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# *Theoretical and experimental contributions on the substitution of refrigerants in the air handling unit*

## *- SUMMARY -*

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# CONTENTS

# INTRODUCTION

Global warming and the depletion of the ozone layer are two global environmental problems at the moment. These two concerns are also related to the use of refrigerants in refrigeration and air conditioning installations, although some of the refrigerants have been banned over the years and other more environmentally friendly refrigerants have been proposed, the refrigerant that does not harm the environment is still being sought [1].

## 1. THE OBJECTIVE OF THE DOCTORAL THESIS

The main objective of this work is to replace the refrigerants used at the moment, in an air handling unit that is equipped with a refrigeration plant, with the aim of having the same refrigeration power of the plant, with a similar energy consumption or smaller, with as little impact on the environment as possible and with as few changes to the installation as possible.

## 2. STRUCTURE OF THE DOCTORAL THESIS

The doctoral thesis is composed of 7 chapters, starting from the bibliographic study of the specialized literature, regarding the current state of refrigerants in the field of air conditioning with air handling units. In chapter 1, the main refrigerants that are repeatedly used in air conditioning installations will be identified. 28 refrigerants were chosen, potential replacements for the known refrigerants, resulting in 3 groups, respectively the R134a refrigerant group, the R407C refrigerant group and the R410A refrigerant group, with the role of complying with the European Union's environmental regulations, maintaining the capacity refrigeration and to make as few changes as possible to the existing installation.

In chapter 2, an energy analysis will be performed at the theoretical thermodynamic cycle level in different operating regimes from the perspective of vaporization temperature for a refrigeration plant for all 3 groups of refrigerants. The energy analysis of 28 new refrigerants will be studied, comparing the energy performance of each group of refrigerants and identifying the best performing refrigerant in each group. This energy analysis is carried out for an air handling unit containing the evaporator of the refrigerating installation, within the Department of Thermotechnics, Engines, Thermal and Refrigerating Equipment, Faculty of Mechanical and Mechatronics Engineering of the National University of Science and Technology POLITEHNICA from Bucharest. The experimental stand within the humid air laboratory is presented in chapter 3, where all the main elements are technically and functionally described, respectively, the air handling unit, the refrigeration plant and the parameter monitoring board, with the attribution of making experimental determinations for new refrigerants.

Chapter 4 focuses on the evaporator in the air handling unit in order to verify the cooling capacity and heat exchange surface with data from the manufacturer's data sheet.

The second heat exchanger in the refrigeration plant is the water-cooled multitube condenser, this can be found in chapter 5, where the condensing thermal power and the heat exchange surface will be checked, compared to the technical data in the technical data sheet of the equipment. In order to find out the optimal and functional solution of the refrigeration plant developed in the laboratory, the exergetic analysis method, from chapter 6, was used to identify and optimize which elements within the system have the highest usable energy (exergy) destroyed. In chapter 7, experimental determinations will be made on the evaporator in the air handling unit that is hydraulically connected to the refrigeration plant. The refrigerant R134a will be substituted with the refrigerants R1234yf and R450A for a comparison keeping the same room temperature of 24°C and implicitly the same refrigerating power, without making changes to the installation.

# CHAPTER 1 - COMPARATIVE STUDY OF THE REFRIGERANTS USED IN THE AIR CONDITIONING INSTALLATION

## 1.1 INTRODUCTION

Refrigerants with suitable thermodynamic properties are selected for cooling or heating applications. Refrigeration technology was used even before electricity was invented in the 1880s. Historically, Oliver Evans (1805) pioneered the refrigeration system using ether as a working refrigerant, Jacob Perkin implemented this idea in his first refrigeration equipment built in 1834. Later, various researchers used gasoline (1860), $NH_3$ (1873), $CO_2$ (1886) and $SO_2$ (1890) as working fluids in refrigeration systems . From the 1830s to the 1930s, the most popular refrigerants were: ether, $NH_3$, $CO_2$, SO, $H_20$, $CCl_4$ , $HCOOCH_3$, HCs, CHC [2].

## 1.2 OBJECTIVE

In this chapter, we will identify the refrigerants that are frequently used in air conditioning installations with air handling units.

The specialized literature will be studied and potential replacements for established refrigerants will be chosen, so that they comply with the new rules of Regulation no. 517/2014 regarding the environment and their negative effects on the environment, but also from an energy point of view, not to operate with a higher electricity consumption than the substituted refrigerant.

## 1.3 MIXTURES OF REFRIGERANTS THAT CAN BE USED IN AIR CONDITIONING INSTALLATIONS

By combining two or more refrigerants, some disadvantages or deficiencies of the lower performing component occur. Various investigations are recently carried out to characterize refrigerant mixtures in order to study their properties and performances [3].

### 1.3.1 Refrigerant mixtures for R134a

The Air Conditioning, Heating and Refrigeration Institute (AHRI) has begun an investigation to study the behavior of new fluorinated refrigerants. Alternative mixtures for refrigerants identified in the literature are presented in Table 1.1. They are classified according to what kind of refrigerant they can replace and their saturation pressure. The full list of AHRI investigations is continuously updated [4].

### 1.3.2 Refrigerant mixtures for R407C

In all stationary refrigeration and air conditioning systems, R407C can be used as an alternative source (GWP<2500), in multipack plant, in commercial refrigeration and air conditioning systems, it can also be used in cascade systems, because their GWP values are above 150 and below 1500 (except ARM-32a whose GWP is above 1500). All refrigerants have high glide values at constant vapor pressure (more than 5°C) [3].

Refrigerant R407C is widely used in water chillers (chillers), it is considered the market leader in stationary average temperature air conditioning [3].

**Table 1.1Composition in percentages regarding refrigerant mixtures** [3]

| Substitute Refrigerants | The mixtures made | R125 | R134a | R152a | R32 | R744a | R1234yf | R1234ze |
|---|---|---|---|---|---|---|---|---|
| **R134a** | R134a | | | | | | | |
| | R1234yf | | | | | | | |
| | R1234ze(E) | | | | | | | |
| | AC5X | | 40% | | 7% | | | 53% |
| | ARM-41a | | 63% | | 6% | | 31% | |
| | D-4Y | | 40% | | | | 60% | |
| | N-13 (R450A) | | 42% | | | | | 58% |
| | XP-10 | | 44% | | | | 56% | |
| | AC5 (R444A) | | | 5% | 12% | | | 83% |
| | ARM-42a | | | 11% | 7% | | 82% | |
| **R407C** | R407C | 25% | 52% | | 2. 3% | | | |
| | ARM-32a | 30% | 25% | | 25% | | 20% | |
| | DR-33 | 25% | 26% | | 24% | | 25% | |
| | ARM-30a | | | | 29% | | | 71% |
| | ARM-31a | | | | 28% | | 21% | 51% |
| | D2Y65 | | | | 35% | | | 65% |
| | DR-7 | | | | 36% | | 64% | |
| | L40 | 10% | | | 40% | 30% | | 20% |
| **R410a** | R410a | 50% | | | 50% | | | |
| | R32 | | | | | | | |
| | DR-55 | 7% | | | 67% | | 26% | |
| | ARM-71A | | | | 68% | | 26% | 6% |
| | L41-2 (R447A) | 3.5% | | | 68% | | | 28.5% |
| | D2Y60 | | | | 40% | | 60% | |
| | L41b | | | | 73% | | | 27% |
| | ARM-70a | | 10% | | 50% | | 40% | |
| | DR-5 | | | | 72.5% | | 27.5% | |
| | HPR1D | | | | 60% | 6% | | 34% |

### 1.3.3 Refrigerant mixtures for R410A

Refrigerant R410A is a mixture composed of R125 / R32 (50% / 50%) with a glide of 0.1°C at atmospheric pressure. It is also considered as an almost azeotropic mixture: it is chemically stable, has low toxicity and is mainly used for air conditioning equipment. R410A has a high operating pressure and a higher heat removal capacity compared to R22 or R12. R410A refrigerant was developed as a replacement for R22 for air conditioning systems [4].

In principle, all the analyzed refrigerants can replace R410A, there are 4 flammable refrigerants with A2L, with close energy performance values [3].

All blends can replace R410A in individual air conditioning systems and stationary refrigeration systems, but not in mobile room air conditioning equipment. R32 is also considered a replacement refrigerant because it has a GWP value below 750, which is the limit for air conditioning [3].

# CHAPTER 2 - ENERGY ANALYSIS REGARDING THE USE OF MIXTURES OF REFRIGERANTS IN THE REFRIGERATION PLANT OF THE AIR HANDLING UNIT

## 2.1 INTRODUCTION

A refrigerant is a substance or a mixture of substances, usually a fluid, which is usually used in thermodynamic systems that go through a reversible phase transition. Refrigerators, air conditioners, heat pumps and many other devices use refrigerants as an intermediate fluid to transfer heat between sources [2].

## 2.2 OBJECTIVE

The main purpose of this chapter is to analyze potential substitutes for refrigerants in the field of air conditioning, both from an energy and environmental impact perspective.

This analysis will be carried out at the theoretical thermodynamic cycle level, for different operating regimes regarding the vaporization temperature for an air conditioning installation.

## 2.3 ANALYSIS OF THE REFRIGERATION CYCLE FOR DIFFERENT REFRIGERANTS

### 2.3.1 Compressor technical data

Cycle level calculations were performed on a given compressor, existing within the refrigeration plant, this is a semi-hermetic compressor, 2KES-05Y-40S, brand BITZER.

It was pre-selected with the Bitzer selection program [5] according to the input data of the air conditioning plant.

### 2.3.2 The specific thermodynamic state of the refrigeration cycle and the state variables of the working refrigerant

In order to determine the state variables of the refrigerant in its evolution at different points of the thermodynamic cycle, the REFPROP program will be used, [6] where the mixture that is to be studied is first defined, thus with the resulting data and the input data analyze the energy performances as well as the influence on the environment.

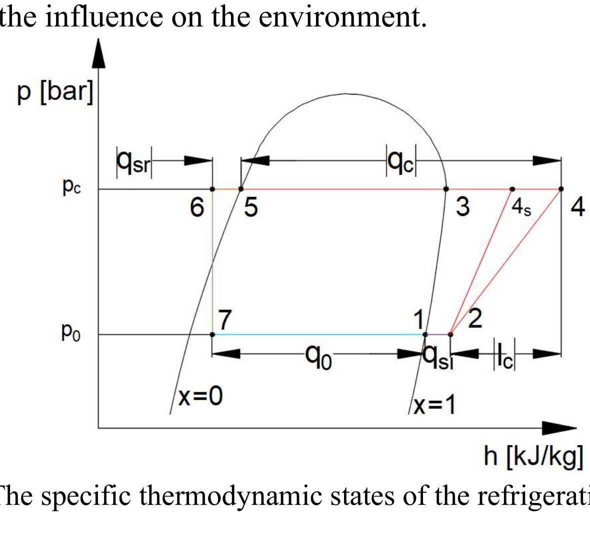


**Fig. 2.1** The specific thermodynamic states of the refrigeration cycle [7]

## 2.4 COMPARISON OF REFRIGERANTS TO REPLACE R134A

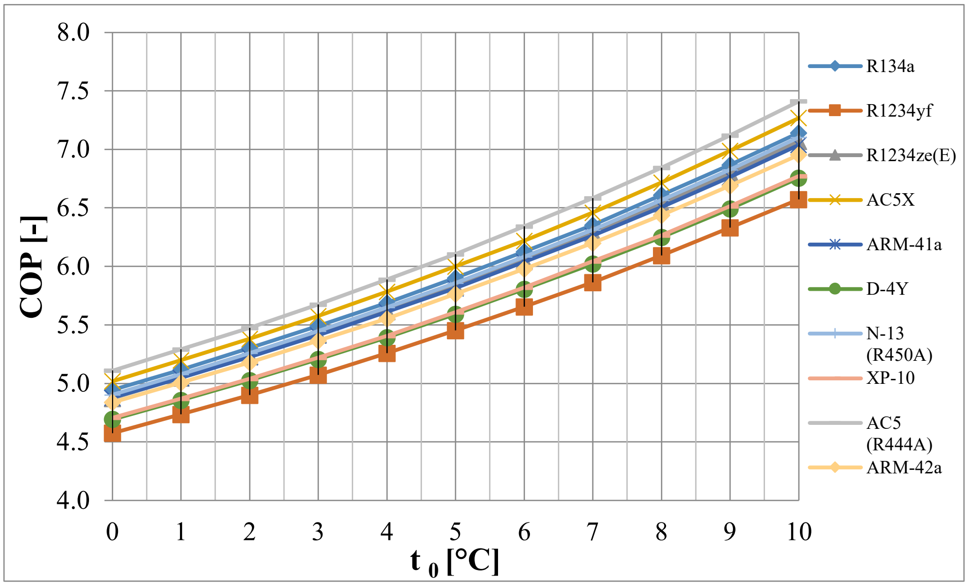


**Fig. 2.2** The influence of vaporization temperature on the coefficient of performance for R134a substitutes

In Fig. 2.2the influence of the vaporization temperature was analyzed in relation to the COP of refrigerants as alternatives to R134a. It can be observed a general trend of increasing COP with increasing vaporization temperature, all the refrigerants studied have the same variation, the trend is the same, thus increasing the value of the function in relation to the variable.

At the bottom end of the graph are the refrigerants with the worst performance, these being: XP-10, D-4Y and R1234yf. R1234yf is an azeotropic refrigerant, and this is the base refrigerant for the two blends, hence the similar performance graph.

In the middle part of the graph, you can find most of the refrigerants around the base refrigerant R134a, with similar energy performance, so they can be analyzed according to other criteria such as that of the environment through the GWP coefficient.

In the upper part of the graph, two refrigerants AC5X and AC5 are highlighted, they have the best energy performance at all vaporization temperatures and compared to R134a they have an increase of 1.5% and 2.4% respectively. The refrigerant behind these blends is R1234ze(E), having the same COP value as R134a, but by creating blends, an refrigerant with good performance at a low GWP was obtained.

