**

Figure (3.6) shows the behavior of an azeotropic refrigerant, as it is stated before, a refrigerant without the glide. Generally, the pressure drop increases with higher inlet vapor

quality, mass flow, and level of superheating. A lower evaporation temperature corresponding to a lower evaporation pressure also results in a higher pressure-drop due to the larger specific volume of the vapor formed. In a certain way for a zeotropic refrigerant result positive, because the saturation temperature increases gradually during the boiling process, whereas the induced pressure drop will act to reduce the evaporation temperature by decreasing the saturation temperature. [22]

### 3.2 General analysis of heat exchangers

Ideally assumptions in Heat Exchangers, are keys that are going to be helpful for the design further. These are some of the considerations:

- They can be considered as stationary flow devices. Heat exchangers usually operate without changes in their operating conditions for long periods.
- The outer surface of the heat exchanger is considered to be perfectly isolated. Heat transfer only occurs between the two fluids and there is no heat loss towards the surrounding environment.
- The specific heat of a fluid can be considered as a constant at some average value, despite changes with temperature. In a specific range of temperatures, it can be considered as a constant at some average value, with little loss in accuracy.[19]
- According to the first law of thermodynamic:

$$\dot{Q} = m \cdot c_p \cdot (T_{out} - T_{in}) [kW] \quad (3.1)$$

The equation for energy balance in evaporator and condenser is also described in chapter 2.6 in equation (1.10) and (1.11) respectively.

The task that is involved in the design of this system corresponds to the selection of a heat exchanger that achieves a specific temperature change of known mass flow in the fluid stream. Further, the logarithmic mean temperature difference is going to be applied for the design.



## 4 Design of a cooling system for low-temperature drying unit

Basically, the system design consists of the selection of equipment to combine and operate most economically for the specified conditions. Beforehand, I detailed in the background, we can divide the system into 4 main processes. Nevertheless, as it is shown in figure (4.1), the subject to study in this thesis is focused on step 3, Cooling System. Although the specific conditions should be applied to all the components of the system, including those on steps 1, 2, and 4.

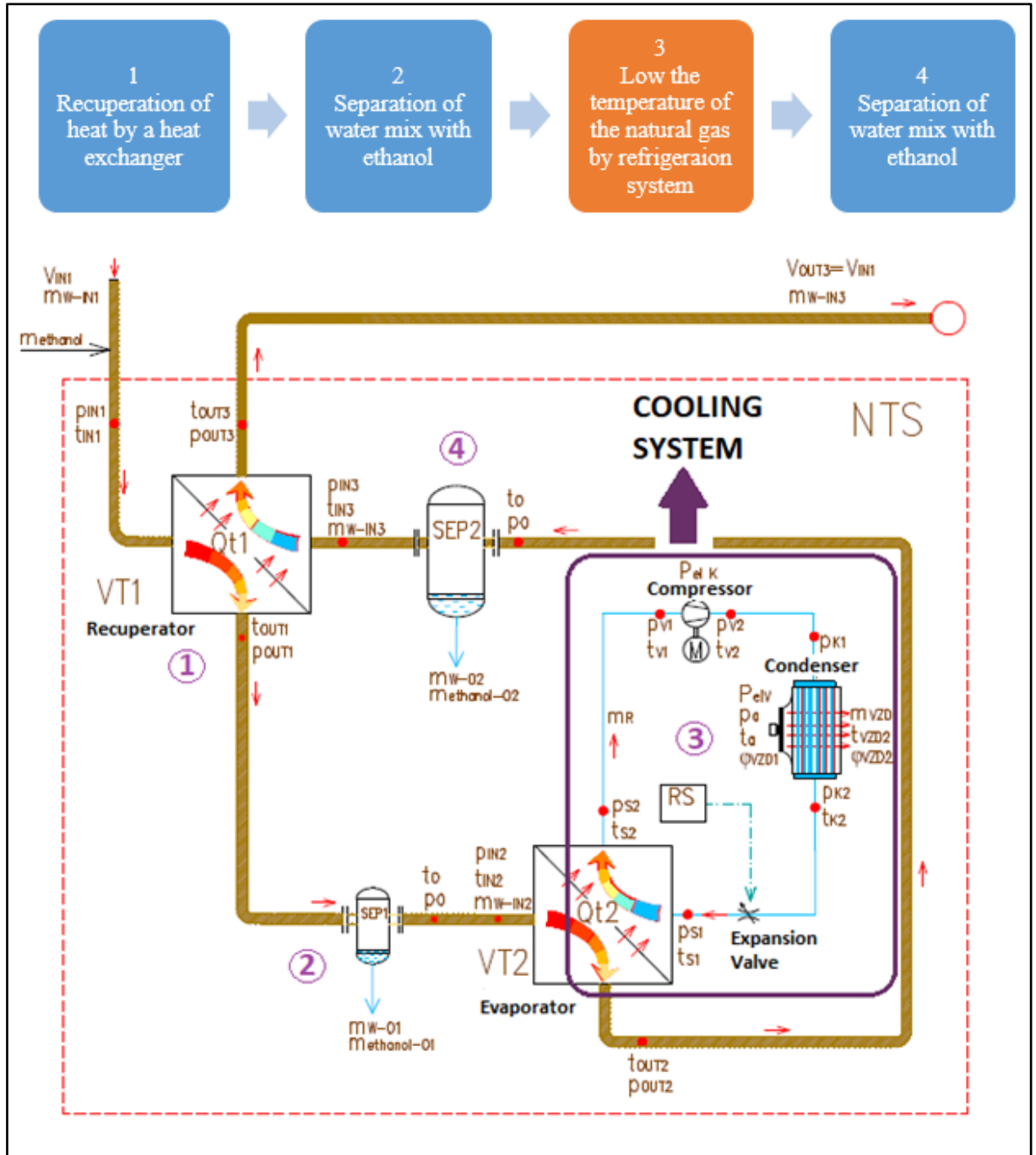


Figure 4.1 – Low the temperature of natural gas by the refrigeration system [author]

## 4.1 Parameters

The basis for the entry is shown below.

- Basic scheme of the natural gas treatment unit – see figure 4.1.
- Table of input and output physical quantities of the process for four variants of basic input-.
- The following basic parameters differ in each of the four variants:
  - ✓ **Option 1:** Flow rate of gas  $V_{in1} = 25000 \text{ [Nm}^3\cdot\text{day}^{-1}]$ , refrigerant R449A, others without changes.
  - ✓ **Option 2:** Flow rate of gas  $V_{in1} = 10000 \text{ [Nm}^3\cdot\text{day}^{-1}]$ , refrigerant R449A, others without changes.
  - ✓ **Option 3:** Flow rate of gas  $V_{in1} = 25000 \text{ [Nm}^3\cdot\text{day}^{-1}]$ , refrigerant R404A, others without changes.
  - ✓ **Option 4:** Flow rate of gas  $V_{in1} = 10000 \text{ [Nm}^3\cdot\text{day}^{-1}]$ , refrigerant R404A, others without changes.

### Main input and performance parameters of the gas treatment unit.

- Maximum flow  $25000 \text{ [Nm}^3\cdot\text{day}^{-1}]$ .
- Minimum flow  $10000 \text{ [Nm}^3\cdot\text{day}^{-1}]$ .
- The maximum value of the pressure loss between the inlet and outlet on the natural gas side at a flow rate of  $25000 \text{ [Nm}^3\cdot\text{day}^{-1}]$  is  $80 \text{ [kPa]}$ .
- The possibility of regulating the condensing temperature in the separator in the range of  $-13 \text{ [}^\circ\text{C]}$  to  $-19 \text{ [}^\circ\text{C]}$  (determined by the regulation of the cooling circulation performance).

### Required natural gas output properties:

At the exit from the LTS to the dispatch pipeline, the temperature of the natural gas  $t_{OUT3}$  shall be guaranteed at least  $+3 \text{ }^\circ\text{C}$ .

The water content (condensable water vapor) in the natural gas for dispatch must be at the maximum corresponding to the dew point of natural gas at  $-19 \text{ }^\circ\text{C}$  at a pressure of  $2,1 \text{ MPa-abs}$ .

## 4.2 Consideration for the mechanical and thermodynamic design

The design of these cooling systems is mainly defined based on the following parameters:

- Operating temperatures: (Evaporation and condensation temperatures),
- system capacity usually referred to in kW defined according to thermal load,
- environmentally friendly refrigerants with a wide cooling effect,
- system operating costs.

## 4.3 Calculation of the cooling load required

The calculation of the system cooling load was performed using as a reference table 6.8 – (Molar fraction and specific heat capacity of the natural gas components) from the thesis *Design*

of low-temperature drying system for treatment of raw natural gas by Jorge Sierra, which is referenced below as table (4.1).

**Table 4.1 – Specific heat of the Natural gas according to its components [author]**

Component	Molar fraction [%]	Specific heat capacity@260K [kJ. kg <sup>-1</sup> . K <sup>-1</sup> ]
Methane	96.388 % mol	2.339
Ethane	3.428 % mol	1.604
Water	0.184 % mol	2.008

The thermal load of the natural gas can be defined as the amount of heat that must be removed from it, by refrigeration to reduce it up to the desired temperature. We are going to treat with a mix of fluid flow between natural gas and ethanol. However, it is not necessary to consider marginal components of natural gas with the content below 0.2%, according to the requirements of the company. Having stated this, we should understand how these components affect the power that evaporator needs to transfer heat from the fluid to the refrigerant (the cooling load), and how each component is going to behave with the transfer of heat, mainly in the difference of temperature, that could be translated in the change of phase. One question that arises is whether is important to consider the ethanol or not for the calculation of cooling load.

As a fact, it is important to know that ethanol doesn't react with the component of the mixture unless water, implying that the amount of ethanol that will be injected in the system, depends on the amount of water contained on the gas mixture. As it is mentioned before, deep details are shown in Chapter 6 of the *Design of a low-temperature drying system for the treatment of raw natural gas* (Jorge Sierra). Firstly it is known that the system is going to manage a maximum and a minimum of natural gas flow rate. Therefore it means that the cooling system has to be capable of managing a range of cooling load that both flow rates are going to demand. For the calculation of Cooling Load, we are going to use equation (4.1). Also, it is important to be aware that the cooling system must have a temperature control range from -19[°C] to -13[°C]. For the flow rate we have the following values:

$$V_{IN}max = 25000 [\text{Nm}^3. \text{day}^{-1}] \rightarrow \dot{m}_{gas}max = 0.2115 [\text{kg}. \text{s}^{-1}]$$

$$V_{IN}min = 10000 [\text{Nm}^3. \text{day}^{-1}] \rightarrow \dot{m}_{gas}min = 0.0846 [\text{kg}. \text{s}^{-1}]$$

Equation for Cooling Load (4.1):

$$\dot{Q}_{in} = \dot{m}_{gas} \cdot \sum m_f \cdot cp(T_{IN} - T_{OUT}) \quad (4.1)$$

Where:

$$T_{IN} = -5 [^{\circ}\text{C}] = 268.15 [K]$$

$$T_{OUT} = -19 [^{\circ}\text{C}] = 254.15 [K] (\text{for } \dot{Q}_{in} \text{ max})$$

$$T_{OUT} = -13 [^{\circ}\text{C}] = 260.15 [K] (\text{for } \dot{Q}_{in} \text{ min})$$

$$\dot{m}_{gas} = \dot{m}_{gas}max \text{ and } \dot{m}_{gas}min$$

$$m_f = \text{molar fraction [\%]} \text{ described in table 4.1}$$

$$cp = \text{Specific heat capacity [kJ}. \text{kg}^{-1}. \text{K}^{-1}] \text{ described in table 4.1}$$

The cooling load required then result as:

$$\dot{Q}_{in}max = 6.850 \text{ [kW]}$$

$$\dot{Q}_{in}min = 2.739 \text{ [kW]}$$

Previously, it was stated that gas would be mixed with ethanol. Hence let's do a comparison of how much does the ethanol affects the cooling load required. All the parameters were taken from chapter 6 (Sierra, 2020) of the first part thesis.

Equation (4.2) stated how much heat does ethanol has to reject to reach the temperature -19°C.

$$\dot{Q}_{in\_ethanol} = \dot{m}_{ethanol} \cdot cp_{ethanol} \cdot (T_{INe} - T_{OUT2}) \quad (4.2)$$

Where:

$$\dot{m}_{ethanol} = \text{mass flow of ethanol} = 5.591 \text{ E } 10^{-5} \text{ [kg. s}^{-1}\text{]}$$

$$cp_{ethanol} = \text{Specific heat capacity} = 2.4 \text{ [kJ. kg}^{-1}\text{. K}^{-1}\text{]}$$

$$T_{INe} = 37 \text{ [}^\circ\text{C]}$$

$$T_{OUT2} = -19 \text{ [}^\circ\text{C]}$$

Then the cooling load that ethanol needs is:

$$\dot{Q}_{in\_ethanol} = 0.00751 \text{ [kW]}$$

To conclude, we can state that this value is very low that we can say is negligible.

#### 4.4 Analysis of the ideal vapor-compression refrigeration cycle

In the first instance, an analysis of the ideal cycle by steam compression is performed.

Assumptions

- Pressure drops are neglected through the lines of the system, product of friction.
- It is assumed that it does not occur the exchange of heat with the surrounding environment.
- The Volumetric and isentropic efficiency of compressors is 100%.

R404a refrigerant is going to be considered for the initial conditions, operating under the following parameters:

- Evaporation Temperature = -27 [°C]
- Condensation Temperature = 42 [°C]
- Maximum cooling load required = 6.850 [kW]
- Minimum cooling load required = 2.739 [kW]

The range of temperature is known due to the differential thermal jump 8 °C for the evaporator and 5 °C for condenser explained in chapter 3.

The properties of the refrigerant R404A are obtained according to the points that correspond to each of the processes of the cooling compression cycle, see figure (4.2) and table (4.1) below, which were calculated from Engineering Equation Software (EES).

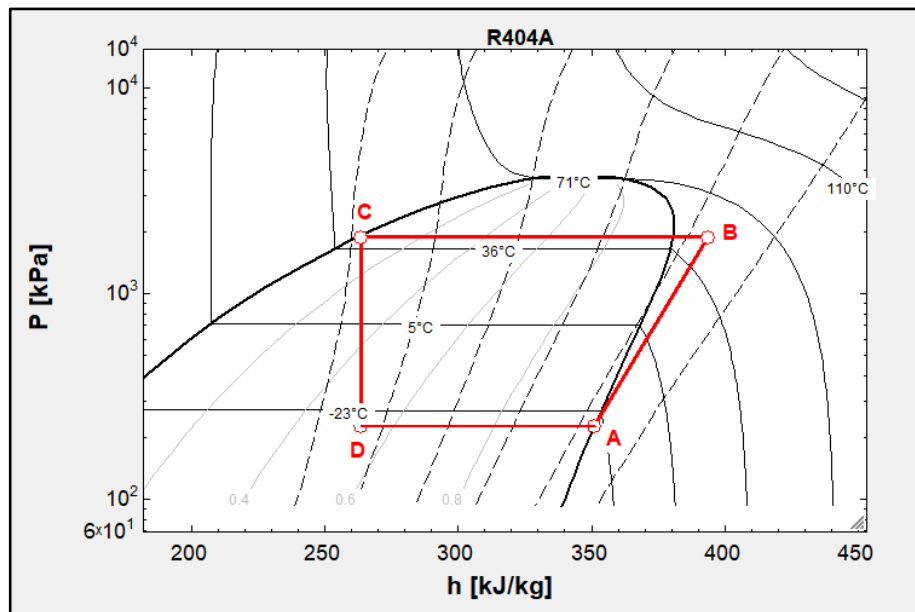


Figure 4.2 - Pressure-enthalpy diagram R-404a for the ideal cycle [EES]

Table 4.2 - R-404a refrigerant properties for the ideal cycle [author]

Stage	(P) Pressure absolute [kPa]	(T) Temperature [°C]	(h) Spec. Enthalpy [kJ/kg]	(s) Spec. Entropy [kJ/kg.K]
A	230	-27	351.18	1.6242
B	1904 (table)	51.07 (by EES)	393.7 (by EES)	1.6242
C	1904	42	263.53	1.2116
D	230	-27	263.53	1.267 (by EES)

As well, the data of the different analysis points are obtained from the tables, according to the Annex A

Now the different variables typical of the steam compression cooling cycle are calculated. As a first instance, it is going to be used as the maximum cooling load for the first calculations. These variable are the following:

- **Cooling effect**

It is the increase of the coolant enthalpy in the evaporator and represents the amount of heat removed from the medium that should be cooled for each kilogram of coolant flowing through the system.

$$C.E = h_A - h_D \quad (4.3)$$

Where:

$C.E$  = cooling effect [kJ.kg<sup>-1</sup>]

$h_A$  = enthalpy on stage A, at the outlet of evaporator = 351.18 [kJ.kg<sup>-1</sup>]

$h_D$  = enthalpy on stage D, at the inlet of evaporator = 263.53 [kJ.kg<sup>-1</sup>]

Then

$$C.E = 87.65[\text{kJ} \cdot \text{kg}^{-1}]$$

- **Mass flow of refrigerant**

$$\dot{m}_{ref} = \frac{\dot{Q}_{in}}{C.E} \quad (4.4)$$

Where:

$$\dot{m}_{ref} = \text{mass flow of refrigerant} [\text{kg} \cdot \text{s}^{-1}]$$

$$\dot{Q}_{in} = \text{cooling capacity} = 6.85[\text{kW}] (\text{from eq. 4.1})$$

$$C.E = \text{cooling effect} = 87.62 [\text{kJ} \cdot \text{kg}^{-1}] (\text{from eq. 4.3})$$

Then

$$\dot{m}_{ref} = 0.07815[\text{kg} \cdot \text{s}^{-1}]$$

- **Work of compressor**

$$W_{comp} = h_B - h_A \quad (4.5)$$

Where:

$$W_{comp} = \text{work done by compressor} [\text{kJ} \cdot \text{kg}^{-1}]$$

$$h_B - h_A = \text{increase in coolant enthalpy in the compressor} [\text{kJ} \cdot \text{kg}^{-1}]$$