In Fig. 2.3the influence of vaporization temperature was analyzed in relation to the COP of refrigerants as alternatives to R407C. It can be observed a general trend of increasing COP with increasing vaporization temperature, all the refrigerants studied have the same variation, the trend is the same, thus increasing the value of the function in relation to the variable. At the bottom end of the chart there is only one refrigerant with the worst performance, which is L40. L40 is a non-azeotropic refrigerant, and its basic refrigerant is R32 with a value of 40%, but it also has 30% $CO_2$ in its composition, which is why it is explained in the graphic representation, having these poor performances.

In the upper part of the graph, you can find most of the refrigerants around the R407C base refrigerant with similar energy performance, so they can be analyzed according to other criteria, such as: that of the environment, through the GWP coefficient, 3 ARM30 refrigerants are highlighted -a, ARM-31a and D2Y65 respectively, these having the best energy performance at all vaporization temperatures and compared to R407C has an increase of 2.1%, 1.1% and 1% respectively.

## 2.5 COMPARISON OF REFRIGERANTS TO REPLACE R407C

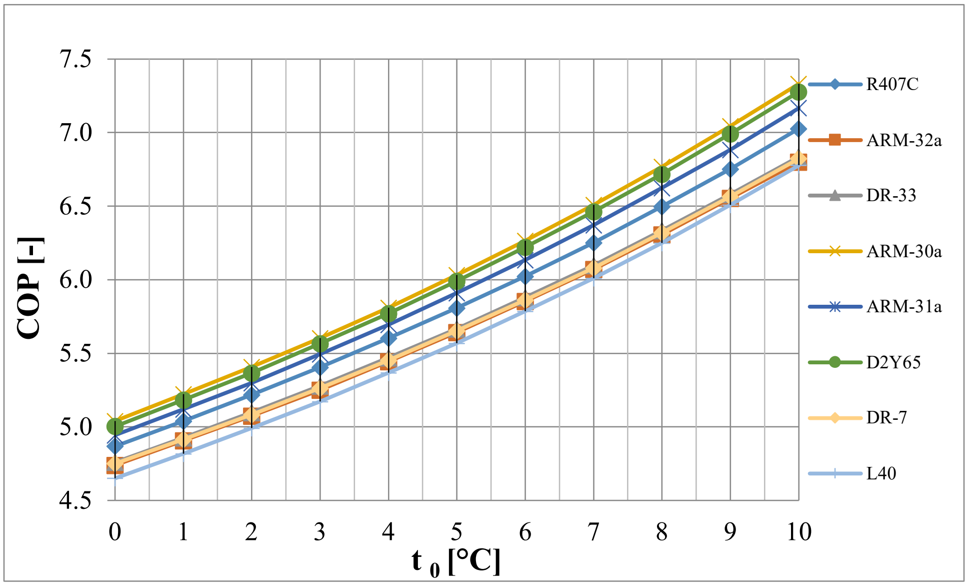


**Fig. 2.3** The influence of the vaporization temperature on the coefficient of performance for substituents of R407C

## 2.6 COMPARISON OF REFRIGERANTS TO REPLACE R410A

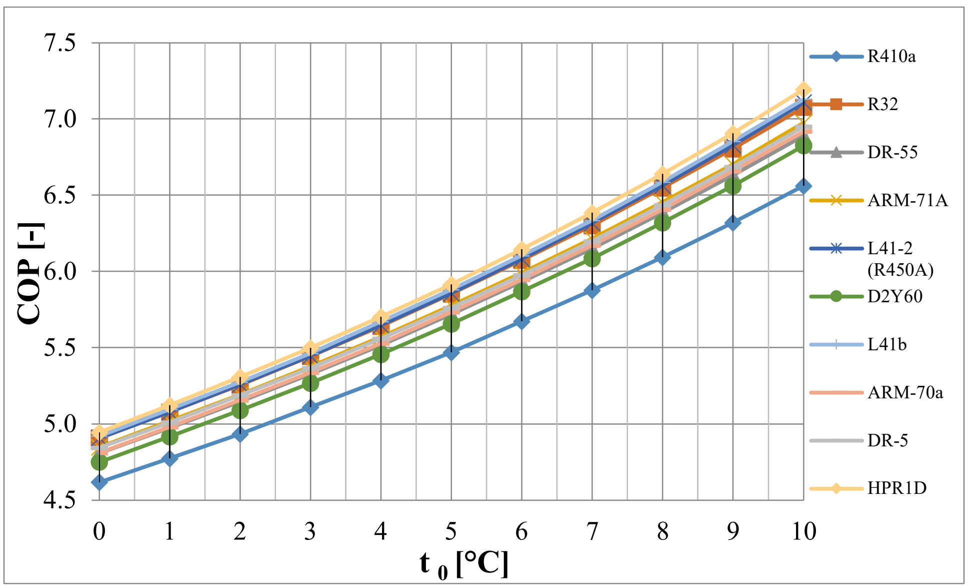


**Fig. 2.4** The influence of the vaporization temperature on the coefficient of performance for substitutes of R410A

In Fig. 2.4the influence of vaporization temperature was analyzed in relation to the COP of refrigerants as alternatives to R410A. It can be observed a general trend of increasing COP with increasing vaporization temperature, all the refrigerants studied have the same variation, the trend is the same, thus increasing the value of the function in relation to the variable. At the bottom end of the graph, there is only one refrigerant with the worst performance, which is R410A.

In the upper part of the graph, most of the refrigerants are highlighted, they have similar energy performance at all vaporization temperatures and compared to R410A, they have an average increase of 2%. The refrigerant behind these blends is R32, all having roughly the same COP value, but by creating blends, an refrigerant with similar performance at a low GWP was obtained.

# CHAPTER 3 - TECHNICAL AND FUNCTIONAL DESCRIPTION OF THE EXPERIMENTAL STAND

## 3.1 INTRODUCTION

In the last decades there is a need to improve buildings through stricter requirements for energy efficiency, thermal performance of buildings and lower energy consumption. In modern low-energy buildings, the big energy consumers are the ventilation and air conditioning systems that are designed to provide fresh and treated air. To save thermal energy in modern buildings, which are well sealed and well insulated, ventilation systems with heat recovery are often recommended. Therefore, the improvement of these systems is of vital importance, to achieve an efficient indoor climate and to reduce energy consumption in buildings [8].

This chapter will discuss: the main components in an air handling unit, the components of the refrigeration plant, what characteristics heat exchangers have and what they are made of.

## 3.2 OBJECTIVE

The main objective of this chapter is to make a technical and functional description of the experimental stand. It consists of an air handling unit, a refrigeration plant and a parameter monitoring board.

The experimental stand has the role of carrying out experimental determinations on the new refrigerants in the air conditioning industry in order to make the systems more efficient in terms of energy consumption and the environment.

## 3.3 AIR HANDLING UNIT

We made the selection of an air handling unit from Fig. 3.1, with the help of the software from the manufacturer [9], with the layout of the modules according to the applications that can be carried out in the laboratory studies, with elements that can improve the air, heat it, cool it, humidify it, dehumidify it and filter it so that the air in that room is at the required parameters.

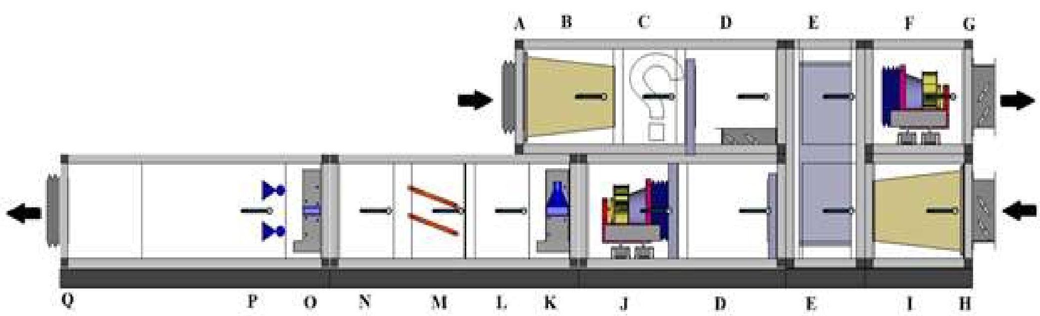


**Fig. 3.1** Scheme of the air handling unit

The air handling unit has the following components:

A. Inlet section;
B. M5 bag air filter module;
C. Automation board;
D. The mixing chamber between outside air and recirculated air;
E. Rotary heat exchanger;
F. Exhaust fan;
G. Adjustment flap with servomotor for exhausting air;

H. Damper with servomotor for fresh air intake;
I. M5 bag air filter module;
J. Supply fan;
K. Direct expansion cooling coil;
L. Free section;
M. Electric heater;
N. Free section;
O. Cold water coil;
P. Steam humidification module;
Q. Discharge treated air [9]

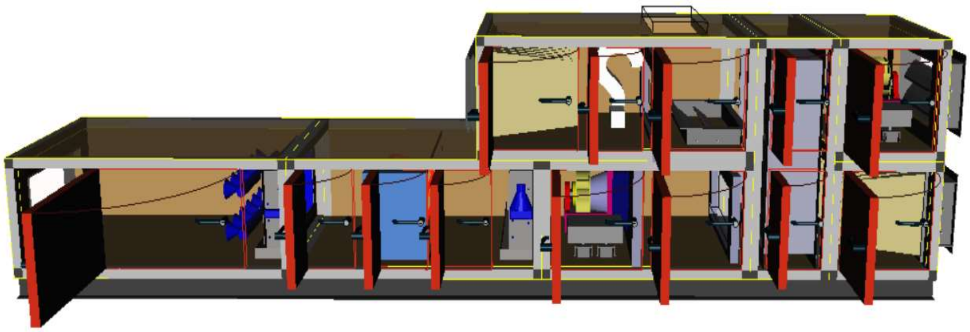

**Fig. 3.2** Air handling unit in 3D format

## 3.4 INSTALLATION EQUIPPING THE AIR HANDLING UNIT

In order to be able to carry out simulations and checks on the air handling unit, it was chosen to connect the evaporator from the ventilation equipment to a refrigerating installation with mechanical vapor compression to carry out the air cooling process.

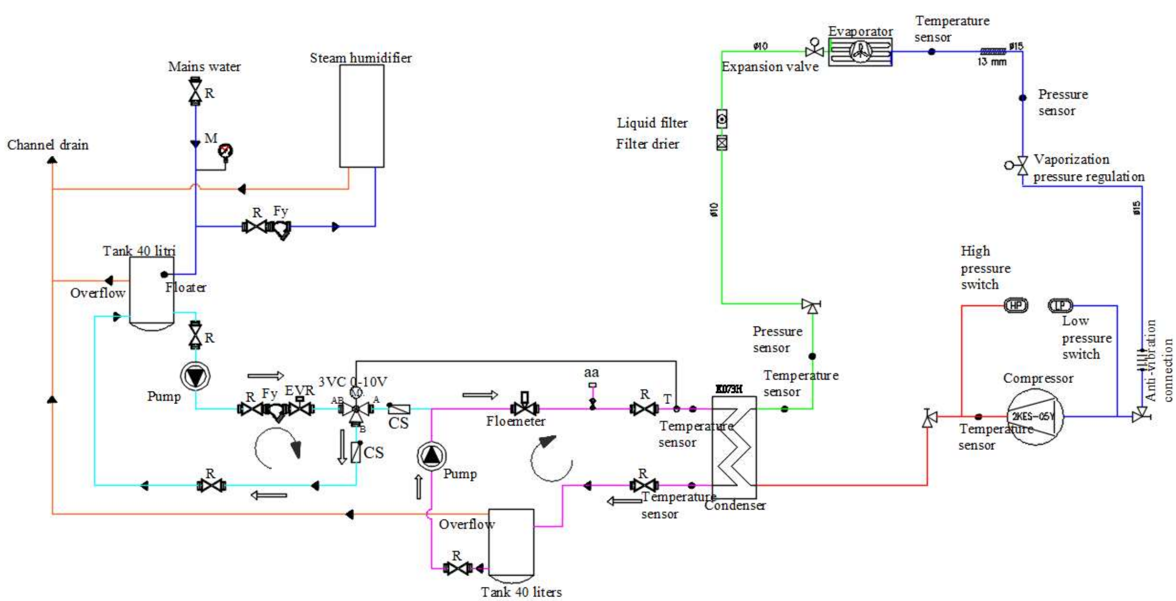


**Fig. 3.3** Hydraulic diagram of the refrigeration plant [7]

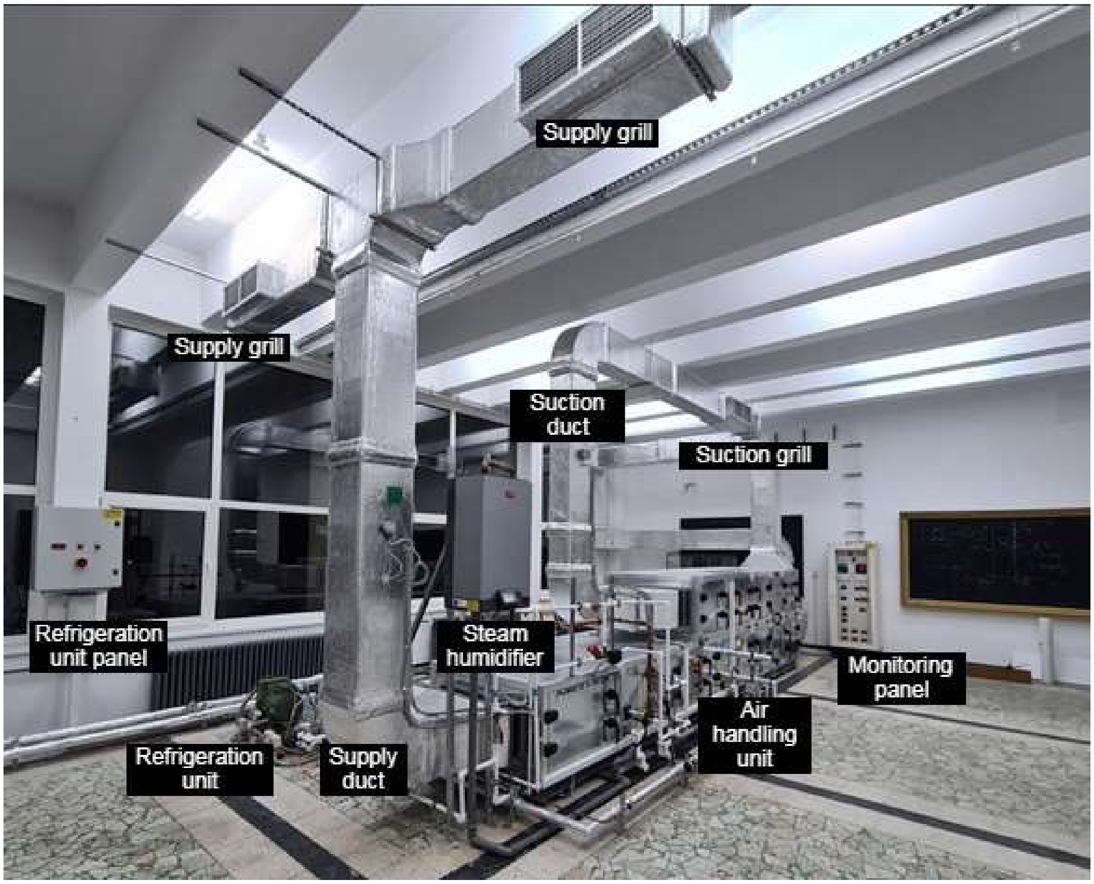


**Fig. 3.4** Installation of air conditioning and ventilation within the laboratory

## 3.5 PARAMETER MONITORING FACILITY

The ventilation and air conditioning equipment in the laboratory can be monitored, turned on and off remotely by means of automations, thus all the parameters of the installation can be recorded, processed and in this way the operating mode of the installation can be established from the XWEB program of the manufacturer Dixell [10].