Result as:

$$W_{comp} = 42.52 [\text{kJ} \cdot \text{kg}^{-1}]$$

- **Theoretical or technical power required by the compressor**

The power required to operate the compressor can be found from compression work and mass flow.

$$Q_{comp} = \dot{m}_{ref} \cdot W_{comp} \quad (4.6)$$

Where:

$$\dot{m}_{ref} = \text{mass flow of refrigerant} = 0.07815[\text{kg} \cdot \text{s}^{-1}]$$

$$W_{comp} = 42.52 [\text{kJ} \cdot \text{kg}^{-1}] (\text{from eq. 4.5})$$

Result as:

$$Q_{comp} = 3.32 [\text{kW}]$$

- **Theoretical displacement or volume flow rate required by the compressor**

It is the volume of gas that the compressor must be able to handle in the ideal cycle

$$V_t = v \cdot \dot{m}_{ref} \quad (4.7)$$

Where:

$$v = 0.084343[\text{kg} \cdot \text{m}^3] \text{specific volume at the inlet of the compressor}$$

Then result as,

$$V_t = 0.395 [\text{m}^3 \cdot \text{min}^{-1}]$$

- **Total heat of rejection in the condenser**

$$Q_{cond} = \dot{m}_{ref} (h_B - h_C) \quad (4.8)$$

Where:

$$h_b = \text{enthalpy on stage B, at the inlet of condenser} = 393.37 [\text{kJ} \cdot \text{kg}^{-1}]$$

$$h_c = \text{enthalpy on stage C, at the outlet of condenser} = 263.53 [\text{kJ} \cdot \text{kg}^{-1}]$$

Then

$$Q_{cond} = 10.15 \text{ [kW]}$$

- **Coefficient of Performance (COP)**

$$COP = \frac{Q_{evap}}{Q_{comp}} \quad (4.9)$$

$$Q_{evap} = Q_{in} = \text{cooling capacity} = 6.85 \text{ [kW]} \text{ (from eq. 4.1)}$$

$$Q_{comp} = \text{Theoretical power required by the compressor} = 3.32 \text{ [kW]} \text{ (from eq 4.6)}$$

Result as

$$COP = 2.06$$

- **Quality**

It's the vapor mass fraction of the two-phase flow of refrigerant. It is an important parameter for the design of the evaporator.

$$X = \frac{h_c - h_{fD}}{h_{gD} - h_{fD}} \quad (4.10)$$

Where:

$$h_c = \text{enthalpy on stage C, at the outlet of condenser} = 263.53 \text{ [kJ. kg}^{-1}\text{]}$$

$$h_{fD} = \text{enthalpy of sat. liquid at stage D} = 163.81 \text{ [kJ. kg}^{-1}\text{]} \text{ (see Annex A)}$$

$$h_{gD} = \text{enthalpy of sat. vapor at stage D} = 351.18 \text{ [kJ. kg}^{-1}\text{]} \text{ (see Annex A)}$$

Therefore

$$X = 53.22 \text{ [%]}$$

- **Pressure Ratio**

It corresponds to the ratio between suction pressure and discharge. If the pressure ratio is high, it affects the volumetric efficiency.

$$R_c = \frac{P_B}{P_A} \quad (4.11)$$

$$R_c = 8.28$$

The calculations made before are summarized in table (4.3) as important variables of operation for the design of the cooling system due to the ideal analysis. It can be seen a comparison of the values using the maximum cooling load and the minimum one.

**Table 4.3 - Analysis of operating variables for the ideal cycle [author]**

Parameters	Qmax	Qmin
Cooling effect (eq 4.3)	87.65 [kJ. kg <sup>-1</sup> ]	87.65 [kJ. kg <sup>-1</sup> ]
Cooling Capacity of Evaporator (eq 4.1)	6.850 [kW]	2.739 [kW]
Mass flow of refrigerant (eq 4.4)	0.07815 [kg. s <sup>-1</sup> ]	0.03125 [kg. s <sup>-1</sup> ]
Compressor Work (eq 4.5)	42.52 [kJ. kg <sup>-1</sup> ]	42.52 [kJ. kg <sup>-1</sup> ]
Theoretical power required by the compressor (eq 4.6)	3.32 [kW]	1.33 [kW]
Displacement of the compressor or Volume flow rate (eq 4.7)	23.7 [m <sup>3</sup> . h <sup>-1</sup> ]	9.48 [m <sup>3</sup> . h <sup>-1</sup> ]
Heat Rejection in the Condenser (eq 4.8)	10.15 [kW]	4.07 [kW]
Coefficient of Performance (eq 4.9)	2.06 [-]	2.06 [-]
Quality (eq 4.10)	52.21%	52.21%
Pressure Ratio (eq 4.11)	8.28 [-]	8.28 [-]

The comparison was made, to understand that the system needs to be adapted to the range of these conditions. Due to the cooling load that could change, the mass refrigerant delivers to the evaporator also have to vary. The important key roles are going to be through the valve expansion, and also the compressor. In the following chapters, these criteria are going to help to select the best option in which those devices could work on those ranges.

#### 4.5 Analysis of the real cycle operation

The ideal system cycle, as a first insight, is convenient for an approximation that shows how should behave the cooling system but is far from what could happen under real operating conditions, since for a real cycle they must be taken into account certain design parameters.

- Subcooling and superheating of the refrigerant.
- Pressures in the system.
- Compressor efficiency.

Superheating is the process in the evaporation in which no liquid refrigerant is present, therefore the temperature of the vapor increase and becomes superheated. [22]

Vapor superheating prevents the return of liquid to the compressor and generates a slight increase in the cooling effect. On the other hand, it has been shown that when the liquid is cooled the cooling effect of the system increases, this without increasing the energy supply to the compressor. In low temperature, where evaporation is generally about -25 °C to -18 °C, return gas temperature was varied from -10 °C to the temperature suggested in AHRI 540 (Air-Conditioning Heating and Refrigeration Institute), it means that superheating can be in the range of 10 °C. Also according to the fabricator, Emerson Copeland to insure proper compressor cooling and lubrication the minimum operating compressor superheat is 20 °F (11.11°C). [25] Furthermore, they also stated that normally accepted design practice is to use as design criteria a suction line pressure drop equivalent to a 2°F (1.11°C) same as for liquid line in the outlet of the condenser.[26]

For design purposes, the values shown in table (4.4) have been considered.

**Table 4.4 - Design values for the sub cooling and superheating system [author]**

<b>Subcooling in the condenser (°C)</b>	1
<b>Superheating in the evaporator (°C)</b>	10
<b>Superheating on the suction line (°C)</b>	1

Once the actual operating variables are established, the corresponding analysis of the system as a real cycle is performed, including the consideration of the subcooling and superheating values.

For this purpose, the software CoolSelector2 (CS2) from the manufacturer Danfoss will be used considering the first parameters for the ideal cycle of the maximum cooling load. Figure (4.2) shown below corresponds to R-404 refrigerant properties according to each cycle analysis point in real operation.



Point	Description	Temperature [°C]	Pressure [bar]	Density [kg/m <sup>3</sup> ]	Enthalpy [kJ/kg]	Entropy [kJ/(kg·K)]
1	Compressor suction	-16.0	2.310	11.3	360.1	1.66
2	Compressor discharge (estimated)	74.6	19.07	81.06	422.7	1.711
2s	Condensation dew point	42.0	19.07	107.3	380.9	1.584
3s	Condensation bubble point	41.6	19.07	956.3	262.8	1.209
3a	Condenser out	40.6	19.07	962.2	261.1	1.204
3	Including additional subcooling	40.6	19.07	962.2	261.1	1.204
4	After expansion valve	-27.3	2.310	22.72	261.1	1.258
4s	Evaporation bubble point	-27.7	2.310	1249	163.2	0.8593
1s	Evaporation dew point	-27.0	2.310	11.97	350.7	1.622
1a	Evaporator out	-17.0	2.310	11.36	359.2	1.656

Figure 4.2 - R-404a refrigerant properties for the real cycle [CS2]

In figure (4.3) below, it is shown the pressure-enthalpy diagram. We can see that the superheating process of 9°C, occurs from 1s to 1a at the outlet of the evaporator, meaning that the superheating process occurs inside the evaporator. Then the suction suffers superheating in its lines, making the gas enters the compressor at -16°C. Also, we can see that the 1°C subcooling of the saturated liquid in the condenser occurs from 3s to 3a.

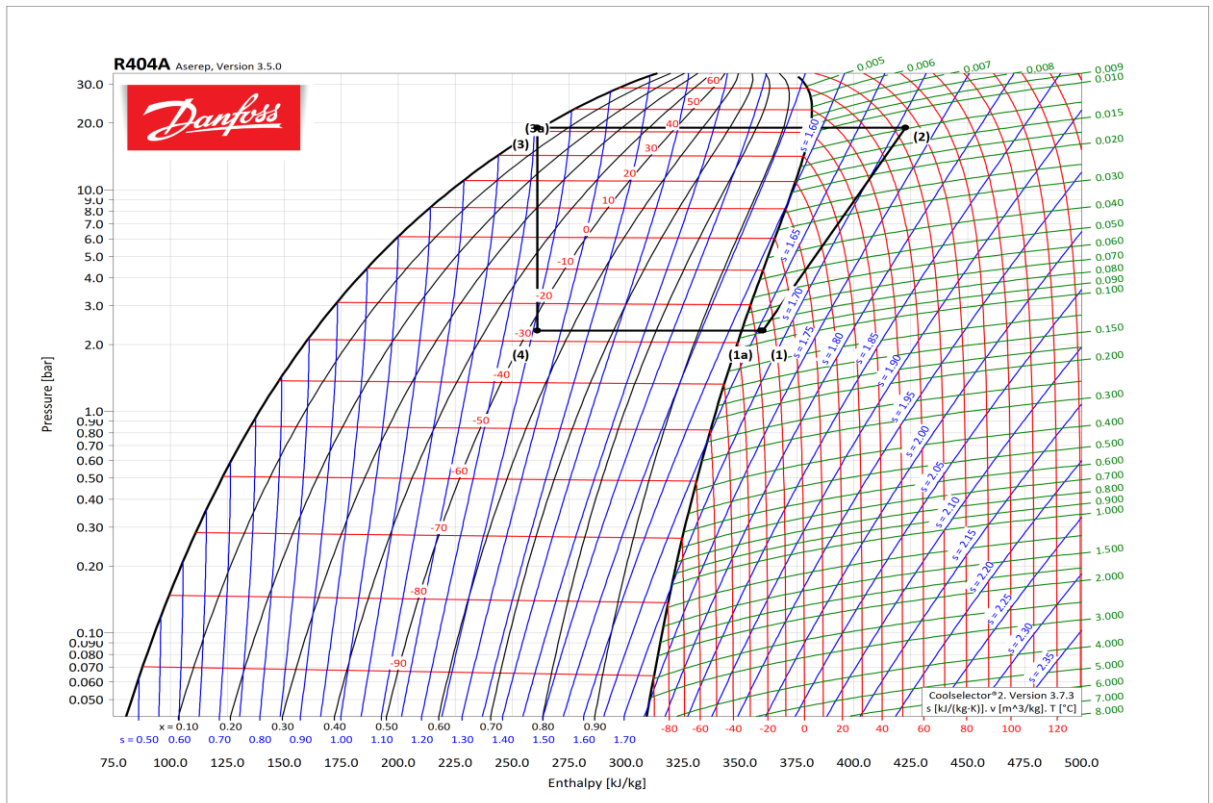


Figure 4.3 - Pressure-enthalpy diagram R-404a for real cycle by [CS2]

For a clear view, figure (4.3) is shown in Annex B

In this way, it can be compared to the values of the parameters that were obtained from the ideal cycle analysis and the real cycle. For this purpose, the table (4.5) is shown below.

**Table 4.5 – Analysis of operating variables for the real cycle [author]**

Parameters	Qmax
Cooling effect (eq 4.3)	98.1 [kJ. kg <sup>-1</sup> ]
Cooling Capacity of Evaporator (eq 4.1)	6.90 kW]
Mass flow of refrigerant (eq 4.4)	0.0704 [kg. s <sup>-1</sup> ]
Compressor Work (eq 4.5)	62.60 [kJ. kg <sup>-1</sup> ]
Technical power required by the compressor (eq 4.6)	4.41 [kW]
Displacement of the compressor or Volume flow rate (eq 4.7)	22.42 [m <sup>3</sup> . h]
Heat Rejection in the Condenser (eq 4.8)	11.37 [kW]
Coefficient of Performance (eq 4.9)	1.56 [-]
Quality (eq 4.10)	51.3%
Pressure Ratio (eq 4.11)	8.25 [-]

We can observe that the Power required of the Compressor has slightly increased, which affect also the Coefficient of Performance. That means for each watt of electric power or joule of energy input into the system, the compressor produces 1.56 watts of cooling power or joules of energy. There is also an increase in the cooling effect, due to the subcooling of the liquid occurred in point 3. Table (4.6) is shown for a clear comparison between both cycles.

**Table 4.6 – Comparative analysis between the operating variables of the ideal cycle and the cycle real [author]**

Parameters	Ideal Cycle	Real Cycle
Cooling effect (eq 4.3)	87.65 [kJ. kg <sup>-1</sup> ]	98.1 [kJ. kg <sup>-1</sup> ]
Cooling Capacity of Evaporator (eq 4.1)	6.850 [kW]	6.90 kW]
Mass flow of refrigerant (eq 4.4)	0.07815 [kg. s <sup>-1</sup> ]	0.0704 [kg. s <sup>-1</sup> ]
Compressor Work (eq 4.5)	42.52 [kJ. kg <sup>-1</sup> ]	62.60 [kJ. kg <sup>-1</sup> ]
Power required by the compressor (eq 4.6)	3.32 [kW]	4.41 [kW]
Displacement of the compressor or Volume flow rate (eq 4.7)	23.7 [m <sup>3</sup> . h <sup>-1</sup> ]	22.42 [m <sup>3</sup> . h <sup>-1</sup> ]
Heat Rejection in the Condenser (eq 4.8)	10.15 [kW]	11.37 [kW]
Coefficient of Performance (eq 4.9)	2.06 [-]	1.56 [-]
Quality (eq 4.10)	53.22%	51.3%
Pressure Ratio (eq 4.11)	8.28 [-]	8.25 [-]

## 5 Design and Selection of Compressor

Once the total thermal load values have been estimated, it is possible to analyze and select a compressor which must be able to move that load. We have to take into consideration that choosing the right compressor is critically important. Improving energy efficiency as well as the reliability of the refrigeration system is essential criteria for the design of the compressor.

### 5.1 Type of compressor

The following table summarizes some advantages and disadvantages of the common type of compressor used in the refrigeration system.

**Table 5.1 - Comparative table of principal types of compressors [27][28]**

Type of Compressor	Advantage	Disadvantage
Reciprocating compressors (positive displacement)	<ul style="list-style-type: none"> <li>* Are very common. They can be found in all models - open, airtight and semi-hermetic - and in all powers, from the lowest to the highest.</li> <li>* They are very durable and can be used in nearly any application in most environments.</li> </ul>	<ul style="list-style-type: none"> <li>* They are not the most energy-efficient options around.</li> <li>* They can be difficult to repair and maintain due to their complexity.</li> </ul>
Scroll compressor (Rotary)	<ul style="list-style-type: none"> <li>* Are generally known to save energy costs over time and use 10-30% less energy than reciprocating</li> <li>* Capable of handling more refrigerant in less time with less power</li> <li>* Low noise</li> </ul>	<ul style="list-style-type: none"> <li>* They are not easy to be disassembled for maintenance because they are fully hermetic.</li> </ul>
Screw compressor (Rotary)	<ul style="list-style-type: none"> <li>* They offer excellent performance and have a power range of between 20 and 1,200 kW.</li> </ul>	<ul style="list-style-type: none"> <li>* They are more expensive and take up more space than piston compressors.</li> </ul>

Despite the power range characterized by each compressor, it could vary depending on the literature, most of them show an average that is common for cooling systems and can be used for design considerations.

### 5.2 Compressor Operating Conditions

As part of the general criteria of the project, it is decided to opt for compressors of the Copeland® brand. An analysis of the different compressor models from this brand is performed for low-temperature applications in refrigeration. According to the manufacturer Copeland®, the scroll compressor features 20% more efficient than piston technologies. Besides, as mentioned before, it has greater reliability in its correct operation and greater longevity of its

useful life, since it does not have suction or discharge valves. For this case we have a total maximum load of 6.85kW according to the calculation made above, therefore this value is used as an initial selection parameter. The parameters for the compression process in the cycle are summarized in table (5.2).

**Table 5.2 - Compressor operating conditions required for the system [author]**

<b>Refrigerant type</b>	R404A
<b>Cooling Capacity of Evaporator</b>	6.850 [kW]
<b>Type of compressor</b>	Scroll
<b>Voltage Line</b>	380-420V/3/50Hz
<b>Discharge Pressure</b>	1907 [kPa]
<b>Suction Pressure</b>	231 [kPa]
<b>Evaporation Temperature</b>	-27[°C]
<b>Condensation Temperature</b>	42[°C]

In the Czech Republic, the standard voltage is 230 V and the frequency is 50 Hz for domestic connection. But the common type of power supply in the industry is 400V/3/50Hz. “For evaporator capacity also it has to be considered the cooling capacity generated from the evaporator capacity as well as the heat gained in the line between the exit of the evaporator to the suction of the compressor”, as it is stated on the Emerson Making Sense Webinar Series[29].

### 5.3 Selection of Compressor

Having calculated the Compressor Capacity and according to the requirements given in the previous table (5.2), the appropriate compressor model is selected. This is done using the information provided by the manufacturer. In Annex C it can be found the specifics parameters according to the catalog of the manufacturer.



**Figure 5.1 - Compressor ZF33K4E-TWD [Annex C]**

**Table 5.3 - Compressor ZF33K4E-TWD operating parameter [Annex C]**

<b>Performance at Specified Point</b>	<b>-27°C/42°C</b>
Compressor Model	ZF33K4E-TWD
Evaporator Capacity	8.26 [kW]
Power	6.52 [kW]
COP	1.27[-]
Current at 400V	13.35 [A]
Suction Mass Flow	0.0842 [kg.s <sup>-1</sup> ]
Isentropic Eff	59.77%
Refrigerant	R404A

- **Efficiency of compressor**

Once the compressor is selected, the respective correction of the isentropic performance values must be performed. In this case, the isentropic efficiency is defined as the ratio of isentropic power (constant entropy compression between suction and discharge pressures) to the electrical power input. The equation (5.2) states the calculation of isentropic efficiency for the calculation of nominal power consumption.

$$\eta_{ise} = \frac{Q_{comp_{ise}}}{Pel_{comp}} \quad (5.2)$$

$$Q_{comp_{ise}} = Pel_{comp} * \eta_{ise} \quad (5.3)$$

Where:

$Q_{comp_{ise}}$  = isentropic power of the compressor [kW]

$\dot{Q}el_{comp}$  = electrical power input = 6.52 [kW](table 5.3)

Therefore

$$Q_{comp_{ise}} = 3.89[\text{kW}]$$

Two more other parameters are going to be introduced to compare the efficiency and economic evaluation of the compressor. These are volumetric efficiency and specific energy consumption.

- **Volumetric Efficiency**

Volumetric efficiency is known as the ratio of the actual displacement of the compressor to the piston displacement. The higher the compression ratio, the lower the volumetric efficiency will be. It can be stated as equation (5.4).

$$\eta_{vol} = \frac{\text{real volume flow rate}}{\text{theoretical volume flow rate}} [\%] \quad (5.4)$$

According to the table (5.3), we can see that the suction mass flow is 0.0842 [kg.s – 1]. Therefore it could be calculated the real volume flow rate at suction condition (-16°C/2.310 bar) as a formula  $V_t = \frac{\dot{m}_{ref}}{\rho} = \frac{(0.0842)}{11.3(\text{see figure 4.2})} = 0.00745 \frac{m^3}{s} = 26.8 \frac{m^3}{h}$

And the theoretical displacement of the compressor is 28.8 [m<sup>3</sup>.h<sup>-1</sup>] according to the catalog [see annex C]. Meaning that:

$$\eta_{vol} = \frac{26.8[\text{m}^3 \cdot \text{h}^{-1}]}{28.8[\text{m}^3 \cdot \text{h}^{-1}]} = 93\%$$

- **Specific energy consumption**

It is an economic evaluation of the working process and it tells us how much energy is needed to produce one cubic meter of compressed gas. Equation (5.5) show us how to calculate it.

$$c = \frac{Pel_{comp}}{\dot{V}_d} \quad (5.5)$$

Where

$C$  = specific energy consumption [kWh. m<sup>-3</sup>]

$Pel_{comp}$  = electrical power input = 6.52 [kW](see table 5.3)

$\dot{V}_d$  = volume flow rate at suction condition = 26.8 [m<sup>3</sup>. h<sup>-1</sup>]

Resulting in:

$$c = 0.24 \text{ [kWh. m}^{-3}\text{]}$$

It means it is needed 0.24 [kW] to produce one cubic meter of compressed gas rated at the suction condition (-16°C/2.310 bar) and at a discharge pressure of 19,07 bar (see figure 4.2)

## 5.4 Temperature Range Control

It has been stated that the system has to be designed for a temperature control range between -13[°C] and -19 [°C]. The common type of regulator used for this purpose is the thermostat.

When the temperature of the medium to be cooled rises above the expected value, in this case, -13°C rises to be -12°C, the compressor must be started until the temperature falls below the set value, -19°C; at this time the compressor will stop. As a suggestion, it can be used a Thermostat from the manufacturer Danfoss®, which offers a wide range of models. Figure (5.2) show the model KP61 that can be used for this type of application.



Figure 5.2 - KP61 Thermostat from Danfoss [30]

Table 5.4 - Thermostat KP61 Characteristics [30]

<b>Product name</b>	KP61 Thermostat
<b>Sensor type</b>	Capillary tube sensor
<b>Charge type</b>	Vapor
<b>Contact function</b>	SPDT
<b>Temperature Range</b>	-30.0 [°C] to -15 [°C]
<b>Differential type</b>	Adjustable

## 6 Selection of Thermal Expansion Valve

Different manufacturers use different recommendations. Normally TEV is sized allowing 20% over the nominal capacity of refrigeration. For the correct dimension of a thermostatic expansion valve for the refrigerating system, it is going to be used as the design parameters we calculate previously for the cycle of refrigeration. These parameters are shown below.

- Type of refrigerant = R404A
- Evaporator capacity = 6.85 [kW]
- Evaporating temperature/pressure = -27 [°C] / 231 [kPa]
- Minimum condensing temperature/pressure = 42 [°C] / 1907 [kPa]
- Liquid refrigerant temperature at valve inlet = 40.6 [°C]
- Pressure drop in the liquid line, distributor and evaporator =  $\Delta p$

First of all, it is necessary to determine the pressure drop across the valve. It is calculated using the equation (6.1)

$$\Delta p_{tot} = p_c - (p_e + \Delta p) \quad (6.1)$$

Where:

$$p_c = \text{condensing pressure} = 1907[\text{kPa}] = 19.07[\text{bar}]$$

$$p_e = \text{evaporating pressure} = 231[\text{kPa}] = 2.31[\text{bar}]$$

$$\Delta p = \text{sum of pressure drops in the liquid line, distributor and evaporator}$$

According to Chapter 1 - Pressure drop considerations of ASHRAE Handbook 2010, “refrigeration systems are typically sized for pressure losses of 1 K” [31]. Therefore let's assume  $\Delta p = 0.8 \text{ bar}$ .

$$\Delta p_{tot} = 19.07 - (2.31 + 0.8)$$

$$\Delta p_{tot} = 15.96 [\text{bar}]$$

The next step is to consider subcooling value, then choose the correction factor according to figure (6.1), which is a source from the Danfoss Catalogue for Selections of the Expansion valve.

Refrigerant	Subcooling [K]										
	2	4	10	15	20	25	30	35	40	45	50
R22	0.98	1	1.06	1.11	1.15	1.20	1.25	1.30	1.35	1.39	1.44
R134a	0.98	1	1.08	1.13	1.19	1.25	1.31	1.37	1.42	1.48	1.54
R404A/R507	0.96	1	1.10	1.20	1.29	1.37	1.46	1.54	1.63	1.70	1.78
R407C	0.97	1	1.08	1.14	1.21	1.27	1.33	1.39	1.45	1.51	1.57

Figure 6.1 - Subcooling correction factor [32]

That means that if the subcooling value is 1K, we might choose 0.96 as a correction factor for refrigerant R404A.

Then it can be determined the required valve capacity using the following equation (6.2)

$$Q_{TEV} = \frac{Q_{evap}}{\text{correction factor}} \quad (6.2)$$

Where:

$$Q_{evap} = \text{Cooling capacity}$$

Therefore,

$$Q_{TEV} = 7.13[\text{kW}]$$

Another important parameter that has to be considered is the minimum cooling load that the system is going to work. For that reason, it has to be secure that the expansion valve could work between those range of capacity, between 6.85 [kW] and 2.739 [kW]

Finally, it can be selected from the catalog the correct valve according to the capacity calculated previously, and it is done by determining the orifice according to the manufacturer tables. It is going to be selected in the TGE series. These models present a fixed orifice size. Figure (6.2) show the model TGE 10-6. Which means that it has an orifice size of 6.



Figure 6.2 - Thermostatic expansion valve TGE 10-6 [CS2]

Table 6.1 - Thermostatic expansion valve 067N6175 Characteristics [CS2]

<b>Product name</b>	Thermostatic expansion valve TGE 10-6
<b>Code</b>	067N6175
<b>Type</b>	TGE
<b>Refrigerants</b>	R404A/R507
<b>Inlet size</b>	½ [in] =12.7[mm]
<b>Inlet connection type</b>	Solder, ODF
<b>Outlet size</b>	5/8 [in] =15.875[mm]
<b>Outlet connection type</b>	Solder, ODF
<b>Orifice capacities</b>	4[TR]

Value	Unit	Inlet	Outlet	Difference
Pressure	bar	19.07	3.110	-15.96
Temperature	°C	40.6	-19.6	-60.3
Bubble point temperature	°C	41.6	-19.9	-61.6
Dew point temperature	°C	42.0	-19.3	-61.3
Density	kg/m <sup>3</sup>	962.2	32.51	-929.7
Enthalpy	kJ/kg	261.1	261.1	0
Quality	-	0.00	0.48	0.48
Velocity	m/s	0.89	15.34	14.44
Mass flow	kg/h	253.2	253.2	0

Figure 6.3 - Performance detail of TGE 10-6 [CS2]



Table (6.1) shows the characteristics of the thermostatic expansion valve that has been selected, while figure (6.3) shows the performance detail of the valve. If we compare the outlet pressure after the expansion occurs, from the previous analysis made in chapter 4, we are going to see that figure (4.2) describes a pressure of 2.31 [bar] instead of 3.110 [bar] described on the figure (6.3). This is due to the considerations of the pressure drop of 0.8[bar] along with the systems. Because the fluid in the system undergoes several processes, in which friction on some loss heat is involved, it is impossible to neglect pressure drops in the line. Figure (6.4) show also the maximum and minimum capacity in which the valve work, meaning that in effect it meets the requirements of the design. These results were taken from the Software CoolSelector from Danfoss. In Annex D is shown the Datasheet results.

Type	TGE 10 - 3	TGE 10 - 4	TGE 10 - 6	TGE 10 - 8	TGE 10 - 9
NS	12.7	12.7	12.7	12.7	12.7
Range	N	N	N	N	N
Nominal capacity [kW]	3.564	4.786	6.983	9.371	11.11
Min. capacity [kW]	0.891	1.197	1.746	2.343	2.777
Load [%]	192	143	98	73	62
DP [bar]	15.96	15.96	15.96	15.96	15.96
Velocity, in [m/s]	0.89	0.89	0.89	0.89	0.89

**Figure 6.4 - Nominal and minimum capacity of the thermostatic expansion valve TGE 10-6 (Annex D)**

## 7 Design and calculation of the evaporator heat exchanger

The aim of this chapter is on designing and calculating an evaporator heat exchanger, which is used in low-temperature drying systems for the treatment of raw natural gas. Due to the complexities surrounding this task, different considerations and assumptions are taken into account for accomplishing an accurate and efficient design.

The thesis project provided by the company Moravské naftové doly (MND) was divided into two assignments; the first assignment deals with the design of low-temperature drying system for treatment of raw natural gas, and the second assignment is related to the design of a cooling system for low-temperature drying unit for treatment of raw natural gas. Nevertheless, the design and calculations of the evaporator heat exchanger is a shared task, which means that it is written by the authors from the assignments mentioned. Therefore, the considerations and assumptions related to the cooling system and refrigerants are found in the master's thesis design of a cooling system for low-temperature drying unit for the treatment of raw natural gas, and the calculations and parameters from the natural gas are found in the master's thesis design of low-temperature drying system for treatment of raw natural gas.

### 7.1 Design specifications of the evaporator heat exchanger

The designing procedure for the evaporator heat exchanger takes into consideration several aspects to obtain an optimally designed equipment. In this case, the most important factor in the evaporator design is the reduction of the natural gas temperature from  $-5$  [°C] to  $-19$  [°C], produced through the transfer of thermal energy between the natural gas and the liquid refrigerant. Table (7.1) presents the design specifications and parameters for the evaporator heat exchanger design.

**Table 7.1 – Design parameters for the evaporator heat exchanger [author]**

Parameter	Hot side	Cold side
Fluid circulated	Natural gas	R 404A
Flow rate	25000 [Nm <sup>3</sup> · day <sup>-1</sup> ]	0.0704 [kg · s <sup>-1</sup> ]
Type of fluid	Single-phase	Two-phase
Inlet temperature	$-5$ [°C]	$-27$ [°C]
Superheating	N/A	10 [°C]
Quality	N/A	0.513 [-]
Outlet temperature	$-19$ [°C]	$-17$ [°C]
Operating pressure	20 [bar]	2.310 [bar]
Maximum pressure drop	0.01 [bar]	0.8 [bar]
Strength dimension of the evaporator	30 [bar]	30 [bar]

The design of the evaporator requires careful attention on the parameters, type of fluids, requirements, considerations, and assumptions for estimating the correct values, determine the size of the heat exchanger and selecting suitable equipment for the low-temperature drying system.

Having the technical parameters for designing the heat exchanger, it must be mentioned the composition of natural gas, which is presented in the table (7.2).

**Table 7.2 – Composition of natural gas [author]**

Composition	Molar percentage [%]
Methane	96.388 % mol
Ethane	3.428 % mol
Water	0.184 % mol

The parameters previously mentioned are the author's criteria for the initial selection (flow arrangement, performance parameters, geometry, etc). The pressure drop will be the parameter, which guides the authors to the desired performance. The maximum pressure loss value between the inlet and outlet of the process is 80 [kPa], as well as a natural gas flow rate of 25000 [ $Nm^3 \cdot day^{-1}$ ]. The authors also assume a maximum pressure loss of 10 [kPa] in the evaporator heat exchanger, because it must also be considered the pressure losses in the recuperative heat exchanger and pipe system in the low-temperature drying process. The procedure followed by the authors for designing the evaporator heat exchanger to comply with the specified parameters and pressure drop requirements as it is shown below.

- Selection of the heat exchanger type.
- Calculation of log-mean temperature difference (LMTD).
- Dimension estimation and sizing approximation.
- Energy balance on the evaporator heat exchanger.
- Estimation of overall heat transfer coefficient (U-value).
- Calculation of heat transfer area.
- Compare the pressure drop limit.
- Optimize the heat exchanger design.

In the following sections, the procedure followed by the authors for designing the evaporator heat exchanger is explained deeply.

## 7.2 Selection of the heat exchanger type

It is very important to analyse and determine the type of heat exchanger for this natural gas process, and considering the following conclusion from the natural gas industry, which states that “*the experience in the natural gas industry operating shell and tube heat exchangers has shown that the mixture of the condensed water and heavy hydrocarbons at low temperatures - 5 [°C] to -15 [°C] forms hydrates*”[33]. Therefore, the suitable heat exchanger for low-temperature separation systems is a counter-flow plate heat exchanger, which is simple and compact equipment and occupies less floor space, which is important for space-limited facilities. Additionally, turbulent flow within the plate heat exchanger reduces hydrate formation, allowing a better heat transfer area. While in tube heat exchangers, there is a danger of hydrate formation, plug, and damage of the heat exchange tubes.

### 7.3 Dimension estimation and sizing approximation

As an initial assumption, the authors use different catalogues from the company SWEP International AB, which manufactures brazed plate heat exchangers, according to the requirements of the company Moravské naftové doly (MND). In general, heat exchanger designs start with assumptions, considerations, and approximations of some input values.

It is necessary to consider the pressure drop limitation in the system, because this variable indicates the correct dimension of the plate heat exchanger. Figure (7.1) presents the plate dimensions in a plate heat exchanger.

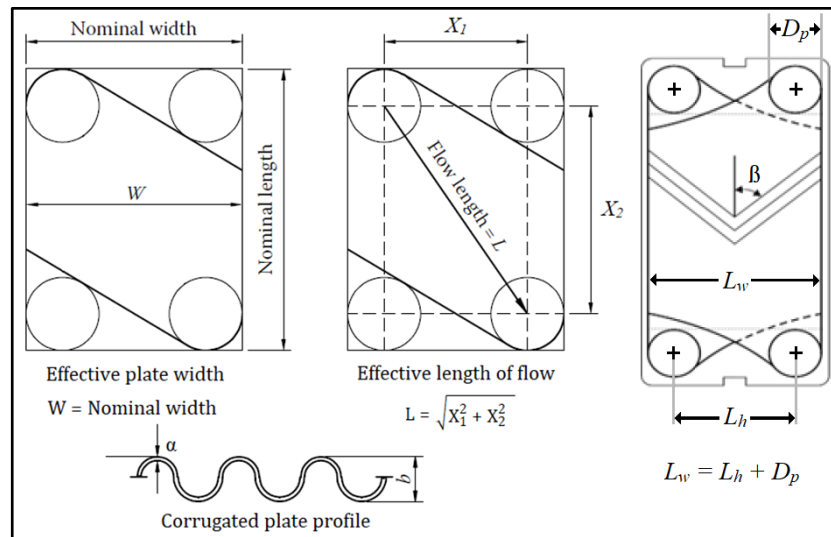


Figure 7.1 – Plate dimensions in a plate heat exchanger [34]

For dimension estimation and sizing approximation, it must be calculated the heat transfer rate, the log-mean temperature difference (LMTD), and estimate the overall heat transfer coefficient (U-value).

- **Calculation of the heat transfer rate**

The heat transfer rate in a basic heat transfer relation for single-pass designs is shown in the equation (7.1).

$$\dot{Q}_{\text{evaporator}} = U \cdot A \cdot (LMTD) \text{ [kW]} \quad (7.1)$$

Where  $U$  is the overall heat transfer coefficient in  $[W \cdot m^{-2} \cdot ^\circ C^{-1}]$ ,  $A$  is the heat transfer area in  $[m^2]$ , and the  $LMTD$  is the log-mean temperature difference in  $[^\circ C]$ .

The energy balance on the evaporator heat exchanger is presented as follows:

$$\dot{Q}_{\text{evaporator}} = 6.850 \text{ [kW]}$$

- **Calculation of the log-mean temperature difference (LMTD)**

According to the equation (7.2), the log-mean temperature difference (LMTD) can be calculated as follows:

$$LMTD = \frac{(T_{\text{hot,in}} - T_{\text{cold,out}}) - (T_{\text{hot,out}} - T_{\text{cold,in}})}{\ln\left(\frac{T_{\text{hot,in}} - T_{\text{cold,out}}}{T_{\text{hot,out}} - T_{\text{cold,in}}}\right)} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} \text{ [}^\circ\text{C]} \quad (7.2)$$

The temperature at the inlet and outlet of the evaporator heat exchanger is  $-5\text{ [}^\circ\text{C]}$  and  $-19\text{ [}^\circ\text{C]}$  respectively, and the temperature at the inlet and outlet of the refrigeration system is  $-27\text{ [}^\circ\text{C]}$  and  $-17\text{ [}^\circ\text{C]}$  respectively. It is important to note that at the evaporator outlet, the refrigerant should totally be at gaseous state, for this reason, a superheating of  $10\text{ [}^\circ\text{C]}$  is considered. The temperature distributions can be represented in figure (7.2).