**Fig. 3.5**The monitoring board of the air conditioning installation and the positioning of the sensors in the air handling unit

In Fig. 3.6 the air handling unit can be seen with the positioning of the sensors and their display in real time.

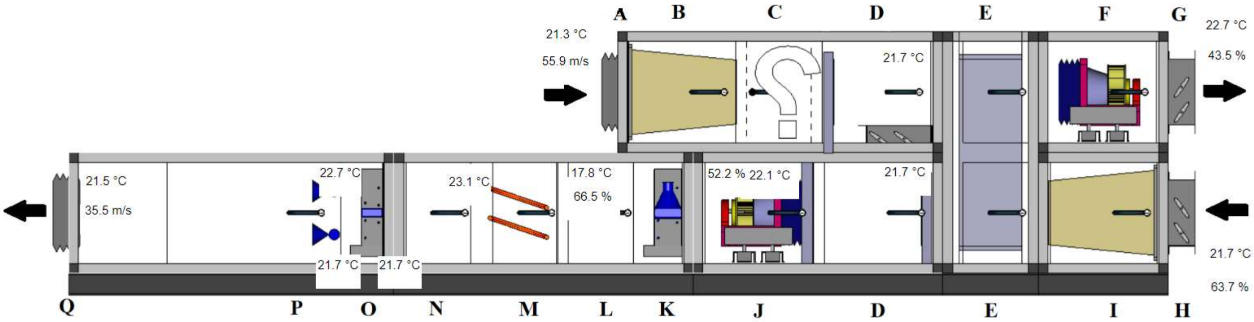


**Fig. 3.6**Parameters monitored at the air handling unit

In Fig. 3.7you can see the hydraulic diagram of the refrigeration plant with the positioning of the sensors and their display in real time, from the XWEB program of the manufacturer Dixell [10].

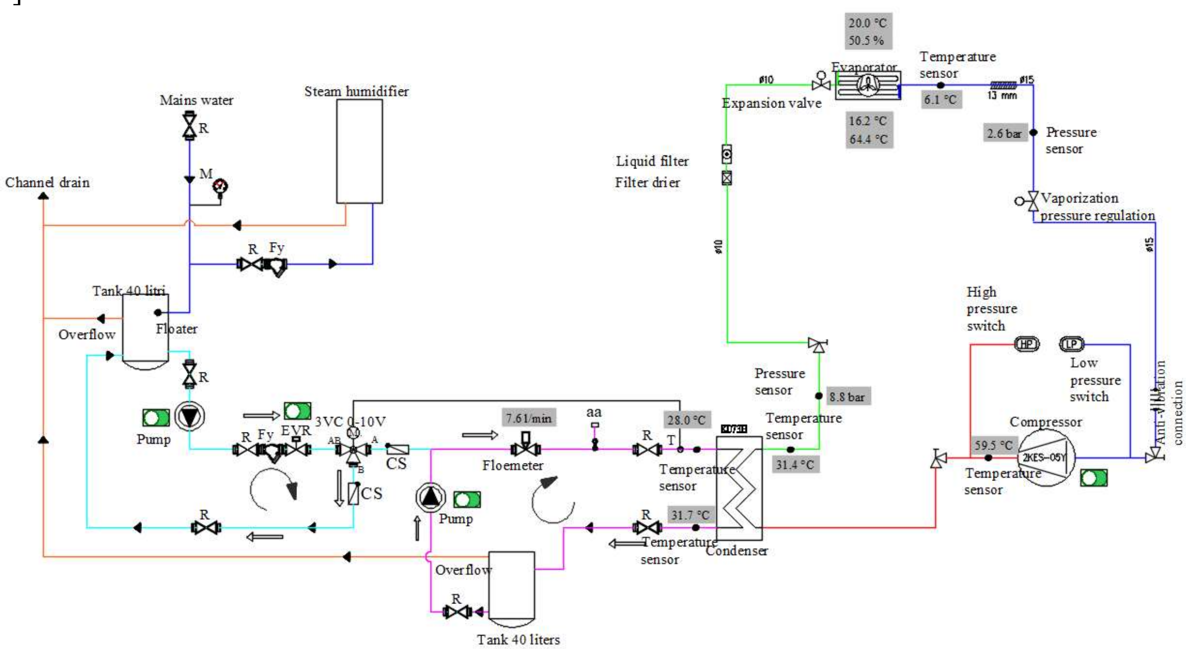


**Fig. 3.7**Parameters monitored at the refrigeration plant [10]

# CHAPTER 4 - MATHEMATICAL MODELING OF THE EVAPORATOR IN THE AIR HANDLING UNIT

## 4.1 INTRODUCTION

Current research in air conditioning is mainly focused on improving overall system performance and efficiency. The evaporator of the installation is a component that significantly influences the overall performance of the entire air conditioning system. However, the effect of evaporator design changes is not always intuitive, and tests are needed to verify and quantify their effects [11].

## 4.2 MAKING THE MATHEMATICAL MODEL OF THE EVAPORATOR

The thermal transfer of the heat that takes place on the surface of the evaporator by cooling the moist air is carried out in the following ways:

- forced convection outside the heat exchanger;
- mass exchange by condensation of relative humidity in the air on the surface of pipes and sheets;
- conduction through the metal wall of pipes and sheets;
- convection boiling;

The evaporator is a surface heat exchanger in which the refrigeration effect is achieved. The process is carried out with phase change, where the refrigerant takes heat from the cooled medium and gives it to the ambient medium with the help of the condenser [12].

### 4.2.1 Checking the evaporator pipe length in the air handling unit

Due to the fact that the evaporator of the refrigerating installation is present in the air handling unit, and the refrigeration power, vaporization temperature and data of the inlet and outlet air are given in the technical sheet of the equipment, the total length of the pipe in this heat exchanger will be checked .

The evaporator in the air handling unit was dismantled, to be measured and checked, and these values are given in the table Table 4.1.

#### *4.2.1.1 Comparison of evaporator criterion equations*

In order to check the length of the evaporator pipe in the air handling unit, the coefficient of construction inside the pipe, the overall heat exchange coefficient and the total pipe length of the evaporator will be calculated with different criterion equations of established authors in the field of refrigeration installations and in particular from the study of heat exchangers.

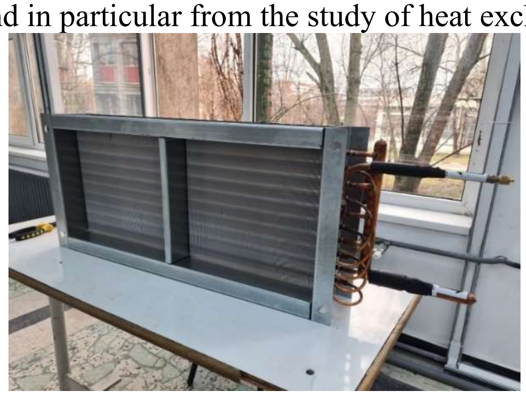

**Fig. 4.1**The evaporator in the air handling unit

**Table 4.1Measured values of the evaporator in the air handling unit**

| **Evaporator** | | | |
|---|---|---|---|
| Length [mm] | 800 | Width [mm] | 30 |
| Height [mm] | 300 | Number of toles [pcs] | 253 |
| Thickness of a sheet [mm] | 0.2 | Inlet connection [mm] | 16 |
| Output connection [mm] | 22 | Number of liquid pipes [pcs] | 5 |
| Number of gas pipes [pcs] | 5 | The diameter of the liquid pipes [mm] | 10 |
| Diameter of gas pipes [mm] | 10 | Length of a pipe [mm] | 800 |
| Total length of pipes [mm] | 8000 | Total number of pipes [pcs] | 10 |
| Pipe inner diameter [mm] | 8 | Pipe material | Cu |
| The material of the sheets | the | | |

A graphical comparison was made with 7 criterion equations as follows: Shah (1982) [13], Gungor and Winterton (1987) [14] and Kandlikar (1999) [15], so as to obtain a value of the exchanger pipe length with that measured in Table 4.1.

These criterion equations were named in this way because of the bibliographic sources where they were found, this does not mean that they are attributed to the specified persons.

In Fig. 4.2shows the total evaporator pipe length obtained using different criterion equations available in the literature.

It can be seen that the second equation of Shah (1982) has different values and does not correspond to the rest of the results.

The criterion equation leading to a pipe length closest to that corresponding to the evaporator mounted in the air handling unit, seen in Table 4.1, is the third equation of Shah (1982). This equation is applicable to a wide range of refrigerants [16].

As a general conclusion, from the results presented in Fig. 4.2, the third criterion equation of Shah (1982) is selected because it leads to a value for the tube length in the evaporator that is closest to the measured one, this can be seen in Table 4.1 and also because it can Table 4.1to a wider range of refrigerants used in the field of air conditioning [13][16].

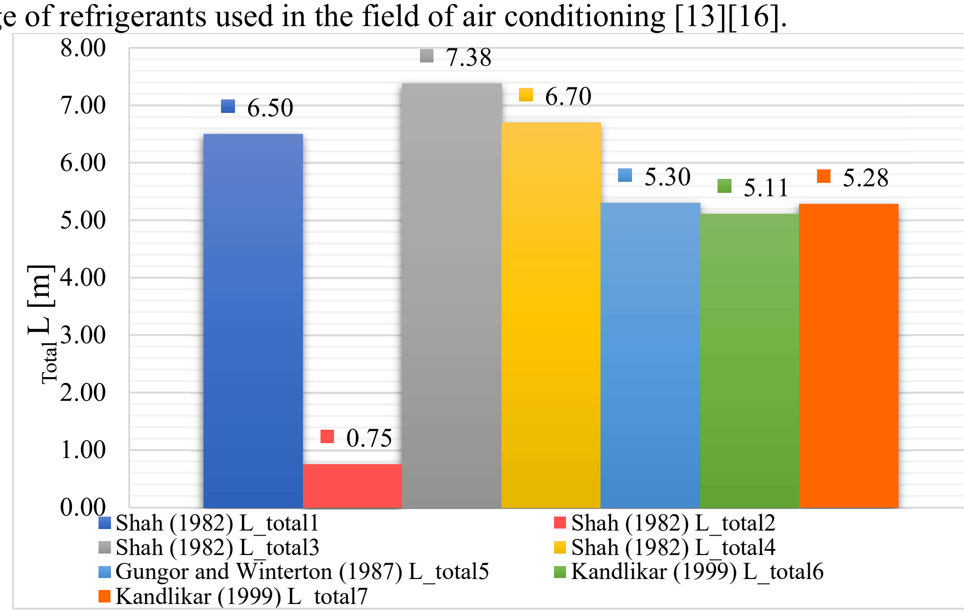


**Fig. 4.2** Graphical representation of $L_{totala}$ with different criterion relations [16]

# CHAPTER 5 - MATHEMATICAL MODELING OF THE MULTITUBULAR CONDENSER

## 5.1 INTRODUCTION

Condensers are important components in a refrigeration and air conditioning installation. Their efficiency must be increased to improve the performance of the entire refrigeration system. Condenser performance can be improved by making efficient heat transfer, increasing the heat exchange surface, by making some fins on the surface of the circular pipe, by making some helical fins inside the pipe where the coolant circulates and thus significantly reduce the size and weight the condenser [17].

## 5.2 MAKING THE MATHEMATICAL MODEL OF THE CONDENSER

The thermal transfer of heat that takes place in the multi-tube condenser through the condensation of the refrigerant takes place as follows:

- the exchange of mass that occurs through the condensation of the refrigerant on the surface of the pipes;
- conduction that occurs through the metal wall of the casing and pipes;
- forced convection inside the pipes;
- convection to condensation;

The condenser is a surface heat exchanger whose role is to evacuate the heat accumulated in the refrigerant to the water. The amount of heat is equal to the sum of the heat absorbed in the evaporator and the heat produced by the compression of the refrigerant vapor in the compressor.

The heat transfer medium is water, which is at a lower temperature compared to that corresponding to the condensing pressure. The process in the condenser can otherwise be compared to the process in the evaporator, except that it is of the opposite "sign", meaning that the conditional change is from vapor to liquid [12].

The refrigeration plant condenser is of the multitubular type and has the dimensions according to Fig. 5.1.

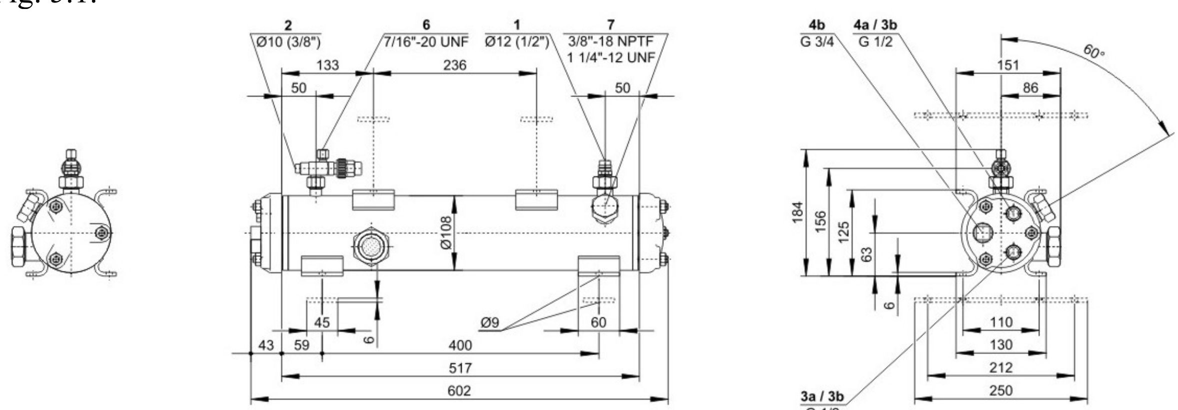


**Fig. 5.1**Condenser dimensions and connections [5]

## 5.3 CHECKING THE PIPE LENGTH OF THE MULTITUBULAR CONDENSER

The multitube condenser is present in the laboratory and is part of a refrigeration installation. It has all the technical data described in the data sheet.

By dismantling it, the following were measured and determined: the number of pipes, the inside and outside pipe diameters, the length of the pipes, the material of the pipes and the material of the casing, according to Table 5.1.

**Table 5.1The measured values of the multitubular condenser**

| | | | |
|---|---|---|---|
| Jacket diameter [mm] | 108 | Width [mm] | 602 |
| Height [mm] | 184 | Refrigerant inlet connection [mm] | 12 |
| Refrigerant outlet connection [mm] | 10 | Water inlet connection [inch] | ¾" |
| Water outlet connection [inch] | ¾" | The number of passes | 4 |
| Pipe outer diameter [mm] | 12.2 | Length of a pipe [mm] | 500 |
| Total length of pipes [mm] | 4000 | Total number of pipes [pcs] | 8 |
| Pipe inner diameter [mm] | 15.9 | Pipe material | With |
| The sheath material | OL | | |

The multi-tube condenser for condensing the refrigerant is a horizontal tube heat exchanger and works with forced convection inside the tubes, with water cooling fluid having a certain flow rate and flow rate. On the outside of the pipes there is a convection to condense the refrigerant, being the highest working pressure in the system, but also by conduction to the wall of the pipes and the heat exchanger casing.

**Fig. 5.2** The multitube condenser

### 5.3.1 Comparison of condenser criterion equations

To check the pipe length of the multi-tube condenser, the convection coefficient on the outside of the pipe, the global heat exchange coefficient and the total pipe length of the condenser will be calculated with different criterion equations of established authors in the field of refrigeration installations and especially in the study of heat exchangers .

A graphic comparison was developed with 8 criterion equations according to the following: Dan Ștefănescu (1987) [18], Ibrahim Karaçayli (2018) [19], Daniel Gstöhl (2004) [20], Adebola S. Kasumu (2017) [21], Warren M. Rohsenow (1998) [22], Chuang-Yao Zhao (2017) [23], Thomas Gebauer (2013) [1], Ravi Kumar (2005) [17], so as to obtain a value of the pipe length of the exchanger with that measured in Table 5.1.

These criterion equations were named in this way because of the bibliographic sources where they were identified, this does not mean that they are attributed to the specified persons.

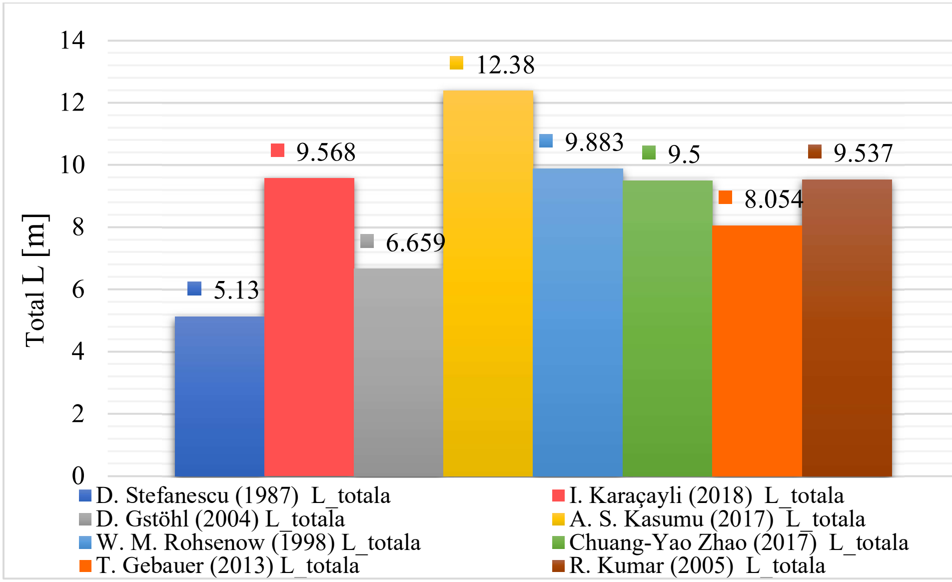


**Fig. 5.3** The graphical representation of $L_{total}$ for different criterion relations

The graph in Fig. 5.3 shows the total pipe length resulting from the criterion equations, where it is observed that if the convection coefficient had small values, larger pipe lengths resulted and where the convection coefficient had larger values, small pipe lengths resulted .

Since the heat exchanger is known and its total measured pipe length is 4 linear meters, there is a benchmark in selecting the appropriate criterion equation.

In the previous chapter, where the calculation of the evaporator and similarly an analysis for the criterion equations to check the length of the heat exchanger was carried out, the results of the equations had varied values, compared to the condenser, where most of the results have values around the measured data.

In the graph, the closest value to the actual length of the existing exchanger is with the criterion equation of Dan Ştefãnescu (1987).

Following these comparisons of criterion equations from the specialized literature, the criterion equation of Dan Ştefãnescu (1987) is chosen because it has the closest total pipe length of the heat exchanger. The criterion equation can be used for a wide range of refrigerants, thus representing a priority in that this condenser will be studied from the point of view of heat transfer for several refrigerants in the field of air conditioning.

# CHAPTER 6 – COMPARISON OF REFRIGERANT MIXTURES USING THE EXERGETIC ANALYSIS METHOD

## 6.1 INTRODUCTION

The increase in electricity consumption is due to economic development and population growth. Thus, the methods of thermodynamic analysis and the operating processes of energy installations are reconsidered, the objective being to increase the efficiency of transformation of heat into mechanical work.

For this purpose, entropy was reintroduced as a measure of the degree of irreversibility of energy processes in which the concept of usable energy introduced by Gouy and updated in the form of exergy proposed by Zorban Rant in 1953 and is represented in an article in 1956.

## 6.2 OBJECTIVE

The objective of this chapter is to carry out an analysis of the air handling unit equipped with a refrigeration plant. Through the exergetic analysis method, it will be identified and optimized which element within the system has the greatest destruction/loss of exergy in order to optimize from a constructive point of view.

For this method of exergy analysis, the main components of the air handling unit that is equipped with a refrigeration plant will be extracted and what elements produce the greatest exergy destruction will be studied and thus create a system that is as efficient as possible in terms of view of the exergetic yield.

## 6.3 STANDARD EXERGY OF PROCESS QUALITY

The way to release energy that can be used in relation to the final equilibrium state can be analyzed from the perspective of the thermodynamic refrigerant system that was in continuous and stationary flow or from the perspective of heat.

Exergy represents the maximum usable mechanical work that a system or an energy carrier can release under the conditions of a specified environment [24].

Exergy can be portable within a control volume in three ways:

- Mechanical work
- heat transfer
- mass transfer [24].

The surrounding environment is represented by the gaseous atmosphere and the earth's crust. It is large enough so that its parameters (p, T, μ) are constant.

The parameters of the environment that relate to any system lead to the variation of the extensive parameters of the environment.

An installation or system is in equilibrium with the external environment when its temperature (T), pressure (p) and composition are the same as those of the environment (dead state) [12].

## 6.4 REALIZATION OF THE MATHEMATICAL MODEL OF EXERGY

The exergy of a substance flow in a given state is composed of two components:

$$ex^{TOT} = ex^{TM} + ex^{CH} \tag{6.1}$$

$$ex^{TM} = ex^{T} + ex^{M} \quad (6.2)$$

The thermo-mechanical exergy will be calculated. In a refrigerant air conditioning system, each stream is in continuous and steady flow.

$$ex^{TM} = h - h_0 - T_0 \cdot (s - s_0) \quad (6.3)$$

$$ex^{TM}(p,T) = h(p,T) - h_0(p_0,T_0) - T_0[s(p,T) - s_0(p_0,T_0)] \quad (6.4)$$

- $h((p,T)\ si\ s(p,T)$– represents the enthalpy and entropy of the system when they are at the pressure $p$and temperature $T$;
- $h(p_0,T_0)$ și $s(p_0,T_0)$– represents the enthalpy and entropy of this system when it is at the pressure $p_0$and temperature $T_0$(in thermodynamic equilibrium with the external environment);

The specific exergy of the refrigerant can be estimated as:

$$ex^{TM} = h - h_0 - T_0 \cdot (s - s_0) \quad (6.5)$$

Humid air is considered an ideal gas containing dry air and a mixture of water vapor. The exergy of the total flow of moist air per kg of dry air was entered through the following equation [25]:

$$ex_a = ex_{Th\_a} + ex_{mec\_a} + ex_{ch\_a} \quad (6.6)$$

$$ex_{Th\_a} = (cp_a + x \cdot cp_v) \cdot T_0 \left(\frac{T}{T_0} - 1 - ln\frac{T}{T_0}\right) \quad (6.7)$$

$$ex_{mec\_a} = (1 + 1{,}608 \cdot x) \cdot R_a \cdot T_0 \cdot ln\frac{p}{p_0} \quad (6.8)$$

$$ex_{ch\_a} = R_a \cdot T_0 \left[(1 + 1{,}608 \cdot x)\ln\left(\frac{1 + 1{,}608 \cdot x_0}{1 + 1{,}608 \cdot x}\right) + 1{,}608 \cdot x \cdot ln\frac{x_0}{x}\right] \quad (6.9)$$

- $cp_{\mathrm{v}}$– represents the specific heat of water vapor at constant pressure;
- $R_{\mathrm{a}}$– represents the specific gas constant of air;
- the constant 1,608 is the ratio of the molar mass of air to the molar mass of water vapor;

## 6.5 DETERMINATION OF THE BEST PERFORMING REFRIGERANT USING THE EXERGETIC METHOD

Comparison of exergy destruction, exergy efficiency, energy consumption and performance are presented below for each of the refrigerants used in the air conditioning study.

### 6.5.1 Exergetic analysis of the installation with refrigerants in order to replace R134a

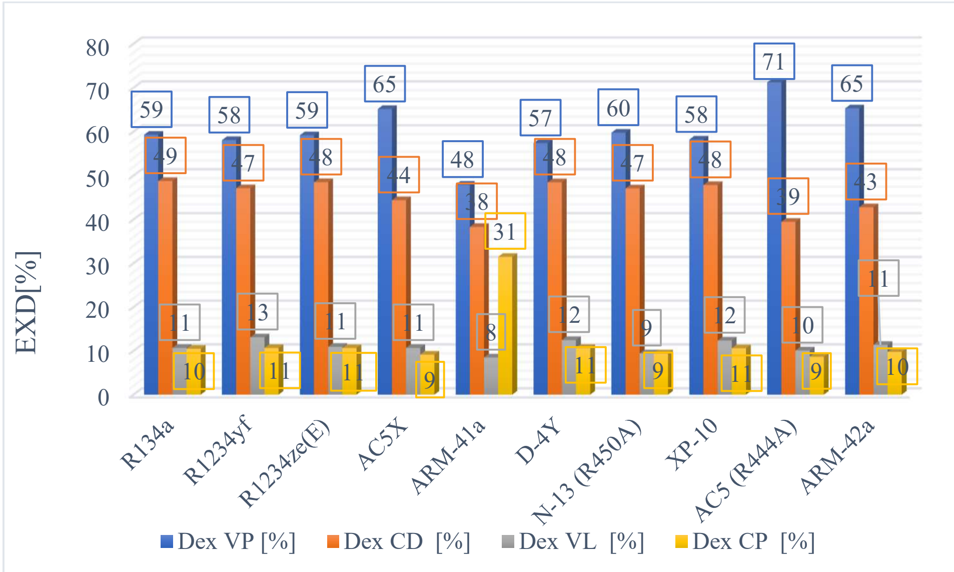


**Fig. 6.1** Graphical representation of exergy destruction in percent for plant components to replace R134a

In theFig. 6.1the substitutes of R134a can be observed with the exergy destructions for each component of the system, where the values are represented as a percentage in order to observe as precisely as possible which component has the greatest destruction. With the lowest percentages of exergy destruction are for the refrigeration compressor (CP) and stand out refrigerant ARM-41a with the highest destruction 31% and refrigerant AC5 with the lowest destruction having the value of 9%.

A component of the plant with similar exergy destruction values to those of the compressor is the lamination valve (VL). It has the highest value for the refrigerant R1234yf at 13% and the refrigerant N-13 has the lowest value at 9%.

The refrigerant condenser (CD) is a multi-tube heat exchanger, where the refrigerant condenses with water, and the exergy destruction is higher compared to the above two components. The refrigerant with the highest value of exergy destruction is R134a with a value of 49%, and the refrigerant ARM-41a has the lowest value, respectively 38%.