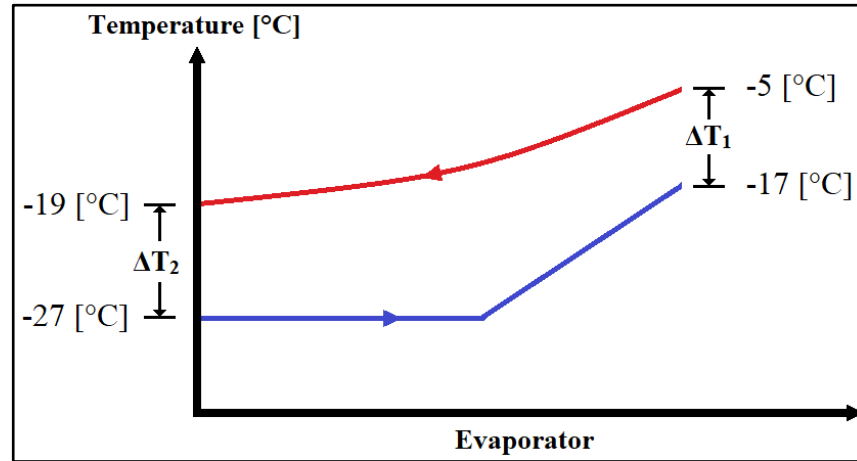


Figure 7.2 – Temperature distribution in the evaporator heat exchanger [author]

Therefore, the log-mean temperature difference (LMTD) is calculated by equation (7.2) as follows:

$$LMTD_1 = \frac{(-5\text{ [}^\circ\text{C]}) - (-17\text{ [}^\circ\text{C]}) - (-19\text{ [}^\circ\text{C]}) - (-27\text{ [}^\circ\text{C]})}{\ln\left(\frac{(-5\text{ [}^\circ\text{C]}) - (-17\text{ [}^\circ\text{C]})}{(-19\text{ [}^\circ\text{C]}) - (-27\text{ [}^\circ\text{C]})}\right)}$$

$$LMTD_1 = \frac{12\text{ [}^\circ\text{C]} - 8\text{ [}^\circ\text{C]}}{\ln\left(\frac{12\text{ [}^\circ\text{C]}}{8\text{ [}^\circ\text{C]}}\right)}$$

$$LMTD_1 = 9.86\text{ [}^\circ\text{C]}$$

However, knowing that exists a complication factor regarding to the fact that the evaporator deals with multiple temperature changes and also considering the existence of a two-phase region in the refrigerant, it can be said that that the average temperature difference could be influenced much more by the temperature difference in the two-phase region (constant temperature of  $-27\text{ [}^\circ\text{C]}$ ) than the superheat region (temperature from  $-27\text{ [}^\circ\text{C]}$  to  $-17\text{ [}^\circ\text{C]}$ ). Having said that, the authors consider for a better approach, a mean temperature between the  $LMTD_1$  with the superheating and  $LMTD_2$  without superheating. Meaning that  $LMTD_2$  without superheating is  $13.839\text{ [}^\circ\text{C]}$ . Therefore, the log-mean temperature difference (LMTD) value used for the design is  $11.85\text{ [}^\circ\text{C]}$ .

$$LMTD = 11.85\text{ [}^\circ\text{C]}$$

- **Estimating of the overall heat transfer coefficient (U-value)**

Basically, by selecting an evaporator heat exchanger from the catalogue, it can be obtained the dimensions and therefore, the area needed for estimating the velocity value, which is very important for calculating the Reynolds number. Therefore, the estimation of the overall heat transfer coefficient is obtained from table (7.3).

**Table 7.3 – Typical overall heat transfer coefficients [35]**

Gasketed-plate exchangers		
Hot fluid	Cold fluid	U [ $W \cdot m^{-2} \cdot ^\circ C^{-1}$ ]
Light organic	Light organic	2500 - 5000
Light organic	Viscous organic	250 - 500
Viscous organic	Viscous organic	100 - 200
Light organic	Process water	2500 - 3500
Viscous organic	Process water	250 - 500
Light organic	Cooling water	2000 - 4500
Viscous organic	Cooling water	250 - 450
Condensing steam	Light organic	2500 - 3500
Condensing steam	Viscous organic	250 - 500
Process water	Process water	5000 - 7500
Process water	Cooling water	5000 - 7000
Dilute aqueous solutions	Cooling water	5000 - 7000
Condensing steam	Process water	3500 - 4500

The table previously mentioned can be used for the preliminary sizing of equipment for starting a detailed thermal design. The overall heat transfer coefficient selected is as follows:

$$U = 350 [W \cdot m^{-2} \cdot ^\circ C^{-1}]$$

Rearranging the equation (7.1), it can be calculated the area by the equation (7.3):

$$A = \frac{\dot{Q}_{evaporator}}{U \cdot LMTD} [m^2] \quad (7.3)$$

$$A = \frac{6,850 [W]}{350 [W \cdot m^{-2} \cdot ^\circ C^{-1}] \cdot 11.85 [^\circ C]} = 1.6515 [m^2]$$

In this case, it was selected the type of heat exchanger shown in figure (7.3), with the basic specifications presented in the table (7.4).



**Figure 7.3 – Evaporator B35TM2[Annex E]**

**Table 7.4 – Basic specifications of B35TM2[Annex E]**

<b>Evaporator model</b>	B35TM2
<b>Max. number of plates (NoP)</b>	260
<b>Max flow</b>	58 [ $m^3 \cdot h^{-1}$ ]
<b>Channel volume</b>	0.18 [ $dm^3$ ]
<b>Material</b>	316 stainless steel plates, copper brazing

Additionally, the standard dimensions of the evaporator B35TM2 are presented in figure (7.4) and table (7.5).

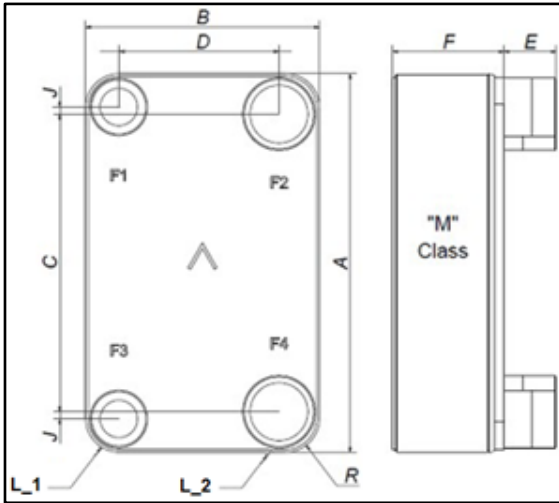


Figure 7.4 – Standard dimensions of Evaporator B35TM2[Annex D]

Table 7.5 – Standard dimensions of evaporator B35TM2[Annex D]

#	[m]
A	0.393
B	0.243
C	0.309
D	0.1665
F	$0.01+0.00226*(NoP)$
G	0.004
J	0.0075
L_1	0.042
L_2	0.062
R	0.035
E_1	0.0271

- **Calculation of the effective plate area**

Using the dimensions previously mentioned in table (7.5), it can be calculated the effective plate area by the equation (7.4) as follows:

$$A_{eff} = L \cdot L_w [m^2] \quad (7.4)$$

Where  $A_{eff}$  is the effective plate area in  $[m^2]$ ,  $L$  is the effective length of flow in  $[m]$ , and  $L_w$  is the distance, which is 0.1665  $[m]$  from table (7.5).

At first, it is necessary to calculate the effective length of flow by the equation (7.5) and the distance  $L_w$  by the equation (7.6) as follows:

$$L = \sqrt{D^2 + C^2} [m] \quad (7.5)$$

Where  $D$  is 0.1665  $[m]$  and  $C$  is 0.309  $[m]$  from table (7.5).

$$L = 0.351 [m]$$

Then from figure (7.1):

$$L_w = L_h + D_p [m] \quad (7.6)$$

Where  $L_h$  is equivalent to  $D = 0.1665 [m]$  from the table (7.5) and  $D_p = (L_1 + L_2)/2 = 0.0525 [m]$  from the table (7.5).

$$L_w = 0.218 [m]$$

Therefore,

$$A_{eff} = 0.07652 [m^2]$$

Having calculated the effective plate area, it can be determined the number of plates by the equation (7.7) as follows:

$$N_p = \frac{A}{A_{eff}} [-] \quad (7.7)$$

$$N_p = 21 [-]$$

Therefore, the number of channels per pass is calculated by equation (7.8).

$$N_{ch} = \frac{N_p - 1}{2} [-] \quad (7.8)$$

Where  $N_{ch}$  is the number of channels in [-] and  $N_p$  is the number of plates, which is 21 [-]. Therefore, the number of channels is:

$$N_{ch} = 10 [-]$$

The calculation of the cross-sectional area is given by the equation (7.9).

$$A_{ch} = B \cdot b [m^2] \quad (7.9)$$

Where  $B$  is the nominal width, which is 0.243 [m] from table (7.5), and  $b$  is the spacing of plate in [m].

And having the volume of the channel, which is 0.00018 [m<sup>3</sup>] from table (7.4) and the effective plate area of 0.07652 [m<sup>2</sup>] from the equation (7.4), the space of plate is calculated by the equation (7.10) as follows:

$$b = \frac{\text{volume of the channel}}{A_{eff}} [m] \quad (7.10)$$

$$b = 0.002352 [m]$$

Therefore, the calculation of the cross-sectional area by equation (7.9) is:

$$A_{ch} = 0.0005716 [m^2]$$

The equivalent diameter is defined by the equation (7.11).

$$D_c = \frac{4 \cdot B \cdot b}{2 \cdot (B + b)} [m] \quad (7.11)$$

Where  $B$  is the nominal width, which is 0.243 [m] from table (7.5), and  $b$  is the spacing of plate, which is 0.002352 [m] from the equation (7.10). Therefore, the equivalent diameter is:

$$D_c = 0.004659 [m]$$

At this point, it can be defined the properties of the raw natural gas and the liquid refrigerant. Due to the complex procedure, it is first stated the natural gas calculations followed by the liquid refrigerant calculations.

## 7.4 Calculation of the overall heat transfer coefficient (U-value)

Several steps must be considered before correcting the overall heat transfer coefficient (U-value). Therefore, the following parameters must be calculated for both types of fluids (raw natural gas and liquid refrigerant).

1. Calculation of Velocity.
2. Calculation of Reynold number.
3. Calculation of Nusselt number.
4. Calculation of the overall heat transfer coefficient (U-value).

First of all, it must be calculated the natural gas parameters.

### 7.4.1 Calculation of natural gas parameters

The following properties of the natural gas components presented in table (7.6) are calculated by the software Engineering Equation Solver (EES). The values  $\mu$  at 260 [K] and 253.15 [K] represent the viscosity at bulk temperature and wall temperature respectively at pressure of 2 000[kPa]. Due to the temperature difference between the natural gas at -19 [°C]



and the plate within the heat exchanger at -27 [°C], it was assumed a mean temperature of -20 [°C] for calculating the viscosity at wall temperature and pressure of 2 000 [kPa]

**Table 7.6 – Properties of natural gas components [EES]**

Natural gas	Molar fraction	$\mu$ at 260 [K] [kg · m <sup>-1</sup> · s <sup>-1</sup> ]	$\mu$ at 253.15 [K] [kg · m <sup>-1</sup> · s <sup>-1</sup> ]	Prandtl number at 260 [K]	Conductivity at 260 [K] [W · m <sup>-1</sup> · K <sup>-1</sup> ]
Methane	96.388 %	0.00001032	0.00001011	0.7737	0.03119
Ethane	3.428 %	0.000008271	0.000008064	0.7872	0.01687
Water	0.184 %	0.000008485	0.000008241	0.9966	0.01585
<b>Total</b>	<b>100 %</b>	<b>0.00001025</b>	<b>0.00001003</b>	<b>0.7746</b>	<b>0.03067</b>

Table 7.6 continues as follows:

Physical properties at 260 [K]	Natural gas
Natural gas pressure, P	2 000 [kPa]
Density, $\rho$	16.274 [kg · m <sup>-3</sup> ]
Viscosity, $\nu$	6.437 · 10 <sup>-7</sup> [m <sup>2</sup> · s <sup>-1</sup> ]
Prandtl number, $P_r$	0.7746 [-]
Conductivity, $k$	0.03066 [W · m <sup>-1</sup> · K <sup>-1</sup> ]

These values summarized in table (7.6) represent the initial parameters for the natural gas calculations.

- **Channel velocity**

Having determined the cross-section area of the channel, we can calculate the channel velocity by the equation (7.12) as follows:

$$V_{ch} = \frac{\dot{m}_{gas}}{\rho \cdot A_{ch} \cdot N_{ch}} [m \cdot s^{-1}] \quad (7.12)$$

Where  $A_{ch}$  is the cross-sectional area, which is 0.0005716 [m<sup>2</sup>] from the equation (7.9),  $N_{ch}$  is the number of channels, which is 10 [-] from the equation (7.8),  $\dot{m}_{gas}$  is the mass flow rate of natural gas, which is 0.2115 [kg · s<sup>-1</sup>] from the table (7.1), and  $\rho$  is the density of the natural gas, which is 16.274 [kg · m<sup>-3</sup>] from the table (7.6).

Therefore,

$$V_{ch} = 2.274 [m \cdot s^{-1}]$$

- **Reynolds number**

The Reynolds number is calculated by the equation (7.13) as follows:

$$Re = \frac{V_{ch} \cdot D_c}{\nu} [-] \quad (7.13)$$

Where  $D_c$  is the equivalent diameter, which is 0.004659 [m] from the equation (7.11),  $V_{ch}$  is the channel velocity, which is 2.274 [m · s<sup>-1</sup>] from the equation (7.12), and  $\nu$  is the kinematic viscosity, which is 6.437 · 10<sup>-7</sup> [m<sup>2</sup> · s<sup>-1</sup>] from the table (7.6).

Therefore,

$$Re = 16\,457.82 [-]$$

- **Nusselt number**

The most widely used heat transfer correlation for turbulent flow is the (Sieder-Tate correlation) and is expressed by the equation (7.14).

$$Nu = 0.374 Re^{0.668} Pr^{0.33} \left(\frac{\mu_b}{\mu_w}\right)^{0.15} [-] \quad (7.14)[34]$$

Where  $Re$  is the Reynolds number, which is 16,836.38 [-] from the equation (7.13),  $\mu_b$  is the bulk temperature viscosity, which is 0.000010245 [ $kg \cdot m^{-1} \cdot s^{-1}$ ] from the table (7.6),  $\mu_w$  is the viscosity at wall temperature, which is 0.00001003 [ $kg \cdot m^{-1} \cdot s^{-1}$ ] from the table (7.6), and the  $Pr$  is the Prandtl number, which is 0.7746 [-] from the table (7.6).

$$Nu = 226.06 [-]$$

- **Heat transfer coefficient**

The heat transfer coefficient is calculated by equation (7.15) as follows:

$$\alpha_{gas} = Nu \cdot \frac{k}{D_c} [W \cdot m^{-2} \cdot ^\circ C^{-1}] \quad (7.15)$$

Where  $Nu$  is the Nusselt number, which is 226.06 [-] from the equation (7.14),  $D_c$  is the equivalent diameter, which is 0.00465962 [ $m$ ] from the equation (7.11), and  $k$  is the conductivity, which is 0.03066 [ $W \cdot m^{-1} \cdot K^{-1}$ ] from the table (7.6). Therefore,

$$\alpha_{gas} = 1487.77 [W \cdot m^{-2} \cdot ^\circ C^{-1}]$$

## 7.4.2 Calculation of refrigerant parameters

Dealing with refrigerant means dealing with a two-phase flow, knowing that the flow enters at liquid state at the inlet and exits at gaseous state at the outlet. The following table (7.7) shows the properties for the refrigerant taking into account both phases. The liquid phase is referred as *refl* and the vapor phase as *refv*.

**Table 7.7 – Physical properties of R 404A [EES]**

Physical properties at 246.15 [K]	R 404A Liquid phase	R 404A Gas phase
Pressure, P	230 [kPa]	230 [kPa]
Density, $\rho$	1,245.2 [ $kg \cdot m^{-3}$ ]	11.856 [ $kg \cdot m^{-3}$ ]
Mass flow rate	0.03440 [ $kg \cdot s^{-1}$ ]	0.036 [ $kg \cdot s^{-1}$ ]
Kinematic viscosity, $\nu$	$1.9924 \cdot 10^{-7}$ [ $m^2 \cdot s^{-1}$ ]	N/A
Bulk dynamic viscosity, $\mu_{ref}$	0.00001055 [ $kg \cdot m^{-1} \cdot s^{-1}$ ]	N/A
Wall dynamic viscosity, $\mu_{refw}$	0.00001052 [ $kg \cdot m^{-1} \cdot s^{-1}$ ]	N/A
Prandtl number	3.801 [-]	N/A
Conductivity, $k$	0.08506 [ $W \cdot m^{-1} \cdot K^{-1}$ ]	N/A

The saturation table (7.8) presents the pressure, specific density, specific volume, specific enthalpy, and specific entropy values for the refrigerant R 404A at different temperatures.

**Table 7.8 – Saturation table R 404A [Annex A]**

Temp. °C	Pressure kPa		spec. Density kg/m <sup>3</sup>		spec. Volume m <sup>3</sup> /kg		spec. Enthalpy kJ/kg			spec. Entropy kJ/kg K	
	liquid	gas	liquid	gas	liquid	gas	liquid	latent	gas	liquid	gas
-30	207,78	202,23	1255,0	10,548	0,0007968	0,094802	159,93	189,51	349,44	0,84601	1,6267
-29	216,41	210,73	1251,8	10,971	0,00079887	0,09115	161,22	188,80	350,02	0,85128	1,6259
-28	225,32	219,51	1248,5	11,407	0,00080097	0,087667	162,52	188,08	350,60	0,85654	1,625
-27	234,50	228,56	1245,2	11,856	0,00080308	0,084343	163,81	187,37	351,18	0,86179	1,6242
-26	243,97	237,89	1241,9	12,32	0,00080521	0,081171	165,11	186,64	351,75	0,86703	1,6234

The proportional flow for each phase is calculated by using the equation (7.16). Where  $x$  is the quality value, which is 0.513 [-] from the table (7.1), and  $\dot{m}_{ref}$  is the mass flow rate value, which is 0.0704 [kg · s<sup>-1</sup>] from the table (7.1).

$$\dot{m}_{ref_l} = (1 - x) \cdot \dot{m}_{ref} \text{ [kg} \cdot \text{s}^{-1}] \quad (7.16)$$

Therefore,

$$\dot{m}_{ref_l} = 0.03440 \text{ [kg} \cdot \text{s}^{-1}]$$

$$\dot{m}_{ref_v} = \dot{m}_{ref} - \dot{m}_{ref_l} = 0.036 \text{ [kg} \cdot \text{s}^{-1}]$$

- **Channel velocity**

The channel velocity for the liquid and gaseous phase is calculated by the equation (7.12) using the properties from the table (7.7).

$$V_{ch} = \frac{\dot{m}_{gas}}{\rho \cdot A_{ch} \cdot N_{ch}} \text{ [m} \cdot \text{s}^{-1}] \quad (7.12)$$

$$V_{ref_l} = 0.004817 \text{ [m} \cdot \text{s}^{-1}]$$

$$V_{ref_v} = 0.5329 \text{ [m} \cdot \text{s}^{-1}]$$

Where  $A_{ch}$  is the cross-sectional area, which is 0.0005716 [m<sup>2</sup>] from the equation (7.9),  $N_{ch}$  is the number of channels, which is 10 [-] from the equation (7.8),  $\dot{m}_{ref}$  is the mass flow rate of refrigerant in the liquid and gaseous phase, which are 0.03440 [kg · s<sup>-1</sup>] and 0.036 [kg · s<sup>-1</sup>] from the table (7.7) respectively, and  $\rho$  is the density of refrigerant in liquid and gaseous phase, which are 1245.2 [kg · m<sup>-3</sup>] and 11.856 [kg · m<sup>-3</sup>] from the table (7.7) respectively.

- **Reynolds number**

The Reynolds number is only calculated for the liquid phase. This is an important parameter for the evaluation of the next steps. Reynold number is calculated by the equation (7.13).

$$Re_{ref_l} = 112.67 \text{ [-]}$$

Where  $D_c$  is the equivalent diameter, which is 0.004659 [m] from the equation (7.11),  $V_{ch}$  is the liquid channel velocity, which is 2.270 [m · s<sup>-1</sup>] from the equation (7.12), and  $\nu$  is the kinematic viscosity of refrigerant, which is 1.9924 · 10<sup>-7</sup> [m<sup>2</sup> · s<sup>-1</sup>] from the table (7.7).

The two-phase heat transfer coefficient could be described as a nucleate boiling regime by the equation (7.17) or convective boiling regime by the equation (7.18), depending on the dominance of the regime. It is defined according to the Shah Correlation. [36]

- **Nucleate boiling regime**

$$\alpha_{TP_{NB}} = 0.6883Co^{-0.2}(1-x)^{0.8}\alpha_{ref_l} + 1058Bo^{0.7}(1-x)^{0.8}Fs\alpha_{ref_l} \quad (7.17)$$

- **Convective boiling regime**

$$\alpha_{TP_{CB}} = 1.136Co^{-0.9}(1-x)^{0.8}\alpha_{ref_l} + 667.2Bo^{0.7}(1-x)^{0.8}S\alpha_{ref_l} \quad (7.18)$$

Where,

$$\alpha_{ref_l} = Nu_{ref_l} \frac{k_l}{D_h} = \text{Heat transfer coefficient based on the liquid phase} \quad (7.19)$$

$$Bo = \frac{Q \cdot A_{ch}}{m_{ref} \cdot h_{fg}} = \text{Boiling number} \quad (7.20)[37]$$

$$Co = \left(\frac{1-x}{x}\right)^{0.8} \sqrt{\frac{\rho_{ref_v}}{\rho_{ref_l}}} = \text{Convection number, Martinelli parameter} \quad (7.21)[38]$$

$Fs$  = Fluid surface parameter

Therefore, before obtaining the two-phase heat transfer coefficient, it is necessary to calculate the other unknown parameters within the equation.

- **Calculation of Nusselt number based on the liquid phase**

The Nusselt number of refrigerant in the liquid state is calculated by the equation (7.14) using the liquid phase parameters.

$$Nu_{ref_l} = 0.374 Re_{ref_l}^{0.668} Pr_{ref_l}^{0.33} \left(\frac{\mu_{ref_l}}{\mu_{ref_{lw}}}\right)^{0.15} [-] \quad (7.14)$$

Where,  $\mu_{ref_{lw}}$  is the wall dynamic viscosity of refrigerant in the liquid state, which is 0.00001052 [ $kg \cdot m^{-1} \cdot s^{-1}$ ] from the table (7.7) and calculated by the software Engineering Equation Solver (EES),  $\mu_{ref_l}$  is the bulk dynamic viscosity of refrigerant in the liquid state, which is 0.00001055 [ $kg \cdot m^{-1} \cdot s^{-1}$ ] from the table (7.7) and calculated by the software Engineering Equation Solver (EES),  $Re$  is the Reynolds number calculated by the equation (7.13), which is 112.67 [-], and  $Pr$  is the Prandtl number, which is 3.801 [-] from the table (7.7).

Therefore,

$$Nu_{ref_l} = 13.647 [-]$$

- **Calculation of the heat transfer coefficient based on the liquid phase**

The heat transfer coefficient based on the liquid phase of refrigerant is calculated by the equation (7.19). Where  $Nu_{ref_l}$  is the Nusselt number of refrigerant in the liquid state, which is 13.647 [-] and calculated by the equation (7.14),  $k_l$  is the conductivity of refrigerant, which is 0.08506 [ $W \cdot m^{-1} \cdot K^{-1}$ ] from the table (7.7), and  $D_c$  is the equivalent diameter, which is 0.004659 [ $m$ ] from the equation (7.11).

Therefore,

$$\alpha_{ref_l} = 249.12 [W \cdot m^{-2} \cdot K^{-1}]$$

- **Calculation of the boiling number**

An empirical correction is used for the single-phase liquid heat transfer coefficient, to predict the two-phase heat transfer known as the boiling number. As it is stated in the equation (7.20).

$$Bo = \frac{Q_{NB} \cdot A_{ch} \cdot N_{ch}}{\dot{m}_{ref} \cdot h_{fg}} [-]$$

Where  $Q_{NB}$  is the boiling heat transfer calculated by the software Engineering Equation Solver (EES), which is 2.831 [kW · m<sup>-2</sup>],  $A_{ch}$  is the cross-sectional area, which is 0.0005716 [m<sup>2</sup>] from the equation (7.9),  $N_{ch}$  is the number of channels, which is 10 [-] from the equation (7.8),  $\dot{m}_{ref}$  is the mass flow rate of refrigerant, which is 0.0704 [kg · s<sup>-1</sup>] from the table (7.1), and  $h_{fg}$  is the specific enthalpy latent, which is 187.37 [kJ · kg<sup>-1</sup>] from the table (7.8).

$$Bo = 0.001227 [-]$$

- **Calculation of the convection number, Martinelli parameter**

*“It is the ratio of the liquid and gas phases as if each flowed alone in the heat exchanger”* [39]. It is described by the equation (7.21).

$$Co = \left(\frac{1-x}{x}\right)^{0.8} \cdot \sqrt{\frac{\rho_{refv}}{\rho_{refl}}} [-]$$

Where  $\rho_{refv}$  is the refrigerant density at gaseous state, which is 11.856 [kg · m<sup>-3</sup>],  $\rho_{refl}$  is the refrigerant density at liquid state, which is 1245.2 [kg · m<sup>-3</sup>] from the table (7.7), and  $x$  is the quality value, which is 0.513 [-] from the table (7.1).

Therefore,

$$Co = 0.09360 [-]$$

The last parameter is the fluid surface and it depends on the fluid and the heater surface characteristics. *“Due to dealing with a brazed plate heat exchanger from stainless steel material, it can be assumed the value of  $F_s=1$  for all fluid according to Kandlikar correlation”*. [40]

- **Calculation of heat transfer coefficient**

At this point, the approximation of the two-phase heat transfer coefficient can be determined by using the equations (7.16) and (7.17) respectively as follows:

$$\alpha_{TPNB} = 0.6883(0.0936)^{-0.2}(1 - 0.513)^{0.8}(249.12) + 1058(0.001227)^{0.7}(1 - 0.513)^{0.8}(1 \cdot 249.12)$$

$$\alpha_{TPNB} = 1513.37 [W \cdot m^{-2} \cdot K^{-1}]$$

$$\alpha_{TPCB} = 1.136(0.0936)^{-0.9}(1 - 0.513)^{0.8}(249.12) + 667.2(0.001227)^{0.7} \cdot (1 - 0.513)^{0.8}(249.12)$$

$$\alpha_{TPCB} = 2198.42 [W \cdot m^{-2} \cdot K^{-1}]$$

Therefore, the maximum value is used for the calculation of the overall heat transfer coefficient, which is the convective boiling regime value.

### 7.4.3 Correction of overall heat transfer coefficient (U-value)

By using the heat transfer coefficient of each fluid, it can be calculated the overall heat transfer coefficient by the equation (7.22).

$$\frac{1}{U} = \frac{1}{\alpha_{gas}} + \frac{tk}{\lambda} + \frac{1}{\alpha_{ref}} + Rf_{gas} + Rf_{refrigerant} [W \cdot m^{-2} \cdot ^\circ C^{-1}] \quad (7.22)$$

Where  $\lambda$  is the thermal conductivity of the plate, which is  $12.76 [W \cdot m^{-1} \cdot ^\circ C^{-1}]$ , calculated by the software Engineering Equation Solver (EES),  $\alpha_{gas}$  is the heat transfer coefficient of natural gas, which is  $1487.77 [W \cdot m^{-2} \cdot ^\circ C^{-1}]$  from the equation (7.15),  $\alpha_{ref}$  is the heat transfer coefficient of refrigerant, which is  $2198.42 [W \cdot m^{-2} \cdot ^\circ C^{-1}]$  from the equation (7.17),  $Rf_{gas}$  is the fouling factor of the natural gas, which is  $0.000614 [m^2 \cdot ^\circ C \cdot W^{-1}]$ , calculated by the software Engineering Equation Solver (EES), and  $Rf_{refrigerant}$  is the fouling factor of refrigerant, taking into account both phases (liquid and vapor), which is  $0.000528 [m^2 \cdot ^\circ C \cdot W^{-1}]$ , calculated by the software Engineering Equation Solver (EES), and  $tk$  is the thickness of the plate, which is  $0.0005 [m]$  from source (<http://thermopedia.com/content/1035/>)

Therefore,

$$U = 439.96 [W \cdot m^{-2} \cdot ^\circ C^{-1}]$$

Having the U-value, it can be calculated the new area by the equation (7.3) as follows:

$$A = \frac{6,850 [W]}{439.96 [W \cdot m^{-2} \cdot ^\circ C^{-1}] \cdot 11.85 [^\circ C]} = 1.314 [m^2]$$

Then, the number of plates by using the equation (7.7):

$$N_p = \frac{A}{A_{eff}} = \frac{1.408 [m^2]}{0.07652 [m^2]} = 18$$

Therefore, it can be said that the approximation used at the beginning of the calculations of 21 plates is appropriate.

- **Comparison of overall heat transfer coefficient**

Having stated that, it is assumed a (U-value) of  $400 [W \cdot m^{-2} \cdot ^\circ C^{-1}]$ , and compared with both *LMTD* values previously calculated in chapter 7, sub-chapter 7.3, table (7.9) presents the process summary.

**Table 7.9 – Comparison of the area required of the evaporator with LMTD [author]**

LMTD [°C]	U [ $W \cdot m^{-2} \cdot ^\circ C^{-1}$ ]	A [ $m^2$ ]	Np [-]
13.84	400	1.125	15
11.85	400	1.314	18
9.87	400	1.577	21

It can be said that for the three different cases of log-mean temperature difference (LMTD), 21 plates would be suitable to ensure the process correctly. Although, before taking a decision, the final considerations of pressure drop should be taken into account. Therefore, the next step demonstrates this statement.

- **Calculation of pressure drop**

The calculation was described according to the source Chemical Engineering Design [35]

The plate pressure drop can be estimated by using the equation (7.23) as follows:

$$\Delta P_p = 8 \cdot j_f \cdot \left(\frac{L}{D_c}\right) \cdot \frac{\rho \cdot V_{ch}^2}{2} [kPa] \quad (7.23)$$

Where  $\rho$  is the density of natural gas or refrigerant in [ $kg \cdot m^{-3}$ ] from the table (7.6) and (7.7) respectively,  $V_{ch}$  is the channel velocity in [ $m \cdot s^{-1}$ ] from the equation (7.12),  $L$  is the

effective length of flow  $[m]$  from the equation (7.5),  $D_c$  is the equivalent diameter in  $[m]$  from the equation (7.11), and  $j_f$  is a friction factor and can be calculated from the equation (7.24).

$$j_f = 0.6 \cdot Re^{-0.3} [-] \quad (7.24)$$

Where  $Re$  is the Reynolds number from the natural gas or refrigerant in  $[-]$  from the equation (7.13).

Additionally, the pressure drop due to the contraction and expansion losses through the ports in the plates is calculated by the equation (7.25):

$$\Delta P_{pt} = 1.3 \cdot \frac{(\rho \cdot \mu_{pt}^2)}{2} \cdot N_{pass} [kPa] \quad (7.25)$$

The velocity through the ports is calculated by the equation (7.26) as follows:

$$\mu_{pt} = \frac{\dot{m}}{\rho \cdot A_p} [m \cdot s^{-1}] \quad (7.26)$$

The area of the port is given by equation (7.27) as follows:

$$A_p = \frac{\pi \cdot d_{pt}^2}{4} [m^2] \quad (7.27)$$

Where  $\dot{m}$  is the mass flow through the ports of natural gas or refrigerant, which is  $0.2115 [kg \cdot s^{-1}]$  and  $0.0704 [kg \cdot s^{-1}]$  from the table (7.1) respectively,  $d_{pt}$  is the port diameter, which is  $0.035 [m]$  from the table (7.5),  $N_{pass}$  is the number of passes, which is  $1 [-]$ , and  $\rho$  is the density of natural gas or refrigerant, which is  $16.298 [kg \cdot m^{-3}]$  for natural gas, and for refrigerant is described in more detail in the following calculations.

Therefore, the total pressure drop is calculated by equation (7.28) as follows:

$$Total \ pressure \ drop = \Delta P_p + \Delta P_{pt} \quad (7.28)$$

Where  $\Delta P_p$  is the plate pressure drop in  $[kPa]$  from the equation (7.23), and  $\Delta P_{pt}$  is the pressure drop due to the contraction and expansion losses through the ports in the plates in  $[kPa]$  from the equation (7.25).

- **Pressure drop on the natural gas side**

The calculation of  $j_f$  is determined by equation (7.24) as follows:

$$j_f = 0.6 \cdot (16,836.38)^{-0.3}$$

$$j_f = 0.0323 [-]$$

Where  $Re$  is the Reynolds number from the natural gas, which is  $16,836.38 [-]$  from the equation (7.13).

Therefore, the plate pressure drop is calculated by equation (7.23) as follows:

$$\Delta P_p = 8 \cdot (0.0323) \cdot \left( \frac{0.351}{0.004659} \right) \cdot \frac{(16.298) \cdot (2.270)^2}{2}$$

$$\Delta P_p = 0.8174 [kPa]$$

Where  $\rho$  is the density of natural gas, which is  $16.298 [kg \cdot m^{-3}]$ ,  $V_{ch}$  is the channel velocity, which is  $2.270 [m \cdot s^{-1}]$  from the equation (7.12),  $L$  is the effective length of flow, which is  $0.351 [m]$  from the equation (7.5),  $D_c$  is the equivalent diameter, which is  $0.004659 [m]$  from the equation (7.11), and  $j_f$  is  $0.0323 [-]$  from the equation (7.24).

The calculation of the port pressure loss is calculated by the equation (7.25):

$$\Delta P_{pt} = 1.3 \cdot \frac{(16.298) \cdot (13.50^2)}{2} \cdot 1$$

$$\Delta P_{pt} = 1.931 \text{ [kPa]}$$

Where  $\dot{m}$  is the mass flow through the ports of natural gas, which is 0.2115 [ $kg \cdot s^{-1}$ ] from the table (7.1),  $N_{pass}$  is the number of passes, which is 1 [-],  $\rho$  is the density of natural gas, which is 16.298 [ $kg \cdot m^{-3}$ ], and  $\mu_{pt}$  is the velocity through the ports, which is 13.50 [ $m \cdot s^{-1}$ ] from the equation (7.26),  $d_{pt}$  is the port diameter, which is 0.0035 [ $m$ ] from the table (7.5),  $A_p$  is the area of the port, which is 0.000962 [ $m^2$ ] from the equation (7.27).

Therefore, the total pressure drop is calculated by equation (7.28) as follows:

$$\text{Total pressure drop} = 2.7484 \text{ [kPa]}$$

- **Pressure drop on the refrigerant side**

Assuming the refrigerant as a single-phase fluid, the density value is a mean value between the liquid-phase and vapor-phase density. The homogenous density  $\rho_h$  can be calculated by the equation (7.29).

$$\frac{1}{\rho_h} = \frac{x}{\rho_v} + \frac{1-x}{\rho_l} \text{ [kg} \cdot \text{m}^{-3}] \quad (7.29)[38]$$

Where  $\rho_h$  is the homogeneous density in [ $kg \cdot m^{-3}$ ],  $\rho_l$  is the density of the gaseous phase in [ $kg \cdot m^{-3}$ ] from the table (7.7), and  $\rho_v$  is the density of the liquid phase in [ $kg \cdot m^{-3}$ ] from the table (7.7).