The refrigerant evaporator (VP) is a forced convection heat exchanger with the highest exergy destruction of all plant components. The refrigerant with the highest value of exergy destruction is AC5 with a value of 71%, and the refrigerant ARM-41a has the lowest value, respectively 48%.

### 6.5.2 Exergetic analysis of the refrigerant installation in order to replace R407C

In Fig. 6.2 the percentage values for each component of the refrigeration plant are exemplified and the percentages of exergy destruction for the refrigeration compressor (CP) can be viewed. It has the lowest values of all refrigerants analyzed as substitutes for R407C. The next component is the lamination valve (VL) with immediately close values. The heat exchangers in the installation have the biggest damages, due to the phase changes. In the present work, the refrigeration condenser (CD) is multi-tubular, it has high exergy destruction percentage values, but the highest percentages are for the evaporator (VP) of the refrigeration plant that equips an air handling unit due to the large temperature differences between the air and the temperature of vaporization and of the latent component of air.

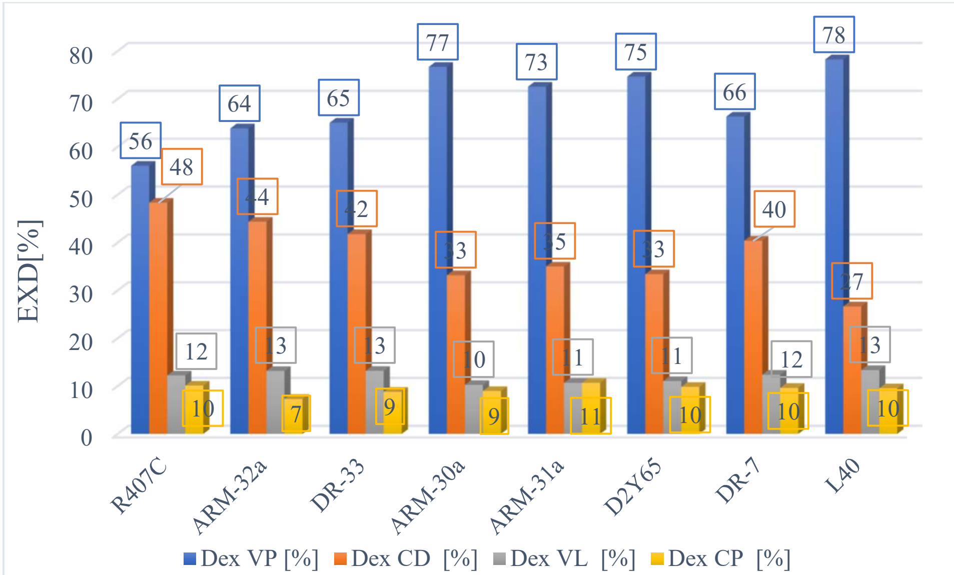


**Fig. 6.2** Graphical representation of exergy destruction in percent for plant components to replace R407C

### 6.5.3 Exergetic analysis of the refrigerant installation in order to replace R410a

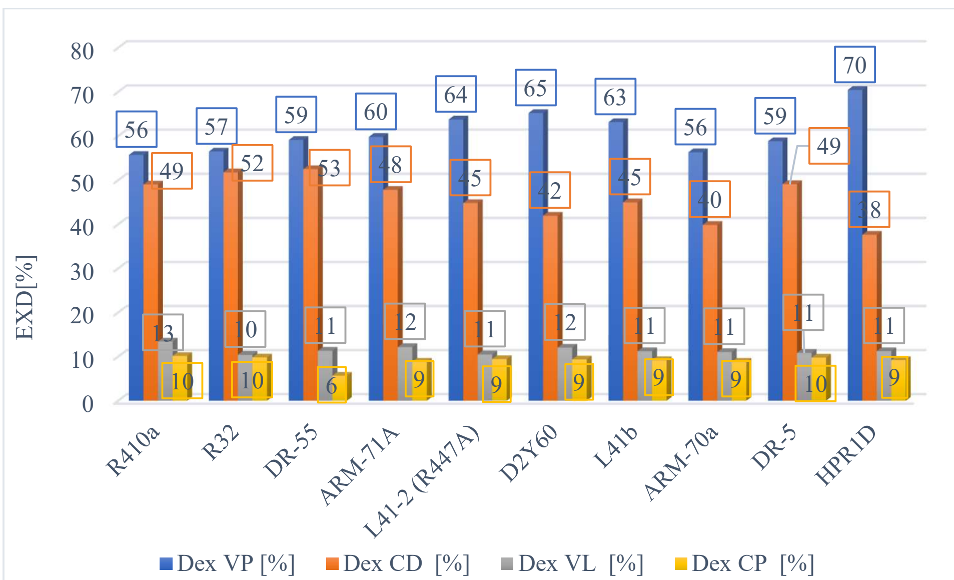


**Fig. 6.3** Graphical representation of exergy destruction in percent for plant components to replace R410a

In this analysis of Fig. 6.3 potential replacements for the refrigerant R410A are studied by making a graph for the main components of the installation and by representing the percentages for these components. The lowest percentage values among all components are given by the refrigeration compressor (CP), where DR-55 has the lowest value. The lamination valve (VL) is the next component, where the lowest value of the total exergy destruction is for the R32 refrigerant. Heat exchangers have the highest exergy losses for all refrigerants analyzed in this study, where the water-cooled condenser (CD) has the lowest losses, the HPR1D refrigerant has the lowest percentage value, and the plant evaporator (VP), has the highest percentages of exergy destruction.

# CHAPTER 7 – EXPERIMENTAL STUDY ON REPLACEMENT OF R134a REFRIGERANT WITH R1234yf AND R450A

## 7.1 THE OBJECTIVE

The main purpose of this chapter is to achieve experimentally what was determined theoretically in the previous chapters, namely the refrigeration power at the evaporator level on the air side and the refrigeration power on the refrigerant side by performing the energy balance, for different vaporization temperatures and for different room temperatures.

In the existing evaporator located in the air handling unit that is equipped with a refrigeration plant, the objective is to keep the same refrigeration power and room temperature of 24°C. Experimentally, this value of 24ºC must be reached with new refrigerants, such as R1234yf and R450A, where the vaporization temperature and the temperature difference are varied.

The target of this experimental study is to replace the R134a refrigerant with the new refrigerants investigated in chapter 1, namely the R1234yf refrigerant and the R450A refrigerant, but to keep the same refrigerating power, with a similar or lower energy consumption, with a GWP coefficient of values within the limits of the rules of Regulation no. 857/2023 and to operate in the field of air conditioning.

## 7.2 ASSESSMENT OF THE REFRIGERATING POWER OF THE EVAPORATOR

In order to evaluate the refrigerating power of the evaporator, three methods of determination are considered as follows:

- the refrigerating power of the evaporator determined in the theoretical way by the criterion equations made in chapter 4.
- the refrigerating power of the evaporator determined experimentally by carrying out the energy balance on the condenser, finding out the mass flow rate of freon, measuring freon pressures and temperatures;
- the refrigerating power of the evaporator determined experimentally by finding the air flow and by measuring the temperature and relative humidity of the air before and after the heat exchanger;

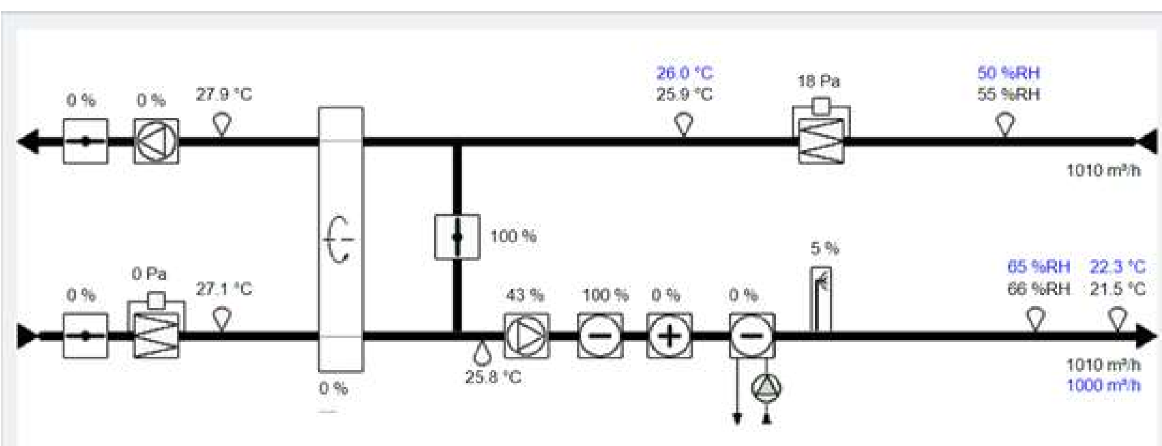


**Fig. 7.1** Automation of the air handling unit [26]

## 7.3 METHOD OF CARRYING OUT EXPERIMENTAL DETERMINATIONS

The air handling unit consists of an automation panel that can control the parameters imposed in the experimental application, so as to obtain the desired measured data in order to calculate the cooling power.

Automatically, the air treatment unit reaches the set values by starting the heating battery with electric resistances and the steam humidifier in percentage, as can be seen in Fig. 7.1.

From the moment the settings have been made in the automation of the air handling unit and it operates at the desired parameters, the monitoring values will have to be followed, according to Fig. 7.2, respectively from the XWEB program [10].

In Fig. 7.2the main values measured in the XWEB program are represented [10], all the values of the experimental parameters are measured and recorded to determine the refrigerating power of the evaporator, where they can also be seen in chapter 4 of this paper.

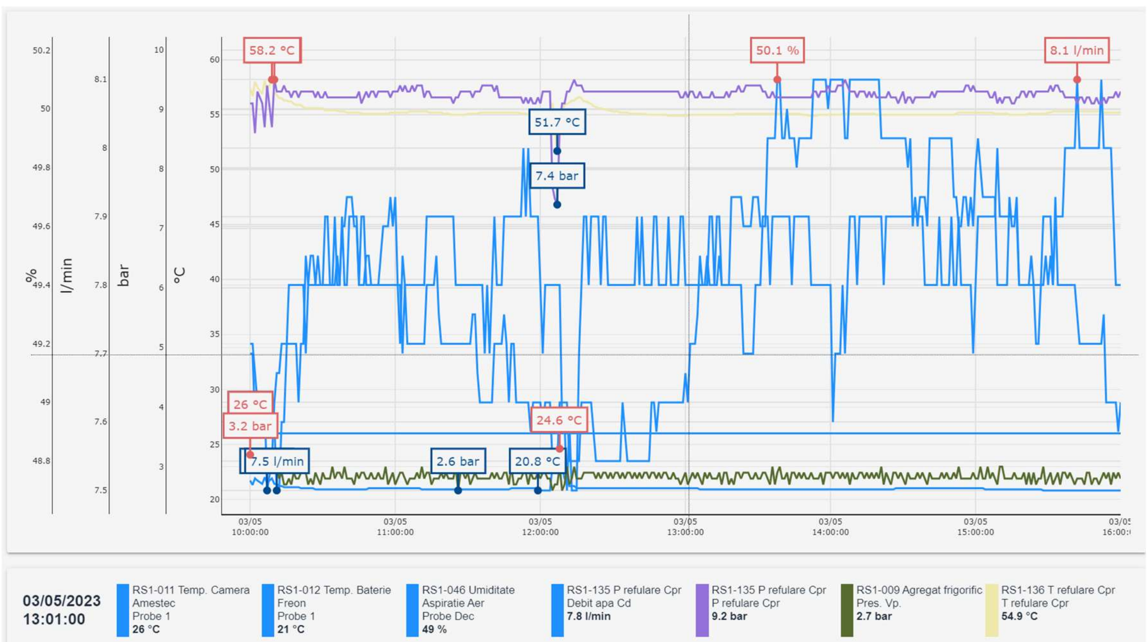


**Fig. 7.2** The monitoring system to determine the cooling capacity [10]

In order to establish an operating regime and select a set of experimental data, set the room temperature in the automation of the air handling unit and adjust the vaporization pressure through the valve at the outlet of the refrigerant from the evaporator, so as to maintain a temperature of quasi-constant vaporization. In order to establish the condensing pressure, a 3-way valve is provided on the cooling path of the multitube condenser, in which a water inlet temperature to the condenser is set and in this way the condensing pressure is quasi-constant at the imposed value.

## 7.4 DETERMINATION OF EXPERIMENTAL DATA

In order to carry out a set of determinations, the air conditioning system operated for 12 hours. During this period of time, an hourly interval was chosen in which the plant operated in a quasi-stationary mode, so that the parameters did not vary much, as can be seen in Fig. 7.2.

This method of choosing experimental plant data was used for all refrigerants and refrigerants analyzed. An example of an operating regime can be seen in Tabel 7.1.

It is known from practice that in air conditioning installations, the evaporator works in a certain regime of ΔT between 14K and 22K, for this reason it was chosen that the operating regime of the evaporator in this paper works in an interval of ΔT of 16K and 20K.