Therefore,

$$\rho_h = 22.90 \text{ [kg} \cdot \text{m}^{-3}]$$

This density value is used for calculating the pressure drop.

The calculation of  $j_f$  is determined by equation (7.24) as follows:

$$j_f = 0.6 \cdot (112.67)^{-0.3}$$

$$j_f = 0.1454 \text{ [-]}$$

Where  $Re$  is the Reynolds number from the refrigerant, which is 112.67 [-] from the equation (7.13).

The plate pressure drop is calculated by equation (7.23) as follows:

$$\Delta P_p = 8 \cdot (0.1454) \cdot \left( \frac{0.351}{0.004659} \right) \cdot \frac{(22.90) \cdot (0.5329)^2}{2}$$

$$\Delta P_p = 0.2849 \text{ [kPa]}$$

Where  $\rho$  is the density of refrigerant, which is 22.90 [ $kg \cdot m^{-3}$ ],  $V_{ch}$  is the channel velocity, which is 0.5329 [ $m \cdot s^{-1}$ ] from the equation (7.12),  $L$  is the effective length of flow, which is 0.351 [ $m$ ] from the equation (7.5),  $D_c$  is the equivalent diameter, which is 0.004659 [ $m$ ] from the equation (7.11), and  $j_f$  is 0.1454 [-] from the equation (7.24).

The calculation of the port pressure loss is calculated by the equation (7.24):

$$\Delta P_{pt} = 1.3 \cdot \frac{(22.90) \cdot (3.20^2)}{2} \cdot 1$$

$$\Delta P_{pt} = 0.152 \text{ [kPa]}$$



Where  $\dot{m}$  is the mass flow through the ports of refrigerant, which is  $0.0704 [kg \cdot s^{-1}]$  from the table (7.1),  $N_{pass}$  is the number of passes, which is 1 [-],  $\rho$  is the density of refrigerant, which is  $22.90 [kg \cdot m^{-3}]$  from the equation (7.29),  $\mu_{pt}$  is the velocity through the ports, which is  $3.20 [m \cdot s^{-1}]$  from the equation (7.26),  $d_{pt}$  is the port diameter, which is  $0.035 [m]$  from the table (7.5),  $A_p$  is the area of the port, which is  $0.000962 [m^2]$  from the equation (7.27).

Therefore, the total pressure drop is calculated by equation (7.27) as follows:

$$Total\ pressure\ drop = 0.4366 [kPa]$$

As the last evaluation stage, the calculated pressure drop shows low values, which indeed match the acceptable parameters of the system. For this reason, it was evaluated the table (7.9), to show the advantage of using 23 plates, which ensures the heat transfer across the whole equipment. Therefore, it can be concluded that the Evaporator Model B35TM2 is suitable for the low-temperature drying process, and comply with the requirements. Table (7.10) shows the technical parameters of the evaporator selected for the natural gas process. Additionally, Annex E presents the evaporator datasheet from the company SWEP International AB.

**Table 7.10 – Summary of the evaporator selected [author]**

<b>Evaporator model</b>	B35TM2
<b>Number of plates [-]</b>	21
<b>Number of channels [-]</b>	10
<b>Heating load managed [kW]</b>	6.850

## 8 Design of Condenser

The type of condenser selected for this design is air-cooled condensers. It consists of a copper coil and aluminum fins separated from each other. This condenser have much smaller dimensions, therefore it is very useful in an industrial installation. The design also requires the air circulation to be forced by fans, meaning that it could increase the capacity of the condenser. For the selection of the condenser, the rejection heat value, previously calculated in the refrigeration cycle analysis in chapter 4, is used. It means that the thermal power that the condenser must give to the ambient air is:

$$Q_{cond} = 11.37 \text{ [kW]} \text{ (From equation 4.8)}$$

- **Thermal power exchanged in the condenser**

In order for the condenser to dissipate the heat contained in the refrigerant into the ambient air, it must have the necessary exchange surface. The following equation (8.1) expressed the thermal power exchanged in the condenser:

$$Q_{cond} = U \cdot A \cdot (T_{cond} - T_{mair}) \text{ [kW]} \quad (8.1)$$

Where  $U$  is the overall heat transfer coefficient in  $[\text{W} \cdot \text{m}^{-2} \cdot ^\circ\text{C}^{-1}]$ ,  $A$  is the heat transfer area in  $[\text{m}^2]$ , and the  $T_{mair}$  is the mean temperature of air  $[\text{°C}]$ . The higher the mean temperature of the air circulating through the condenser and the condensation temperature of the refrigerant, the greater the thermal power will be given to the ambient air. However, the thermal differential does not matter that it is large since this will suppose that the condensation temperature is also large and therefore will make the compressor do more work, resulting in a greater expenditure of electrical energy.

The average air temperature can be approximated by the arithmetic mean between the air inlet and outlet temperatures, see equation (8.2).

$$T_{mair} = \frac{T_{ainlet} + T_{aoutlet}}{2} \text{ [°C]} \quad (8.2)$$

Where  $T_{ainlet}$  and  $T_{aoutlet}$  are the temperature of air  $[\text{°C}]$  at the inlet and outlet respectively.

- **Thermal power transferred to the air**

As previously mentioned, the refrigerant transfers heat to the air that passes through it, the thermal power transferred to the air can be expressed by the following equation (8.3).

$$Q_{cond} = C_{p_{air}} \dot{m}_{air} (T_{aoutlet} - T_{ainlet}) \text{ [kW]} \quad (8.3)$$

Where  $C_{p_{air}}$  is the specific heat capacity in  $[\text{kJ} \cdot \text{kg}^{-1} \cdot \text{K}^{-1}]$  and  $\dot{m}_{air}$  is the mass flow of the air in  $[\text{kg} \cdot \text{s}^{-1}]$ . The greater the amount of air circulating through the condenser, the greater the amount of heat extracted by the refrigerant.

### 8.1 Design specification of the condenser heat exchanger

Most air-cooled condensers are designed to condense the refrigerant at a temperature of about  $15^\circ\text{C}$  above ambient temperature. In this case, the refrigeration system operates in low ambient temperature conditions most of the year, therefore certain lower value design considerations have to be taken. It could be said that the temperature of condensation is  $5^\circ\text{C}$

above the maximum temperature of the whole period. That means if the maximum temperature ambient is 37°C, the temperature of condensation is 42°C. Table (8.1) show the parameters that are going to be considered for the design of the condenser.

**Table 8.1 - Design parameters for the condenser heat exchanger [author]**

Parameter	Hot side	Cold side
Fluid circulated	Air	R 404A
Flow rate	3.61 [kg · s <sup>-1</sup> ] (eq 8.4)	0.0704 [kg · s <sup>-1</sup> ] (tab 4.5)
Type of fluid	Single-phase	Two-phase
Inlet temperature	37 [°C]	74.6 [°C]
Condensation temperature	N/A	42 [°C]
Outlet temperature	40 [°C]	41 [°C]
Operating pressure	1.01325 [bar]	19.07 [bar]
Relative Humidity of inlet air	45%	N/A

The flow rate of the air could be calculated rearranging the equation (8.3) into the following equation (8.4):

$$\dot{m}_{air} = \frac{Q_{cond}}{Cp_{air} (T_{aoutlet} - T_{ainlet})} \quad (8.4)$$

Taking the values of the inlet and outlet temperature of the air from table 8.1 and  $Cp_{air} = 1.039$  [kJ · kg<sup>-1</sup> · K<sup>-1</sup>] from EES. The mass flow rate of air result as:

$$\dot{m}_{air} = 3.65 \text{ [kg} \cdot \text{s}^{-1}\text{]}$$

Knowing also the thermal power exchanged in the condenser, it could be done an approximation of the surface need for the transfer. For this, it is going to be needed an overall heat transfer value assumed from the table of the figure (8.1) - Typical overall heat transfer coefficients for air-cooled exchanger.

Air-cooled exchangers	
Process fluid	
Water	300–450
Light organics	300–700
Heavy organics	50–150
Gases, 5–10 bar	50–100
10–30 bar	100–300
Condensing hydrocarbons	300–600

**Figure 8.1 - Typical overall heat transfer coefficients for air-cooled [35]**

Meaning that:

$$U = 200 \text{ [W} \cdot \text{m}^{-2} \cdot \text{°C}^{-1}\text{]}$$

Therefore rearranging equation (8.1), the surface area can be approximate

$$A = \frac{Q_{cond}}{U \cdot (T_{cond} - T_{mair})} \text{ [m}^2\text{]}$$

Where

$$T_{mair} = 38.5 \text{ [°C]} \text{ (from equation 8.2)}$$

Resulting

$$A = 1.61 [m^2]$$

As a first instance an area of 1.61[m<sup>2</sup>] seem a good approximation. It is important to take into account that the ambient temperature considered is the maximum in the whole period, and it occurred only in summer. Most of the time the condenser has to work in low ambient temperature. Meaning that considering the oversize is good, to manage both working conditions. Later it will be important to explain some of the regulations the unit should undertake.

## 8.2 Selection of condenser

Many manufacturers based the selection in the same several steps. But before the selection, they also recommended some coefficients factor that it should be applied to the heat rejection. For this purpose, it is going to be taken the catalog from LENNOX EMEA, which manages the brands FRIGA-BOHN, HK REFRIGERATION, and LENNOX. [41]

First all according to the equation (8.2), the condenser total heat rejection that should be considered is expressed by:

$$Q_{cond} = \frac{Q_{cond}}{C1 \times C2 \times C3 \times C4 \times C5} \quad (8.2)[41]$$

Where

C1 = Altitude Factor

C2 = DT1 Factor

C3 = Ambient Temperature Factor

C4 = Refrigerant Factor

C5 = Fin Material Factor

For the altitude factor, it has to be considered the location, and since most of the wells are located in the area of Hodonin, it is convenient to use a value of Ha=of 164m (source from website: (<https://www.freemaptools.com/>)). Therefore equation (8.3) results as

$$C1 = (1 - 0.000075H) \quad (8.3)$$

$$C1 = 0.987$$

The other coefficients are going to be selected from the following tables.

**Table 8.2 - Factor C2 for DT coefficient [41]**

DT1	8	9	10	11	12	13	14	15	16	17	18
C2	0,53	0,60	0,67	0,73	0,80	0,87	0,93	1	1,07	1,13	1,20

DT means the differential temperature between the inlet temperature of air and the condensation temperature of the refrigerants. Meaning that DT= (42-37) [°C] = 5[°C]. It is considered the value from 8 of the table (8.2). Therefore C2 =0.53

**Table 8.3 - Factor C3 for ambient temperature coefficient [41]**

t <sub>u</sub>	15	20	25	30	35	40	45	50
C3	1,03	1,02	1	0,98	0,96	0,94	0,92	0,91

Knowing the ambient temperature for the design is 37[°C]. From the table (8.3), a value between 35 and 40 could be selected. Hence C3 =0.95

**Table 8.4 - Factor C4 for refrigerant coefficient [41]**

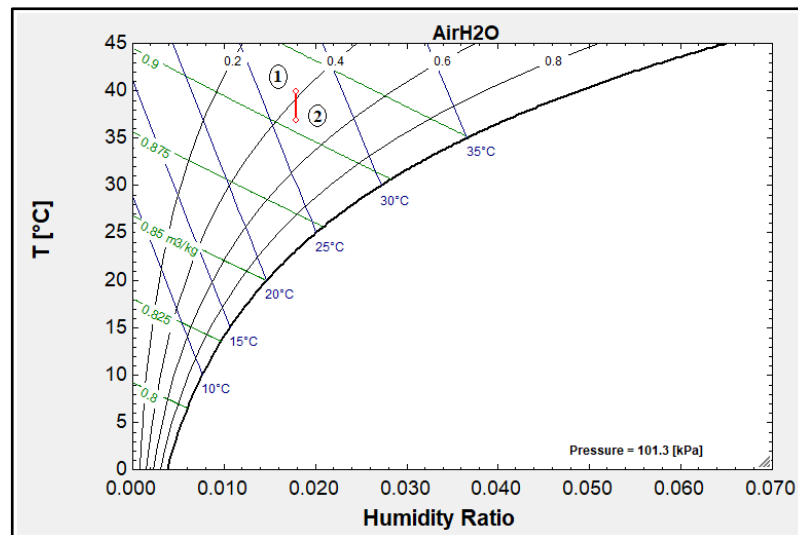
Refrigerant	R404A	R134a	R507A	R407A	R407C	R407F
<b>C4</b>	<b>DT1 = 15K</b>	1	0,96	1	0,89	0,89

The refrigerant that has been considered for the first variation since the beginning is R404A. The factor C4 then has a value of 1.

**Table 8.5 - Factor C5 for fin material correction [41]**

	Aluminium	Coated aluminium
<b>C5</b>	1	0,97

Evaluating the conditions that the material is going to be in contact with, a good decision should be considered the fin as coated aluminum. An approximation could be made evaluating the humidity or moisture carried in the air, giving the conditions that the condenser is going to be dealing most of the time with the template environment. For this purpose, the software EES is going to be used. Figure (8.2) is going to show the psychrometric graph originated by EES.



**Figure 8.2 - Psychrometric chat generated [EES]**

Knowing the inlet relative air humidity  $\phi_1=45\%$  and the inlet temperature  $T_1 =37^\circ\text{C}$ , which therefore by definition of humidity ratio, is not going to change until it gets to the saturation point. Meaning that if the outlet temperature is  $T_2=40^\circ\text{C}$ , then  $\phi_1=38.2\%$ .

Therefore C5 = 0.97.

Once the factors are calculated, it can be applied the equation (8.2)

$$Q_{cond} = \frac{11.37}{0.987 \times 0.53 \times 0.96 \times 1 \times 0.97} = 23.34 \text{ [kW]}$$

This is the value that is going to be used for the selection of the condenser in the catalog. Also, we have to include the flow rate of air calculated previously, but with the conversion in [m<sup>3</sup>.s<sup>-1</sup>]. Knowing the density equal to 1.121[kg.m<sup>3</sup>] (calculated from EES at ambient temperature). Result as:

$$\dot{v}_{air} = 11\,722 \text{ [m}^3 \cdot \text{h}^{-1}\text{]}$$

**Table 8.6 - Comparison of condensers according to the application [41]**





















	Mini Maxi R404A	Centrifugal Axial - 	EC motor	Factory wired	Position and blowing	Coil protection	EUROVENT certification	Commerces proximit� / Restauration	Distribution centres	Industrial process	Air conditioning / "Data processing"	Cogeneration / Generator plants	Food Service	Supermarket	Process / Industry
MA 	3 12	1 > 2 	-						-	-	-	-		-	-
WA 	7,5 99	1 > 6 	-									-		-	-

Table (8.6) shows a comparison between the model MA and WA. It is going to be selected the WA model due to the specification of the power capacity that can handle. Table (8.7) below also shows the filter applies for the selection, in conclusion, the model selected was WA 30.

**Table 8.7 - Selection of the condenser according to the parameter [41]**

		WA ..	15	19	22	30	39	44	48	58	
Capacity	DTI = 15K	04P (Δ)	kW	14,1	18,6	21,1	28,6	37,0	42,5	43,6	55,1
		06P (Y)	kW	12,7	16,2	18,0	25,6	32,3	36,2	38,9	48,4
Surface			m <sup>2</sup>	18	26	35	35	53	70	53	79
Circuit volume			dm <sup>3</sup>	3	4	6	6	9	12	9	13
Fan *	Air flow	04P (Δ)	m <sup>3</sup> /h	7500	6940	6450	15010	13870	12910	22520	20810
		06P (Y)	m <sup>3</sup> /h	6050	5510	5070	12100	11020	10130	18140	16540
			Nb x mm	1x Ø 500	1x Ø 500	1x Ø 500	2x Ø 500	2x Ø 500	2x Ø 500	3x Ø 500	3x Ø 500
Energy efficiency class		04P (Δ)		E	E	E	E	E	E	E	
		06P (Y)		E	E	D	E	E	D	E	
Acoustic	Lw (1)	04P (Δ)	dB(A)	74	73	73	77	76	76	79	78
		06P (Y)	dB(A)	69	68	68	72	71	71	74	72
	Lp (2)	04P (Δ)	dB(A)	43	42	42	46	45	45	47	46
		06P (Y)	dB(A)	38	37	36	41	40	39	42	41
Net weight			kg	36	40	44	63	72	80	92	104
Circuits			Nb	2	4	4	4	6	8	8	8
Dimensions	A		mm	730	730	730	1390	1390	1390	2050	2050
	B		mm	240	520	340	340	495	390	390	470
	C		mm	150	150	150	150	155	155	155	155
	D		mm	20	25	25	25	30	30	30	30
	E		mm	55	40	55	55	45	55	55	45
	F		mm	73	53	34	73	53	34	73	53
	G		mm	78	81	81	81	88	88	92	88
	X		mm	680	680	680	1340	1340	1340	2000	2000
Inlet		ODF (4)	1/2"	5/8"	5/8"	3/4"	7/8"	7/8"	7/8"	1"1/8"	
Outlet		ODF (4)	1/2"	5/8"	5/8"	5/8"	7/8"	7/8"	7/8"	7/8"	

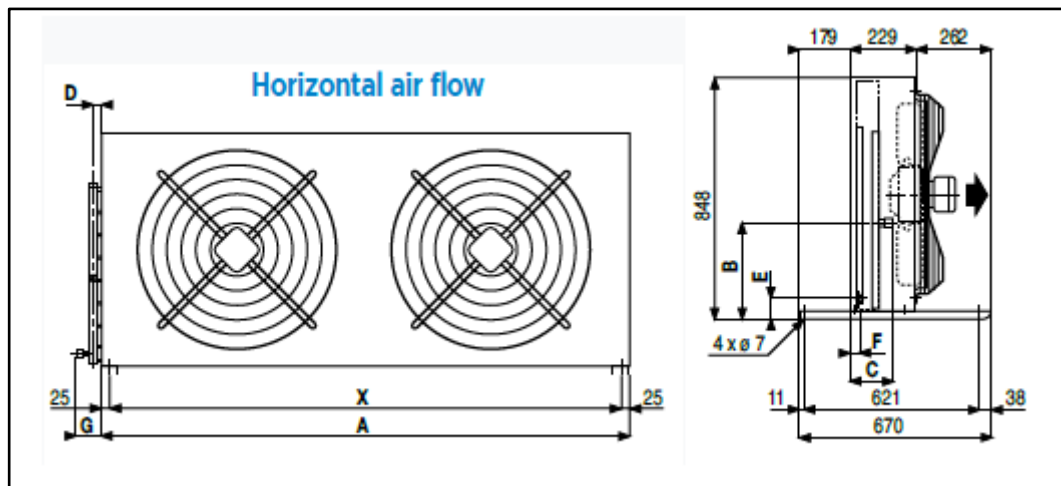


Figure 8.3 - Dimension of the Air-cooled Condenser selected W30 [41]

Figure (8.2), shows the dimension that corresponds to the selected model. In fact, we can observe by calculation that area value is very close to the one calculated in subchapter (8.1), meaning that it meets the requirement.  $X=1.340\text{m}$  and the high is  $0.848$ , therefore the area is close to  $1.13 \text{ m}^2$ .