**Tabel 7.1 Experimental data for the calculation of $\dot{Q}_0$ at $t_0$ = 4°C**

| **Room temperature: t $_{approx}$ =20°C** | | **Vaporization temperature: t $_0$ =4°C** | **ΔT=16K** |
|---|---|---|---|
| Date: 21.04.2023 // Time interval: 07:20 – 07:54 // **R134a** | | | |
| Inlet air temperature $t_{a_i}[°C]$ | 20,20 | Superheat temperature $t_{s\hat{\imath}}[°C]$ | 8.00 |
| Outlet air temperature $t_{a_e}[°C]$ | 16.37 | Discharge temperature $t_r[°C]$ | 59.32 |
| The relative humidity of the inlet air $\varphi_{a_i}[\%]$ | 52.70 | Liquid temperature $t_l[°C]$ | 32.90 |
| Outlet air relative humidity $\varphi_{a_e}[\%]$ | 67.95 | Inlet water temperature $t_{wi}[°C]$ | 31.12 |
| Air speed $w_a[m/s]$ | 3.52 | Outlet water temperature $t_{we}[°C]$ | 34.70 |
| Vapor pressure $p_0[bar]$ | 3.39 | Condenser water flow rate $\dot{V}_w$[l/min] | 7.43 |
| Condensation pressure $p_c[bar]$ | 8.73 | | |

## 7.5 EXPERIMENTAL DATA PROCESSING METHODOLOGY

### 7.5.1 Experimental data processing methodology for its determination $\dot{Q}_0$for air

#### *7.5.1.1 Determination of the state quantities at the characteristic points of the air between the inlet and outlet of the evaporator*

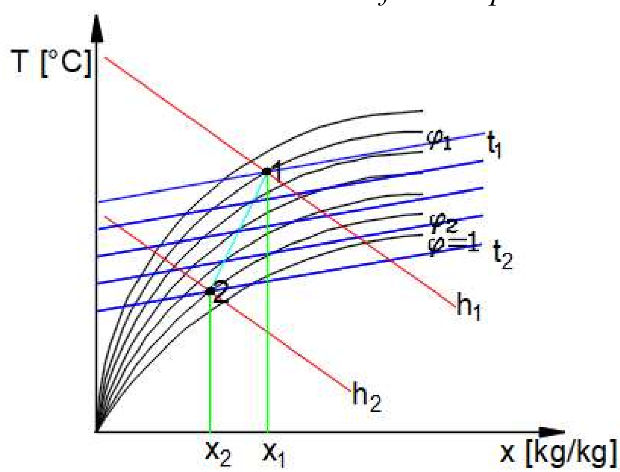


**Fig. 7.3** The specific thermodynamic states of the air entering and exiting the evaporator [7]

### 7.5.2 The methodology of experimental data processing for its $\dot{Q}_0$ theoretical determination

For its theoretical determination $\dot{Q}_0$, the calculation program created in chapter 4 was used, where the criterion equation was identified and the pipe length of the evaporator in the air handling unit in the laboratory was checked.

In this calculation program, the data collected from the experimental determinations were entered, both on the refrigerant side and on the air flow side, and thus the refrigerating power achieved with the chosen criterion equation was obtained.

## 7.6 EXPERIMENTAL RESEARCH ON REPLACEMENT OF R134a REFRIGERANT

The experimental data processing methodology for determining the refrigerating power was applied for each set of experimental determinations above, the results are presented graphically.

### 7.6.1 Refrigerating power resulting in 3 calculation modes for each refrigerant at different Δ T

#### *7.6.1.1 Refrigerating power in 3 modes for R134a at different ΔT:*

In Fig. 7.4shows the influence of the evaporation temperature on the cooling capacity at a difference of 16K between the room temperature and the evaporation temperature, using the R134a refrigerant. The parameters necessary to determine the refrigeration powers were experimentally determined.

In the graph, three curves of variation of the cooling power are represented. These are calculated for the existing evaporator in the air handling unit, as follows: the refrigerating power determined experimentally by the air enthalpy difference (this is shown at the bottom of the graph), the refrigerating power determined experimentally by performing the energy balance on the condenser (this is represented in the middle part) and the refrigerating power theoretically determined by the criterion equations (this is at the top of the graph).

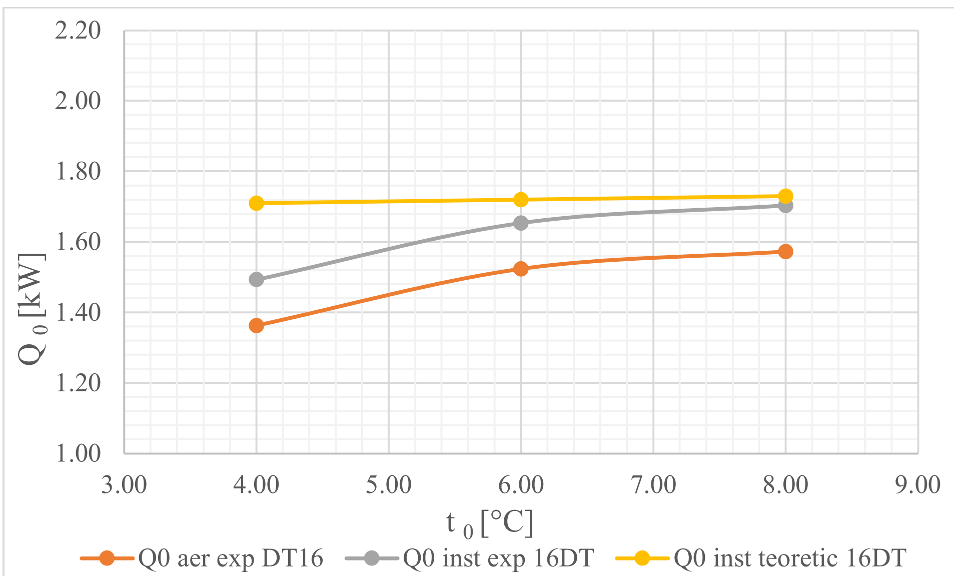


**Fig. 7.4** The influence of vaporization temperature on the cooling power at ΔT=16 degrees

At the bottom of the graph, you can see the variation curve of the cooling power on the air side in relation to the vaporization temperature, where it can be seen that with the increase in the vaporization temperature, the cooling power also increases.

In the middle part of Fig. 7.4shows the variation curve of the cooling power on the refrigerant side on the vaporization temperature, where with the increase in the vaporization temperature, the cooling power also increases. This variation curve is similar to the previous curve, has the same shape, but has an increase of about 10%, in relation to the cooling power of the air.

In the upper part of the graph, the variation curve of the theoretical cooling power on the vaporization temperature is shown, where a quasi-constant variation can be observed with the increase of the vaporization temperature.

The biggest difference between the cooling capacities is at the vaporization temperature of 4°C, where the theoretical cooling capacity is about 25% higher compared to the cooling capacity on the air side. The refrigerating powers with the closest values are at the vaporization temperature of 8°C, where the refrigerating power on the refrigerant side is similar to the theoretical refrigerating power, but with a difference of about 10%.

#### *7.6.1.2 Resulting refrigerating power in 3 modes for R1234yf at different ΔT:*

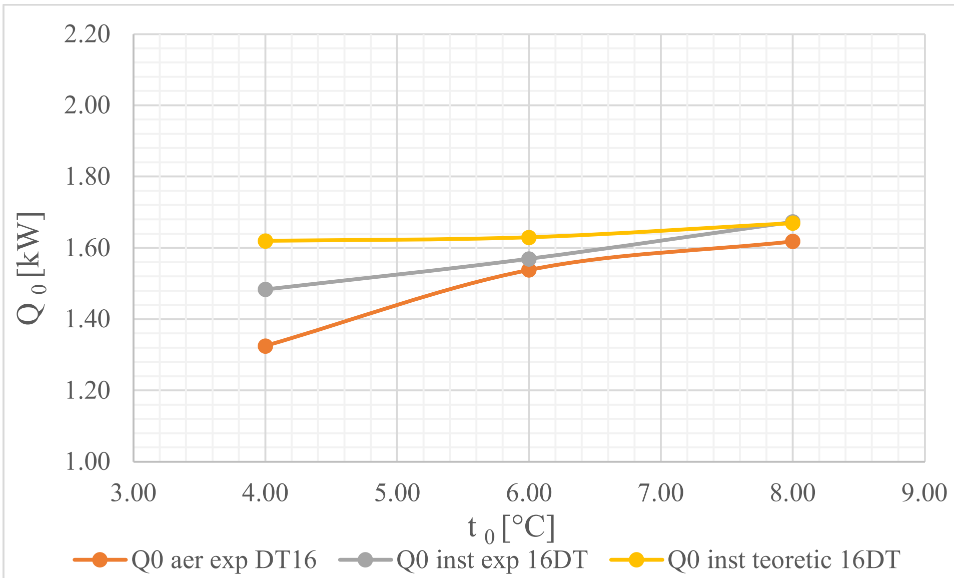


**Fig. 7.5** The influence of vaporization temperature on the refrigerating power at ΔT=16 degrees

In Fig. 7.5the influence of the evaporation temperature on the cooling capacity is represented at a difference of 16 degrees between the room temperature and the evaporation temperature, using the refrigerant R1234yf. The parameters necessary to determine the refrigeration powers were experimentally determined.

The variation curves have a linear increase at all vaporization temperatures, where the cooling power variation curve on the air side with a quasi-linear increase at all vaporization temperatures is highlighted.

The variation curve of the theoretical refrigerating power has the highest values, and it intersects with the refrigerating power variation curve on the refrigerant side, around the value of 7.5°C vaporization temperature.

The two experimentally determined variation curves intersect around the 6°C vaporization temperature value, and at the top of the graph is the theoretical cooling power variation curve with the highest values.

The lowest value is given by the cooling power on the air side at the value of 4°C of the vaporization temperature.

#### *7.6.1.3 Resulting refrigerating power in 3 modes for R450A at different ΔT:*

In the graph in Fig. 7.6the influence of the vaporization temperature on the cooling power is observed at a difference of 16 degrees between the room temperature and the vaporization temperature, using the refrigerant R450A. The parameters necessary to determine the refrigeration powers were experimentally determined.

The general trend for all cooling capacity variation curves is of linear increase with respect to the function variable.

In the lower part of the figure is represented the curve of variation of cooling power on the air side with an increase for all vaporization points analyzed.

In the middle part, the variation curve of the cooling power on the refrigerant side can be observed with a similar shape to that of the air cooling power, with a linear increase for all measured points.

In the upper area, the variation curve of the theoretical cooling power can be found. It has a quasi-constant allure at all vaping temperatures.

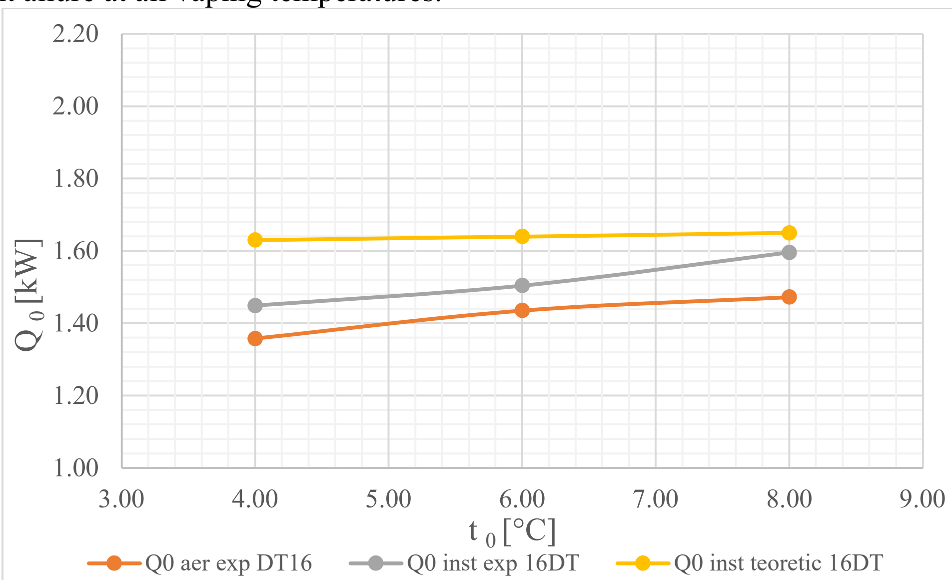


**Fig. 7.6** The influence of vaporization temperature on the refrigerating power at ΔT=16 degrees

At the vaporization temperature of 4°C, there is the largest cooling capacity difference between the cooling capacity on the air side and the theoretical cooling capacity of about 19%. The cooling capacities are closer at the vaporization temperature of 8°C, with a higher value for the theoretical cooling capacity of about 12%, compared to the cooling capacity on the air side.

### 7.6.2 Comparative analysis between refrigerants analyzed at the same cooling power and at the same ΔT

#### *7.6.2.1 Comparative analysis between refrigerants and the resulting cooling power on the air side at the same ΔT:*

In the graph in Fig. 7.7the influence of the vaporization temperature on the cooling power on the air side is observed at a difference of 16 degrees between the room temperature and the vaporization temperature for all refrigerants analyzed experimentally.

The general trend for all variation curves of air cooling capacity is of linear increase with respect to the function variable.

From Fig. 7.7it can be seen that at the vaporization temperature of 4°C all the refrigerants analyzed have the same power, and as the vaporization temperature begins to increase, the variation curves begin to differentiate.

At the bottom, with a lower capacity of refrigerating power is the refrigerant R450A, followed by the refrigerant R134a and with the highest capacity at the vaporization temperature of 8°C is the refrigerant R1234yf.

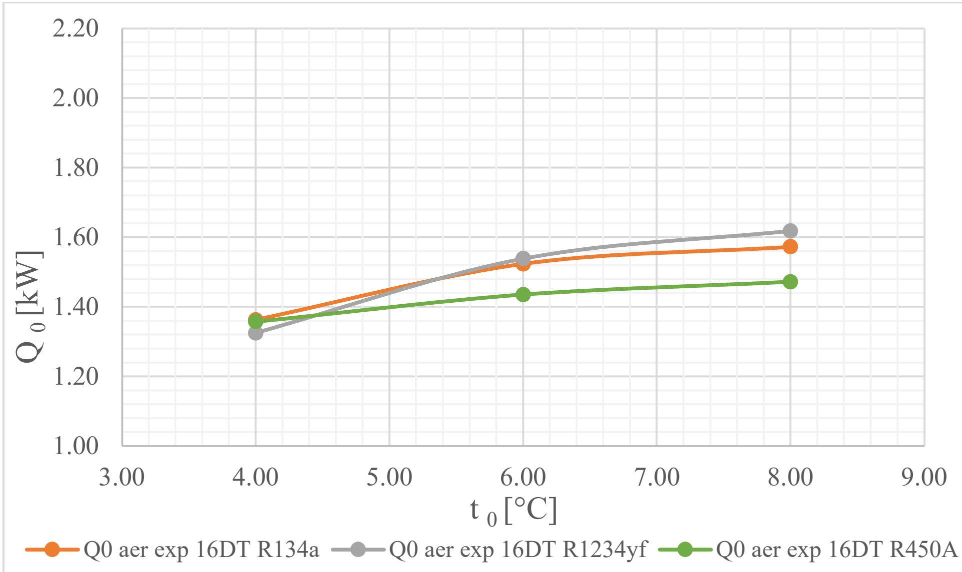


**Fig. 7.7** The influence of vaporization temperature on the cooling power on the air side at ΔT=16 degrees for the analyzed refrigerants

*7.6.2.2 Comparative analysis between refrigerants and the refrigerating power resulting from the thermodynamic cycle of the installation at the same ΔT:*

In the graph in Fig. 7.8represents the influence of the vaporization temperature on the cooling power on the refrigerant side at a difference of 16 degrees between the room temperature and the vaporization temperature for all refrigerants analyzed experimentally.