To summarize the specific parameters of the condenser that was selected the table (8.8) is shown. It is important to notice that in maximum condition of ambient temperature as  $37^\circ\text{C}$  the maximum capacity is  $28.6 \text{ [kW]}$ , otherwise during template conditions the capacity needed should be reduced

Table 8.8 - Parameters of the condenser selected [author]

<b>Product name</b>	WA-Axial Fan Condenser
<b>Designation</b>	WA 30 04/06P
<b>Capacity at maximum</b>	$28.6 \text{ [kW]}$
<b>Refrigerants</b>	R404A
<b>Fan Airflow</b>	$15010 \text{ [m}^3/\text{h]}$
<b>Input Power</b>	$710 \text{ [W]}$ max
<b>Fan Diameter</b>	$500 \text{ [mm]}$
<b>Number of Fan</b>	2

### 8.3 Regulation and control of the condenser

After the selection of the condenser, there are several important factors it should be considered for the correct operation and this why is the question arises what happened when the ambient temperature gets low?

This question can be analyzed, by observing equation (8.3) and (8.2).

Meaning that in  $Q_{cond} = C p_{air} \dot{m}_{air} (T_{a \text{ outlet}} - T_{a \text{ inlet}})$ .  $T_{a \text{ outlet}}$  and  $T_{a \text{ inlet}}$  decreases. Therefore  $T_{cond}$  in equation (8.1) must decrease also. Resulting in a decrease in the pressure of the system. It has to be remembered that the standard expansion valve requires a minimum pressure drop between the liquid in the zone of high pressure (pressure of condensation) that enters the valve, and the fluid in the low-pressure zone after the exit. Hence, if the pressure of

the condensation gets low, the valve will not function correctly [42]. The solution is to maintain a discharge pressure above the level. For this, I will recommend some methods of regulation that can be applied, however, it is not going to be explained a process of selection of these equipment involved in the methods.

- **Installation of a Liquid Recipient**

The liquid receiver stores the excess refrigerant throughout the system. It handles excess refrigerant charge that occurs with air-cooled condensers, this due to fluctuating charges in the low side so that the liquid condenser is drained to maintain an effective condensing surface, mainly in systems where the charge of the operation of condenser varies for different loading conditions.[42]

- **Fan cycle controls**

Control load pressure by monitoring:-compressor discharge pressure - Ambient temperature.

This control is performed by stopping the condenser fan when the load pressure decreases to a minimum disconnect pressure value. It means that the fan will not push cold air through it, thus increasing the surface temperature by increasing the load pressure. [42]

- **Load pressure regulating valve**

This is about loading the unit with a special valve, causing strangulation between the condenser and the liquid vessel, which makes it difficult to evacuate the condensed liquid to the vessel. As I mentioned earlier this regulator tends to store the fluid in the condenser. Retaining the liquid in the condenser increases the cooling zone and as a result, decreases the area used by the condenser to convert the steam coming from the discharge pipe into liquid. See figure (8.4).

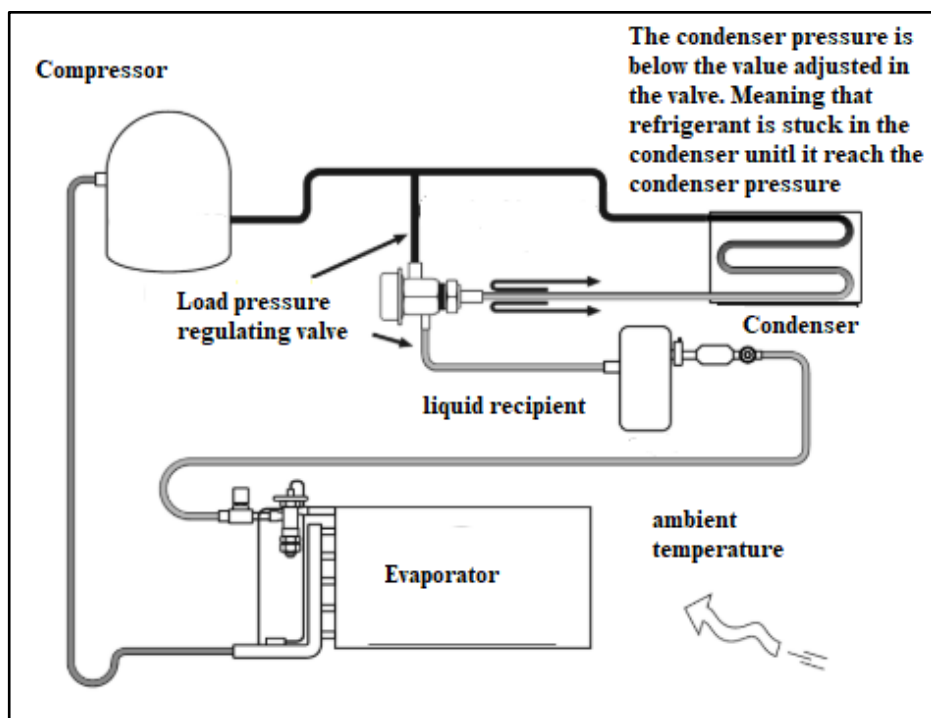


Figure 8.4 - Regulation by controlling the load pressure on the condenser [author]



## 9 Analysis of the refrigeration cycle for refrigerant R449A

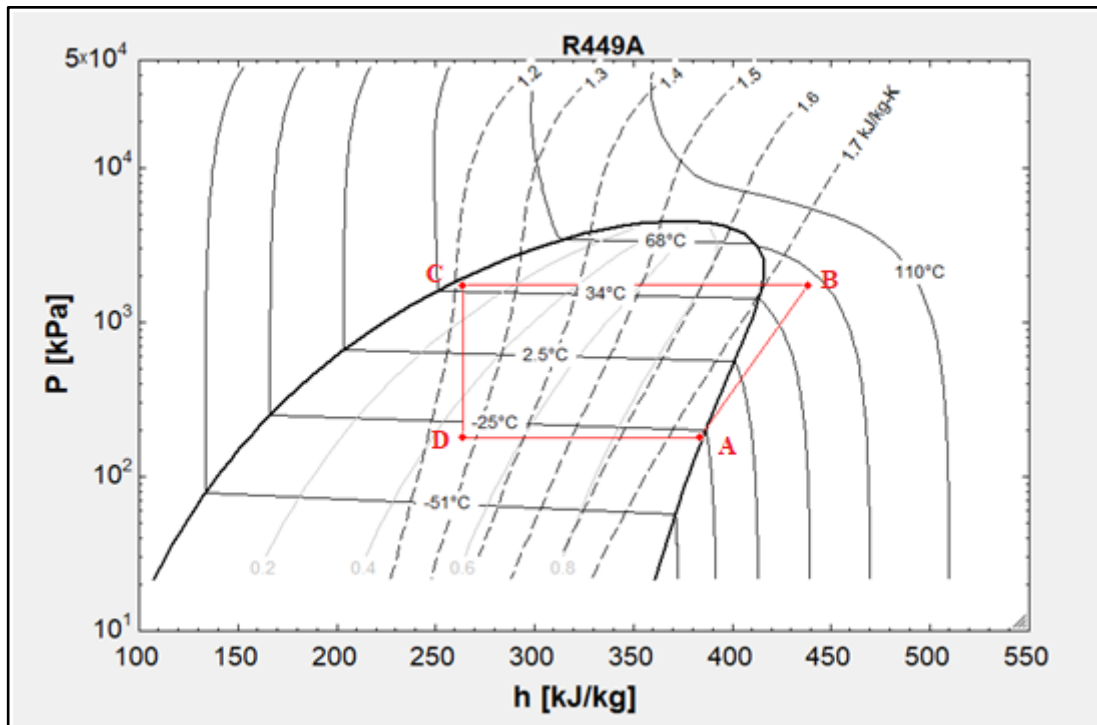
Basically, for the refrigerant R449 is going to be applied the same process analysis that was used for R404A previously. It is going to be considered also the same assumption and the same conditions, firstly for the ideal cycle.

- Evaporation Temperature =  $-27[^\circ\text{C}]$ .
- Condensation Temperature =  $42[^\circ\text{C}]$ .
- Maximum cooling load required =  $6.850 \text{ [kW]}$ .
- Minimum cooling load required =  $2.739 \text{ [kW]}$ .

Tables (9.1) and (9.2) show the parameters involved in each stage of the cycle and also the important variables of operation for the design of the cooling system. As well, figure 9.1 shows the enthalpy pressure diagram for refrigerant R449. See Annex F for R449A properties.

**Table 9.1 - R449A refrigerant properties for the ideal cycle [author]**

Stage	(P) Pressure absolute [kPa]	(T)Temperature $[^\circ\text{C}]$	(h) Spec. Enthalpy [kJ/kg]	(s) Spec. Entropy [kJ/kg.K]
A	183.15	-27	384	1.768
B	1745.93 (table)	60.85 (by EES)	438.6(by EES)	1.768
C	1745.93	42	264	1.213
D	183.15	-27	264	1.273 (by EES)

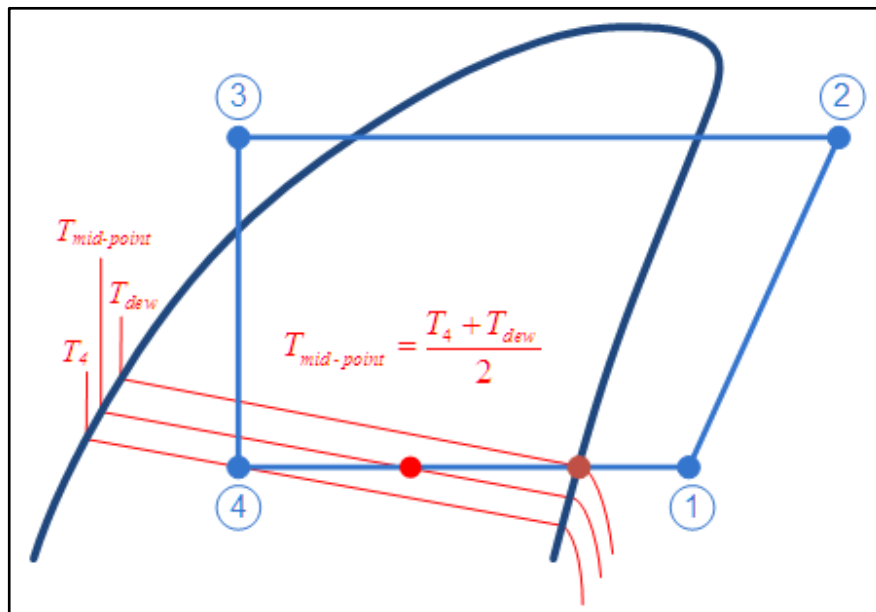


**Figure 9.1 - Pressure-enthalpy diagram R449A for the ideal cycle [EES]**

**Table 9.2 - Analysis of operating variables for the ideal cycle of R449A [author]**

Parameters	Q <sub>max</sub>	Q <sub>min</sub>
Cooling effect (eq 4.3)	120[kJ. kg <sup>-1</sup> ]	120[kJ. kg <sup>-1</sup> ]
Cooling Capacity of Evaporator (eq 4.1)	6.850 [kW]	2.739 [kW]
Mass flow of refrigerant (eq 4.4)	0.05708[kg. s <sup>-1</sup> ]	0.02283[kg. s <sup>-1</sup> ]
Compressor Work (eq 4.5)	54.6 [kJ. kg <sup>-1</sup> ]	54.6[kJ. kg <sup>-1</sup> ]
Theoretical power required by the compressor (eq 4.6)	3.12 [kW]	1.25[kW]
Displacement of the compressor or Volume flow rate (eq 4.7)	24.66[m <sup>3</sup> . h <sup>-1</sup> ]	9.84[m <sup>3</sup> . h <sup>-1</sup> ]
Heat Rejection in the Condenser (eq 4.8)	9.97 [kW]	3.99 [kW]
Coefficient of Performance (eq 4.9)	2.20 [-]	2.20 [-]
Quality (eq 4.10)	53.48 %	53.48%
Pressure Ratio (eq 4.11)	9.53 [-]	9.53 [-]

Refrigerants like R449A, present glide characteristics as it was mention before. Therefore the condensation temperature has to be treat as the mid-point and not the dew point. Having said that, for the real cycle analysis, is going to be considered figure (9.2), for the calculation of the mid-point, the figure was provided by Danfoss Software CoolSelector.



**Figure 9.2 - Midpoint vs dew point in the refrigeration cycle [CS2]**

Point	Description	Temperature [°C]	Pressure [bar]	Density [kg/m <sup>3</sup> ]	Enthalpy [kJ/kg]	Entropy [kJ/(kg·K)]
1	Compressor suction	-14.3	1.957	8.414	393.4	1.8
2	Compressor discharge (estimated)	115.2	18.42	56.52	495.9	1.923
2s	Condensation dew point	44.3	18.42	86.45	412.8	1.686
3s	Condensation bubble point	39.7	18.42	1021	259.8	1.198
3a	Condenser out	38.7	18.42	1027	258.2	1.193
3	Including additional subcooling	38.7	18.42	1027	258.2	1.193
4	After expansion valve	-28.6	1.957	19.85	258.2	1.25
4s	Evaporation bubble point	-31.3	1.957	1301	157.6	0.8408
1s	Evaporation dew point	-25.4	1.957	8.887	384.3	1.764
1a	Evaporator out	-15.3	1.957	8.454	392.6	1.796

**Figure 9.3 - R449A refrigerant properties for the real cycle [CS2]**

Figure 9.3 shows that between the points located after the expansion valve and the evaporation dew point, is the midpoint of -27°C. This also represents the parameters for the real cycle. For the analysis was used the same subcooling and superheating conditions as was shown in table (4.4) Then with figure (9.3) and table (9.3), could be made the right selection of equipment for these cycles.

**Table 9.3 - Analysis of operating variables for the real cycle [author]**

Parameters	Qmax
Cooling effect (eq 4.3)	135.2[kj. kg <sup>-1</sup> ]
Cooling Capacity of Evaporator (eq 4.1)	6.850[kW]
Mass flow of refrigerant (eq 4.4)	0.05067 [kg. s <sup>-1</sup> ]
Compressor Work (eq 4.5)	102.5 [kj. kg <sup>-1</sup> ]
Technical power required by the compressor (eq 4.6)	5.19 [kW]
Displacement of the compressor or Volume flow rate (eq 4.7)	21.66[m <sup>3</sup> . h <sup>-1</sup> ]
Heat Rejection in the Condenser (eq 4.8)	12.04 [kW]
Coefficient of Performance (eq 4.9)	1.32 [-]
Quality (eq 4.10)	50 %
Pressure Ratio (eq 4.11)	9.41 [-]

## 9.1 Selection of Equipment

The selection of equipment is going to be based on the same process and equation used in previous chapters for the R404A.

### 9.1.1 Compressor

With the parameters calculated previously in chapter 9 and using the Copeland catalog, it has been chosen the compressor 3DAD-75X as the best option. This compressor is a reciprocating type with a feature that can regulate the load power depending on the pressure

suction. Been more convenient in this case, than the scroll one. In the Annex G, it could be seen widely its parameters.



Figure 9.4 - Compressor 3DAD-75X [Annex G]

Table 9.4 - Compressor 3DAD-75X Operating parameters [Annex G]

Performance at Specified Point	-25.4°C/44.3°C
Compressor Model	3DAD-75X
Evaporator Capacity	7.83 [kW]
Power	5.09 [kW]
COP	1.54
Current at 400V	10.84 [A]
Suction Mass Flow	0.05830[kg/s]
Isentropic Eff	65.15%
Refrigerant	R449A

For the temperature range control, it has to be used a thermostat as well. For more details, look for subchapter 5.4.

### 9.1.2 Expansion Valve



Figure 9.5 - T2-5 Expansion valve [CS2]

Table 9.5 - Parameters of T2-5 Expansion Valve [CS2]

Product name	Thermostatic expansion valve T2-5
Code	068Z3727
Type	T2/TE2
Refrigerants	R449A
Inlet size	3/8 [in] = 9.525 [mm]
Inlet connection type	Solder,ODF
Outlet size	1/2 [in] =12.7[mm]
Outlet connection type	SAE Flare
Orifice capacities	5[TR]

The expansion valve was selected using the catalog of Danfoss. Also, one of the most important parameters it has to be checked before selected is the range of capacities which should it work. In this case figure (9.7), it shows us that it operates between a nominal capacity of 7.411 [kW] and a minimum capacity of 1.853 [kW]. Then figure 9.8 shows the behavior of the parameter at the inlet and outlet. It can be seen the difference of pressure that should be taken

into consideration for the regulation of condenser and the quality which should be a parameter for the next selection of evaporator. See Annex H

Value	Unit	Inlet	Outlet	Difference
Pressure	bar	18.42	2.757	-15.66
Temperature	°C	38.7	-20.4	-59.1
Bubble point temperature	°C	39.7	-22.8	-62.5
Dew point temperature	°C	44.3	-16.9	-61.2
Density	kg/m <sup>3</sup>	1027	29.94	-996.8
Enthalpy	kJ/kg	258.2	258.2	0
Quality	-	0.00	0.41	0.41
Velocity	m/s	0.99	21.68	20.69
Mass flow	kg/h	183.5	183.5	0

**Figure 9.6 - Performance detail of T2-5 [CS2]**

Type	T2 - 3	T2 - 4	T2 - 5	T2 - 6
NS	10	10	10	10
Range	N	N	N	N
Nominal capacity [kW]	3.849	5.662	7.411	8.813
Min. capacity [kW]	0.962	1.416	1.853	2.203
Load [%]	178	121	92	78
DP [bar]	15.66	15.66	15.66	15.66
Velocity, in [m/s]	0.99	0.99	0.99	0.99

**Figure 9.7 - Nominal and minimum capacity of the thermostatic expansion valve T2-5 [CS2]**

### 9.1.3 Evaporator

Chapter 7 explain all the process and equations of the method used for the design of the evaporator. Having said that, it is going to rely also on SWEP manufacturer catalog. Table (9.6) show the parameters used for the design of the evaporator.

Due to the complexity of the process for the selection of the evaporator, it is going to be summarized the important parameters used for the calculations of the variables in the following tables. The approximations were made by Excel.

**Table 9.6 - Design parameters for the evaporator heat exchanger R449A [author]**

Parameter	Hot side	Cold side
Fluid circulated	Natural gas	R 449A
Flow rate	25,000 [Nm <sup>3</sup> · day <sup>-1</sup> ]	0.0507 [kg · s <sup>-1</sup> ]
Type of fluid	Single-phase	Two-phase
Inlet temperature	-5 [°C]	-28.6 [°C]
Superheating	N/A	10 [°C]
Quality	N/A	0.41 [-]
Outlet temperature	-19 [°C]	-15.3 [°C]
Operating pressure	20 [bar]	1.957 [bar]
Maximum pressure drop	0.01 [bar]	0.8 [bar]
Strength dimension of the evaporator	30 [bar]	30 [bar]

Having stated the initial parameters, it is also selected the same evaporator as chapter 7, shown on figure (7.3), figure (7.4), table (7.4) and table (7.5). Table (9.7) show the initial values and table (9.8) show the characteristic of the plate which is going to be crucial for the design of the evaporator.

**Table 9.7 - Initial values[author]**

<b>Q evap</b>	6850 [W]
<b>U estimation</b>	350 [ $W \cdot m^{-2} \cdot ^\circ C^{-1}$ ]
<b>LTMD</b>	11.85 [ $^\circ C$ ]
<b>Area</b>	1.652 [m <sup>2</sup> ]

**Table 9.8 - Characteristic of the Plates[Annex E]**

<b>Effective length of flow</b>	0.35100321 [m] (eq 7.5)
<b>Effective plate area</b>	0.0765187 [m <sup>2</sup> ] (eq 7.4)
<b>Number plates</b>	25 (eq 7.7)
<b>Number of channel per pass</b>	12 (eq 7.8)
<b>Spacing of plate</b>	0.00235237 [m] (eq 7.10)
<b>Cross-sectional area</b>	0.00057162[m] (eq 7.9)
<b>Equivalent diameter</b>	0.00465962 [m] (eq 7.11)

For the calculation of the overall heat transfer value was used the table (7.6), describing the parameters of the natural gas, and it is summarized in table (9.9). While table (9.10) shows the calculation of the heat transfer of natural gas needed for the overall heat transfer value.

**Table 9.9 - Properties of Natural gas for Evaporator [EES]**

<b>Physical properties@260K</b>	<b>Natural gas</b>
Natural gas pressure, P	2 000 [kPa]
Density, $\rho$	16.298 [ $kg \cdot m^{-3}$ ]
Mass flow, $\dot{m}_{gas}$	0.2115 [ $kg \cdot s^{-1}$ ]
Viscosity, $\nu$	6.283E-07 [ $m^2 \cdot s^{-1}$ ]
Prandalt, $P_r$	0.7744 [-]
Conductivity, $k$	0.03066 [ $W \cdot m^{-1} \cdot C^{-1}$ ]
Wall viscosity dyn @264K	0.00001003 [ $kg \cdot m^{-1} \cdot s^{-1}$ ]
bulk viscosity dyn	0.00001024 [ $kg \cdot m^{-1} \cdot s^{-1}$ ]

**Table 9.10 - Natural gas heat transfer value [author]**

<b>Channel Velocity</b>	2.16 [ $m \cdot s^{-1}$ ] (eq 7.12)
<b>Reynolds Number</b>	16034.65 [-] (eq 7.13)
<b>Nusselt Number</b>	222.11 [-] (eq 7.14)
<b>Heat transfer coefficient</b>	1461.44 [ $W \cdot m^{-2} \cdot ^\circ C^{-1}$ ] (eq 7.15)

The same applies to the refrigeration R449A. Table (9.11) shows the properties of the refrigerant and Table (9.12) the parameters for the calculation of heat transfer value.

**Table 9.11 - Properties of refrigerant R449A[EES]**

Physical properties@246.15K	R449A liquid phase	R449A gas phase
Pressure, $P$	195.7[kPa]	195.7[kPa]
Density, $\rho$	1290 [kg · m <sup>-3</sup> ]	8.347 [kg · m <sup>-3</sup> ]
Mass flow rate	0.029913 [kg · s <sup>-1</sup> ]	0.020787[kg · s <sup>-1</sup> ]
Kinematic Viscosity, $\nu$	0.000000208 [m <sup>2</sup> · s <sup>-1</sup> ]	-
Bulk viscosity dyn, $\mu_{ref}$	0.0002685 [kg · m <sup>-1</sup> · s <sup>-1</sup> ]	-
Wall viscosity dyn, $\mu_{refw}$	0.0002457 [kg · m <sup>-1</sup> · s <sup>-1</sup> ]	-
Prandalt number.	3.299 [-]	-
Conductivity, $k$	0.1052 [W · m <sup>-1</sup> · C <sup>-1</sup> ]	-

**Table 9.12 - Parameters for the calculation of heat transfer value. Refrigerant R449A[author]**

<b>Channel Velocity</b>	0.00386 [ $m \cdot s^{-1}$ ] (eq 7.12)
<b>Reynolds Number</b>	86.55 [-] (eq 7.13)
<b>Nusselt Number</b>	11.06[-] (eq 7.14)
<b>Heat transfer coefficient</b>	249.73 [ $W \cdot m^{-2} \cdot ^\circ C^{-1}$ ] (eq 7.19)
<b>boiling number</b>	0.0020633[-] (eq 7.20)
<b>Martinelli parameter</b>	0.1076279 [-] (eq 7.21)
<b>Heat transfer coefficient two phases</b>	2823.7113 (eq 7.18)

Once all the parameters all calculated, the U-value is approximated, confirming also the number of plates selected. After some corrections, and using also the considerations of the table (7.9) of different cases of LTMD, the evaporator selected is described in the following table.

**Table 9.13 - Summary of the evaporator selected for Refrigerant R449A [author]**

<b>Evaporator model</b>	B35TM2
<b>Number of plates</b>	21
<b>Number of channels</b>	10
<b>Heating load managed</b>	6.850 [kW]

### 9.1.1 Condenser

Table (9.14) shows the parameter used for the selection of the condenser for refrigerant R449A. It was based on the same alignment presented in the process as was described in chapter 6. Despite many European brands share the same process selection, consideration is taken according to each manufacturer.

**Table 9.14 - Design parameters for the condenser heat exchanger [author]**

Parameter	Hot side	Cold side
Fluid circulated	air	R 449A
Flow rate	3.61 [kg · s <sup>-1</sup> ] (eq 8.4)	0.0507 [kg · s <sup>-1</sup> ]
Type of fluid	Single-phase	Two-phase
Inlet temperature	37 [°C]	115.2 [°C]
Condensation temperature	N/A	42 [°C]
Outlet temperature	40 [°C]	38.7 [°C]
Operating pressure	1,01325 [bar]	18.42 [bar]
Relative Humidity of inlet air	45%	N/A

For this selection was used the Larkin catalog, and it can be reviewed on the Annex I. Table (9.15) summarize the values used for the selection. Figure (9.8) and (9.9) show the model selected and its characteristics.

**Table 9.15 - Factors for condenser selection. R449A [author]**

<b>Total heat rejected, THR</b>	12.04[kW] = 10352[kcal/h]
<b>Altitude Factor, AF</b>	1.01
<b>Design condenser, TD</b>	5[°C]

$$THR = \frac{THR \times AF}{TD} = 2091.04 \text{ [kcal.c}^\circ\text{.h}^{-1}\text{]}$$

FCB Model	Capacity (kcal / hr / 1°C)				
	R-404A / R-507	R-407C *	R-407A / R-407F	R-448A *	R-449A *
1	310	291	304	298	307
1.5	360	338	353	346	356
2	420	395	412	403	416
3	640	602	627	614	634
5	990	931	970	950	980
8	1,640	1542	1607	1574	1624
10	1,990	1871	1950	1910	1970
<b>12</b>	2,490	2341	2440	2390	<b>2465</b>
14	2,860	2688	2803	2746	2831
16	3,270	3074	3205	3139	3237
21	4,280	4023	4194	4109	4237
23	4,630	4352	4537	4445	4584
26	5,430	5104	5321	5213	5376

**Figure 9.8 - Selection of Condenser FCB model 12 [Annex I]**

FCB Models	CFM/m³h	Fan		Motor Data					
		No.	Dia. In./mm	HP <sup>1</sup>	FLA <sup>1</sup>	HP <sup>2</sup>	FLA <sup>2</sup>	HP <sup>3</sup>	FLA <sup>3</sup>
12	12400 21100	2	26 660	1/2	7.8	1/3	5.2/2.6	1/3	1.9

**Figure 9.9 - Condenser FCB model 12 characteristic [Annex I]**



## 10 Summary of output values for the cooling system design

Table (10.1) shows the output values calculated along with all the documents that serve as the important operational parameters of the cooling system.

**Table 10.1 -Output values for the cooling system [author]**

			<b>R404A</b>	<b>R404A</b>	<b>R449A</b>	<b>R449A</b>
<b>Input amount of natural gas</b>	$V_{IN1}$	[Nm <sup>3</sup> /day]	<b>25000</b>	<b>10000</b>	<b>25000</b>	<b>10000</b>
<b>Mass flow of refrigerant</b>	$m_R$	[kg/s]	0.0704	0.0277	0.0507	0.0203
<b>Coolant pressure before compressor</b>	$p_{V1}$	[MPa]	0.231	0.231	0.196	0.196
<b>Refrigerant temperature in front of the compressor</b>	$t_{V1}$	[°C]	-16	-16	-14.3	-14.3
<b>Cooling capacity collected in the exchanger VT2</b>	$Q_{t2}$	[kW]	98	98	135.2	135.2
<b>Compressor power consumption</b>	$P_{el}$	[kW]	6.52	2.22	5.09	1.92
<b>Specific power consumption of the compressor on the 1Nm<sup>3</sup> of the gas</b>	$A_{el}$	[kWh/Nm <sup>3</sup> ]	0.25	0.25	0.20	0.20
<b>Coolant pressure after compressor</b>	$p_{V2}$	[MPa]	1.907	1.907	1.842	1.842
<b>Refrigerant temperature in front of the compressor</b>	$t_{V2}$	[°C]	74.6	74.6	115.2	115.2
<b>Refrigerant pressure before condenser</b>	$p_{K1}$	[MPa]	1.907	1.907	1.842	1.842
<b>Refrigerant temperature in front of condenser</b>	$t_{K1}$	[°C]	74.6	74.6	115.2	115.2
<b>Refrigerant pressure after condenser</b>	$p_{K2}$	[MPa]	1.907	1.907	1.842	1.842
<b>Condenser Coolant temperature</b>	$t_{K2}$	[°C]	40.6	40.6	38.7	38.7
<b>Refrigerant pressure before evaporator</b>	$p_{S1}$	[MPa]	0.231	0.231	0.196	0.196
<b>Refrigerant temperature before evaporator</b>	$t_{S1}$	[°C]	-27	-27	-28.6	-28.6
<b>Refrigerant pressure at the outlet of the evaporator</b>	$p_{S2}$	[MPa]	0.231	0.231	0.196	0.196
<b>Evaporator outlet refrigerant temperature</b>	$t_{S2}$	[°C]	-16	-16	-14.3	-14.3

Table 10.1 continues in this section

			R404A	R404A	R449A	R449A
<b>Input amount of natural gas</b>	$V_{IN1}$	[Nm <sup>3</sup> /day]	<b>25000</b>	<b>10000</b>	<b>25000</b>	<b>10000</b>
<b>Mass flow of cooling air</b>	$m_{VZD}$	[kg/s]	3.64	1.43	3.86	1.55
<b>Atmospheric cooling air pressure</b>	$p_a$	[kPa]	101.325	101.325	101.325	101.325
<b>Cooling air temperature Outdoor = before condenser</b>	$t_a$	[°C]	37	37	37	37
<b>Relative air humidity in front of condenser</b>	$j_{VZD1}$	[%]	45	45	45	45
<b>Relative air humidity after condenser</b>	$j_{VZD2}$	[%]	38.2	38.2	38.2	38.2
<b>Cooling air temperature after condenser</b>	$t_{VZD2}$	[°C]	40	40	40	40
<b>Condenser Cooling Capacity</b>	$Q_k$	[kW]	11.37	4.47	12.04	4.82
<b>Electric power consumption of the fan</b>	$P_{el\ v}$	[kW]	0.480	0.189	0.373	0.149

## 10.1 Analysis and recommendation for the design

There are important variables when we talk about linking design with sustainability. These factors that should be considered are cost, energy efficiency, and environmental influence.

The energy efficiency of a system depends though on various factors such as the heat transfer and the components involved in it, the technology applied to the compression mechanism. It means that the technology characterized by the compression plays an important role in cost and efficiency, including the mechanical, electrical and electronic architecture of the system.

Compressor scroll has been proofed to be very efficient and to have some advantage against the reciprocant type, according to many brands such as Copeland. Nevertheless, during the selection of compressor for both uses, R404A and R449A, I presented in the first one a scroll type and the second on a reciprocating type, see table (10.2).

**Table 10.2 - Comparison of the compressor [author]**

<b>Compressor properties</b>	<b>R449A Compressor 3DAD-75X</b>	<b>R404A Compressor ZF33K4E-TWD</b>
Evaporator Capacity	7.83 [kW]	8.67 [kW]
Power	5.09 [kW]	6.59 [kW]
Suction Mass Flow	0.05830[kg/s]	0.0884 [kg/s]
COP	1.54	1.32

If we compare both compressors selected, you can see the compressor *3DAD-75X* present a feature called digital discuss, which is in charge of the continuous capacity modulation, and it is reached by a valve that actuates as a piston controlling the flow of gas at the suction area. On the other hand, the scroll compressor *ZF33K4E-TWD* range for low and medium

temperature doesn't show this digital discuss feature, despite that the scroll compressor is more characterized to have it. Due to we are not dealing with the high cooling load required, the selection of an accurate option of compressor might be limited. However, the compressor *3DAD-75X* is suitable also for refrigerant R404A.

- Efficiency and affordability

Despite that for this case, the efficiency of the scroll could seem an advantage, it has to be taken into account the power and cooling capability of this compressor comes in a high tag. Therefore sometimes it is recommended to set and adjust depending on the capacity of the unit you want to manage. For this case as was said before, the cooling load required is not that big, hence considering the reciprocating compressor for this rate of volume managed by the low-temperature separation unit, might be the most adequate.

- Environmental awareness

As was mentioned previously, in chapter 2.4 refrigerant, nowadays the use of refrigerants is more regulated than ever. Therefore the transition of refrigerant r404a to other alternatives was marked this year as an important commitment in the refrigeration industry. And R449A was shown to closely match R404A performance, within the calculation. Meaning that it is suitable for both retrofit and new systems. Definitely, the relative mass flow was seen to be decreased as well with the design using R449A, meaning less cost but also in case of retrofitting of a system, it should be taken into account the piping of it.

## 11 Conclusion

The main focus of the thesis was to present a base of calculation that can be applied for the selection of equipment for a refrigeration system, involved in the treatment of natural gas, and therefore always seeking to meet the requirements presented by the company. For this goal, it was an important background in natural gas properties and principles of the refrigeration cycle, which is presented in the theoretical part. Then the methodology of the selection of the equipment was made first by the analysis of the ideal cycle. With these terms clear, the selection of equipment was going to be made by considering a range of operating conditions in which the cycle works.

The comparison of the refrigerant r404A and r449A play a key role in this thesis, due that it marks the transition of the r404A to an alternative refrigerant that has been prof that closely matches in its performance.

Every equipment that forms part of the refrigeration cycle has a different method of design and selection. And it had been shown also that for this type of process, i.e treatment of gas, it should be considered, to take advantage of the design, in making it not only energetic efficiency and environmentally friendly, but efficiently in operation and reliable. Then the results could be translated into a low cost, and therefore high returns.

Also is imperative to add, that it is totally important to design a system of control and regulation of the refrigeration system for this type of process, in which the flow rate of the gas could be varied throughout the year. What it should be looking for, is the proper range of operation, in which the compressor performs efficiently most of the time. That's the reason why the expansion valve plays a key role.

The design uses many initial criteria that depend on the requirement of the company and others by experience. As it is mentioned in the Heat Exchanger Design Book by T.Kuppan, it is unlikely that any two designers will arrive at exactly the same design for a given set of conditions, as the design process involves many judgments while carrying out the design.

Definitely, nowadays these calculations are made by software assistants, and depending on each brand, they set up features that could help the adequate decision for certain types of conditions.

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