The variation curve at the bottom is given by refrigerant R450A. This has an increasing function with respect to the variable.

In the middle area of the graph, the variation curve of the refrigerant R1234yf is found, where it has a similar appearance to that of the refrigerant R450A, with values higher on average by 3%.

In the upper part of the graph, the variation curve of the refrigerant R134a is represented. It has an increase in refrigerating power at all vaporization temperatures analyzed.

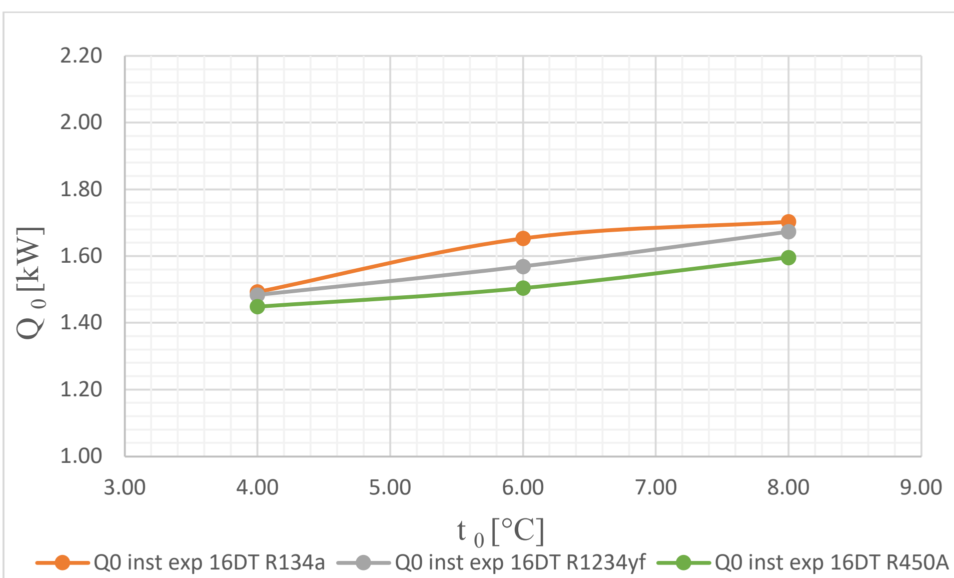


**Fig. 7.8** The influence of vaporization temperature on the refrigerant power on the refrigerant side at ΔT=16 degrees for the analyzed refrigerants

#### 7.6.2.3 *Comparative analysis between refrigerants and theoretically resulting cooling power at the same ΔT:*

In the graph in Fig. 7.9the influence of vaporization temperature on the theoretically resulting refrigerating power at a difference of 16 degrees between room temperature and vaporization temperature for all refrigerants analyzed experimentally is represented.

A general quasi-constant trend of refrigerating capacities with increasing vaporization temperature can be seen in all variation curves.

At the bottom of the graph is the variation curve of the R450A refrigerant with approximately similar values to the R1234yf refrigerant. The two curves have a similar allure.

At the top of the graph is the variation curve of the refrigerant R134a with the highest values. This is quasi-constant over all analyzed vaporization temperatures.

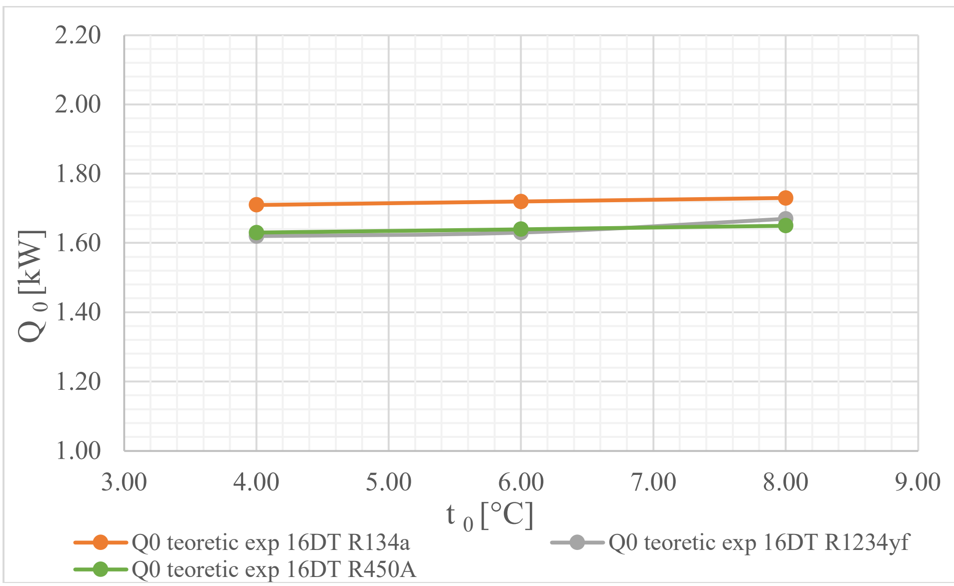


**Fig. 7.9** The influence of vaporization temperature on the theoretically resulting refrigerating power at ΔT=16 degrees for the analyzed refrigerants

### 7.6.3 Results of the variation of refrigerating power and condensing power at the same ΔT

#### 7.6.3.1 *Experimental results of the variation of refrigerating power and condensing power in relation to vaporization temperature for refrigerants R134a and R1234yf*

In the graph in Fig. 7.10the influence of the vaporization temperature on the refrigerating power and the condensing power obtained experimentally at a difference of 16 degrees between the room temperature and the vaporization temperature for the refrigerants R134a and R1234yf is represented.

The general trend for all the variation curves of the refrigerating power and the condensing power is to increase with respect to the function variable.

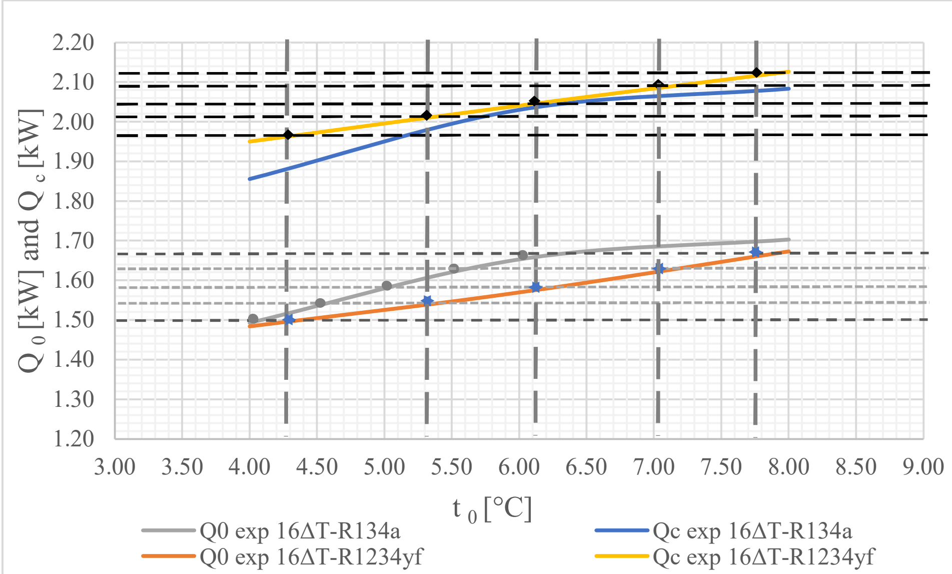


**Fig. 7.10** The influence of the vaporization temperature on the refrigerating power and the condensing power at ΔT=16 degrees for R134a and R1234yf

In this analysis, it is desired to replace the R134a refrigerant with the R1234yf refrigerant, but to obtain the same room temperature and respectively the same refrigerating power.

For the refrigerant R134a, at the vaporization temperature of 4°C, the cooling power value is 1,5 kW to have the same cooling power and for the R1234yf refrigerant, it is necessary that the vaporization temperature be at the value of 4,3°C, but with a higher condensing power of approximately 1,93 kW.

At the vaporization temperature of 4,5°C for the refrigerant R134a, a cooling power value of 1,54 kW is obtained. To have the same refrigerating power with the R1234yf refrigerant, it is necessary for the vaporization temperature to be at the value of 5,3°C, but with a higher condensing power of approximately 2,02 kW.

At the vaporization temperature of 5°C for the R134a refrigerant, a cooling power value of 1,58 kW is obtained. To have the same refrigerating power with the R1234yf refrigerant, it is necessary for the vaporization temperature to be at the value of 6,1°C, but with a higher condensing power of approximately 2,04 kW.

At the vaporization temperature of 5,5°C for the R134a refrigerant, a cooling power value of 1,63 kW is obtained. In order to have the same refrigerating power with the R1234yf refrigerant, it is necessary for the vaporization temperature to be at the value of 7°C, but with a higher condensing power of approximately 2,08 kW.

At the vaporization temperature of 6°C for the R134a refrigerant, a cooling power value of 1,66 kW is obtained. To have the same refrigerating power with the R1234yf refrigerant, it is necessary for the vaporization temperature to be at the value of 7,75°C, but with a higher condensing power of approximately 2,12 kW.

### *7.6.3.2 Experimental results of the variation of refrigerating power and condensing power in relation to vaporization temperature for refrigerants R134a and R 450A*

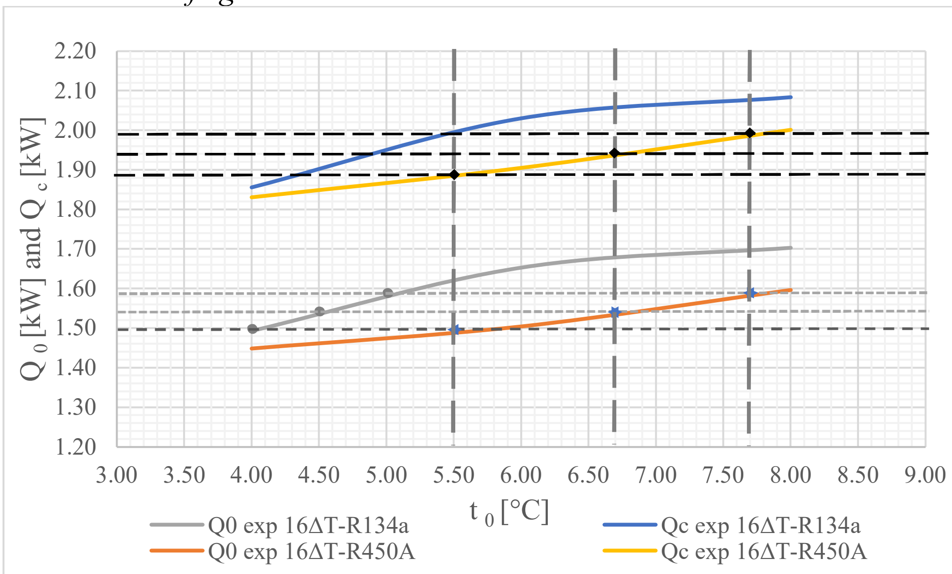


**Fig. 7.11**The influence of the vaporization temperature on the refrigerating power and the condensing power at ΔT=16 degrees for R134a and R450A

In the graph in Fig. 7.11the influence of the vaporization temperature on the refrigerating power and the condensing power obtained experimentally at a difference of 16 degrees between the room temperature and the vaporization temperature for the refrigerants R134a and R450A is represented.

The general trend for all the variation curves of the refrigerating power and the condensing power is to increase with respect to the function variable.

In this analysis, it is desired to replace the R134a refrigerant with the R450A refrigerant, but to obtain the same temperature in the room and respectively the same refrigerating power.

At the vaporization temperature of 4°C for the R134a refrigerant, a cooling power value of 1,5 kW results. In order to have the same refrigerating power with the R450A refrigerant, it is necessary for the vaporization temperature to be at the value of 5,5°C, but with a condensing power of approximately 1,88 kW.

At the vaporization temperature of 4,5°C for the refrigerant R134a, a cooling power value of 1,54 kW is obtained. To have the same refrigerating power with the R450A refrigerant, it is necessary for the vaporization temperature to be at the value of 6,7°C, but with a condensing power of approximately 1,94 kW.

At the vaporization temperature of 5°C for the R134a refrigerant, a cooling power value of 1,58 kW is obtained. In order to have the same refrigerating power with the R450A refrigerant, it is necessary for the vaporization temperature to be at the value of 7,7°C, but with a condensing power of approximately 2,00 kW.

### 7.6.4 Experimental results to determine the operating frequency of the refrigeration compressor at the same ΔT

*7.6.4.1 The results are plotted to determine the operating frequency for the analyzed refrigerants*

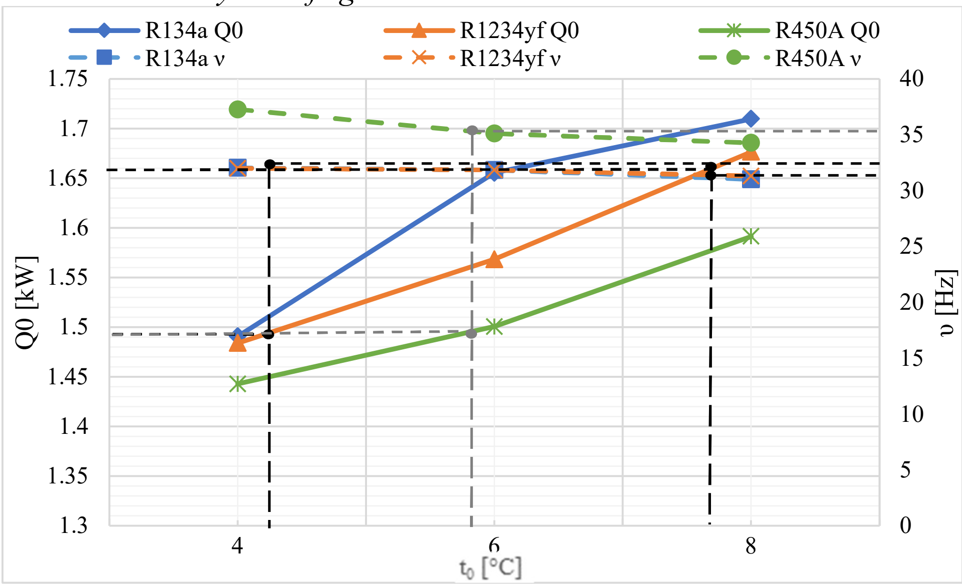


**Fig. 7.12**Influence of vaporization temperature on cooling power and frequency at ΔT=16 degrees for refrigerants R134a, R1234yf and R450A

In Fig. 7.12the influence of the vaporization temperature on the cooling power is represented on the left side of the graph for the refrigerants studied experimentally at a difference of 16 degrees between the room temperature and the vaporization temperature, and on the right side of the figure the operating frequency is shown.

It can be seen that there is a general tendency for all experimentally analyzed refrigerants to increase the refrigerating power in relation to the vaporization temperature. On the other hand, the operating frequency curve has a decreasing slope for all values of vaporization temperatures.

To replace the refrigerant R134a with the refrigerants analyzed experimentally, it is recommended to install a frequency converter on the electric motor of the refrigeration compressor to keep the same refrigeration power, but with an increase in the vaporization temperature and with an increase in the mass flow rate. By means of it, an increase in the condensing power will also result and thus the same temperature in the room will be obtained with another refrigerant.

At the vaporization temperature of 4°C of the R134a refrigerant, a cooling power value of 1,49 kW results. To obtain the same refrigerating power of the R1234yf refrigerant, it must vaporize at a temperature of 4,2°C and contain a frequency value of 33 Hz.

At the vaporization temperature of 4°C of the R134a refrigerant, a cooling power value of 1,49 kW results. To obtain the same refrigerating power of the R450A refrigerant, it must vaporize at a temperature of 5,6°C and contain a frequency value of 36 Hz.

At the vaporization temperature of 6°C of the refrigerant R134a, a cooling power value of 1,66 kW results. To obtain the same refrigerating power of the R1234yf refrigerant, it must vaporize at a temperature of 7,6°C and contain a frequency value of 31 Hz.

# Conclusions

## C1. GENERAL CONCLUSIONS

The main directions that were the basis of the doctoral thesis were: the substitution of classic refrigerants in the air conditioning industry, keeping the refrigeration plant functional at the initial design parameters, with the same refrigeration performance, with a consumption of electricity at similar or lower values and with a GWP coefficient with values within the norms of Regulation no. 857/2023, to an existing air conditioning installation, without making important changes to it.

The doctoral thesis can be summarized in the following structure based on the results obtained:

- Bibliographic research of specialized literature:
    - The most common refrigerants in air handling units are: R134a, R407C and R410A. They have a high GWP coefficient and a major impact on the environment, so that according to the rules of Regulation no. 857/2023 will no longer be able to be used.
    - The new mixtures of refrigerants that can be used in air conditioning systems with air handling units.
    - The selection of 28 new refrigerants from the specialized literature, and they were classified according to the saturation pressure of the 3 reference refrigerants to be replaced.
    - Identifying the criteria equations that fit an air conditioning evaporator from an air handling unit by conducting a bibliographic research of the specialized literature.
    - Identifying the criterion equations that fit a multi-tube condenser in a refrigeration installation by carrying out a bibliographic research of the specialized literature.
- Mathematical modeling:
    - Elaboration of an energy analysis at the theoretical thermodynamic cycle level for each substituted refrigerant from the three groups developed as follows: R134a refrigerant group, R407C refrigerant group and R410A refrigerant group.
    - Designing a mathematical modeling of the evaporator starting from the idea of checking the heat exchange surface and implicitly the pipe length, this value being measured on the existing heat exchanger.
    - Development of a mathematical modeling of the multitube condenser to verify the heat exchange surface and pipe length, this value being measured on the existing heat exchanger.
    - Performing an optimization of the elements within the entire installation by the exergetic analysis method for the three groups of new refrigerants determined in chapter 1 as follows: R134a refrigerant group, R407C refrigerant group and R410A refrigerant group.
- Calculation programs:
    - Using the REFPROP program for new refrigerants to determine the state variables of the refrigerant in its evolution at different points in the thermodynamic cycle.
    - Development of a calculation program for the energy analysis, where it was run for the 28 refrigerants discovered. The results from the calculator were interpreted graphically as a function of vaporization temperature.
    - Completion of a calculation program in EES software, and it has been run for several refrigerants in the air conditioning industry. The results from the calculation program were developed graphically, and in them you can see the title

of vapors entering the evaporator as it directly influences the convection coefficient inside the pipe.
- Design of a calculation program in EES software and this has been run for several refrigerants in the air conditioning industry. The results from the calculation program were developed graphically, and in them it can be seen that with the decrease in the flow of cooling fluid, the condensation temperature increases, the thermal power increases and implicitly the compressor of the installation will work with a higher energy consumption.
- Development of a calculation program for exergetic analysis, where it was run for the 28 refrigerants discovered. The results from the calculation program were interpreted graphically, according to the exergy destruction on each component.

- Experimental stand:
  - Creating a technical and functional description of all the elements within the experimental stand. The main equipment is the air handling unit. This is to obtain fresh and treated air, filtered, at a controlled temperature and humidity, with low energy consumption.
  - In order to have a low temperature in the room, a refrigeration plant was added to the evaporator in the air handling unit. This has the role of obtaining cooled air at the required parameters.
  - Implementation of a monitoring table for the experimental stand in order to achieve experimental determinations, both for the air handling unit and for the refrigeration plant.
  - To check the theoretical values with the practical ones, temperature sensors, pressure sensors, relative humidity sensors, air and water flow sensors were installed.
  - I actively participated in the design, execution and commissioning of the experimental stand. It was fully conceived and realized during the preparation of the doctoral thesis.
  - The plant was operated with the refrigerant R134a during the first experimental determinations, as all the equipment was sized and selected for it.
  - The refrigerant R1234yf is the second refrigerant for which experimental determinations have been made. This has been shown to be a refrigerant at similar working pressures, with approximately 10% less cooling power compared to R134a.
  - R450A is the third refrigerant for which experimental determinations have been made. It has a lower working pressure and refrigerating power compared to the two refrigerants tested by about 15% compared to R134a and 5% compared to R1234yf.
  - characteristic operating curves of the air handling unit were made on the experimental stand by imposing the vaporization temperature at the values of 4°C, 6°C and 8°C, with room temperatures of 20°C, 22°C, 24°C, 26°C, 28°C, 30°C, with refrigerants R134a, R1234yf and R450A.
  - Through these characteristic curves, a graphic method was developed to find the vaporization temperatures of the substituted refrigerant, so that it can be applied in the present case both for the refrigerant R1234yf and for R450A. The substituted refrigerant will have a higher vaporization temperature value, which implies the following variations: increases the mass flow rate, increases the temperature and the heat of condensation. This can be compensated with an increase in the water

flow rate, but the useful effect remains constant, namely the refrigeration power and the temperature set in the room, without significant changes to the installation.

- The second important method of substituting an existing refrigerant is that of the frequency converter characteristic of the electric motor driving the refrigeration compressor. This was determined experimentally for each refrigerant analyzed. For this method, it was determined experimentally what value can be set in the frequency converter if the vaporization temperature changes, respectively an increase in the refrigeration power, by increasing the speed of the electric motor which leads to an increase in the mass flow rate.
- This method does not involve changes to the installation through this frequency converter, it is only connected electrically to the drive motor of the refrigeration compressor, and to it the values determined experimentally for the refrigerants R134a, R1234yf and R450A can be set.
- R1234yf and R450A refrigerants can replace R134a in drop-in mode and the installation works, but to keep the same room temperature of 24°C and implicitly to have the same refrigerating power, it is recommended to install a frequency converter on the motor electric of the refrigerating compressor. In this mode, the initial refrigerating power remains, but with an increase in the vaporization temperature, the mass flow rate and the condensing thermal power, which is compensated by an increase in the water flow rate.

## C2. PERSONAL CONTRIBUTION

Starting from the general conclusions presented above, personal contributions can be summarized as follows:

- Bibliographic research of specialized literature:
  - regarding the study of new refrigerants that can be used in air conditioning installations proposed to replace R134, R407C, R410A;
  - the selection of 28 new refrigerants, from the specialized literature, classifying them according to the saturation pressure of the 3 reference refrigerants, which must be substituted.
  - identifying the appropriate criterion equation for the evaporator in the air handling unit.
  - finding out the criterion equation that corresponds to the technical data of the multi-tube condenser within the refrigerating installation;
- Calculation programs
  - creating refrigerant mixtures using the REFPROP program;
  - creation of a calculation program for the energy analysis of the 28 refrigerant mixtures;
  - drawing up a calculation program with the help of the EES software, for checking the refrigerating power and the heat exchange surface of the evaporator in the air handling unit with the technical data sheet of the equipment, based on the equations and the criteria from the bibliographic research;
  - creation of a calculation program with the help of the EES software, for checking the condensing thermal power and the heat exchange surface of the water-cooled multitubular condenser with the technical data sheet of the equipment, based on the equations and the criteria from the bibliographic research;

  - modeling and exergetic analysis of the air handling unit equipped with a refrigeration plant;
- The experimental stand
  - active participation in the design, execution and commissioning of the experimental stand with an air handling unit equipped with a refrigeration plant within the TMETF-UNSTPB department;
  - creating a system for monitoring the parameters of the air treatment facility with the installation of the following sensors: temperature, pressure, relative humidity, air and water flow, which can be monitored remotely, and the measured and recorded data can be interpreted in the tabular format;
  - experimental determinations made at the air conditioning installation, in different operating modes, with the replacement of refrigerants;

C3. ORIGINAL CONTRIBUTION

- identification of the criterion equation that checks the pipe length of the evaporator in the air handling unit;
- identification of the criterion equation that verifies the pipe length of the multitubular condenser;
- the characteristic operating curves of the air handling unit were made on the experimental stand by imposing the vaporization temperature at the values of 4°C, 6°C and 8°C, with room temperatures of 20°C, 22°C, 24°C, 26°C, 28°C, 30°C. with refrigerants R134a, R1234yf and R450A.
- a graphic method for finding the vaporization temperatures of the substituted refrigerant was developed, so that it can be applied in the present case both for the refrigerant R1234yf and for R450A. The substituted refrigerant will have a higher vaporization temperature value which implies the following variations: increases the mass flow rate, increases the temperature and the thermal power of condensation. This can be compensated with an increase in the water flow rate, but the useful effect remains constant, namely the refrigeration power and the temperature set in the room, without significant changes to the installation.
- a second important method of replacing an existing refrigerant is that of the frequency converter feature of the electric motor driving the refrigeration compressor. This was determined experimentally for each refrigerant analyzed. For this method, it was determined experimentally what value can be set in the frequency converter if the vaporization temperature changes, respectively an increase in the refrigeration power, by increasing the speed of the electric motor which leads to an increase in the mass flow rate. This method does not involve changes to the installation. This frequency converter is only electrically connected to the drive motor of the refrigeration compressor, and through it the values determined experimentally for the refrigerants R134a, R1234yf and R450A can be set.

C4. FUTURE RESEARCH DIRECTIONS

Future research directions may include:

- carrying out experimental determinations for several refrigerants, such as R1234ze(E), R444A and R513A;
- creation of a calculation program for the refrigeration compressor of the installation;
- installation of a frequency converter on the electric motor of the refrigeration compressor;

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**Uta Iulian**; Apostol Valentin; Pop Horatiu; Pavel Constantin; Alqaisy Saleh Jassim Saleh; Badescu Viorel; Taban Daniel; Ionita Claudia, „Mathematical modeling of an Evaporator by using different criterial equations", INMATEH-AGRICULTURAL ENGINEERING, Vol. 67, Issue 2, Page 562-572, Published MAY-AUG 2022, Indexed 2022-11-21. The work is indexed in BDI (Scopus)DOI 10.35633/inmateh-67-55 and ISI Thomson Reuters WOS:000883607300001, ISSN 2068-4215, eISSN 2068-2239. According JCI Category AGRICULTURAL ENGINEERING *in ESCI edition,* Category Rank14/17, Category Quartile Q4.
https://www.webofscience.com/wos/woscc/full-record/WOS:000883607300001

Ionita C; Vasilescu EE, Popa, L., Pop Horatiu, Alqaisy, Saleh Jassim Saleh, **Uta Iulian**, „Exergy analysis of liquid air energy storage system based on Linde cycle". INMATEH-AGRICULTURAL ENGINEERING, Volume 67, Issue 2, Page 543-552, ISSN 2068-4215, eISSN 2068-2239. The work is indexed in ISI Thomson Reuters WOS:000869058900001,DOI 10.35633/inmateh-67-53, Published MAY-AUG 2022, Indexed 2022-10-22. According JCI Category AGRICULTURAL ENGINEERING *in ESCI edition,* Category Rank14/17, Category Quartile Q4.
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https://www.webofscience.com/wos/woscc/full-record/WOS:000996501400046

Horaţiu Lucian POP (first author), Valentin Gheorghe APOSTOL (coordinator), Constantin PAVEL, Melisa Gabriela TOADER, **Iulian UȚA**, PROCESE ÎN INSTALAȚIILE DE CONDIȚIONARE A AERULU, ISBN 978-606-9608-57-9, 2023. Politehnica Press.

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