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Fin&tube heat exchanger optimization in a propane heat pump: theoretical analysis and experimental tests

Relatore: Prof. Anna Stoppato

Studente: Paolo Rossi

Matricola: 2018742

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Preface

This thesis was carried out in CLIVET S.p.A. under the supervision of Anna Stoppato, Andrea Abitani and Alessandro Talini.

It is assumed that the reader has a basic knowledge of thermodynamics and heat transfer in order to better understand the main topics covered and the main results.

Riassunto

Questo lavoro di tesi riguarda l'ottimizzazione di scambiatori di calore fin and tube funzionanti a propano tramite il dimensionamento teorico e il confronto con prove sperimentali.

Sono state prese tre pompe di calore originalmente operanti a R32 e R513A e dimensionati gli scambiatori di calore per poter funzionare con il propano.

È stata condotta un'analisi teorica via software (Coildesigner) degli scambiatori a geometria fissata e al variare del numero di circuiti sono stati analizzati: caduta di pressione, carica di refrigerante, temperature di condensazione ed evaporazione

Per selezionare scambiatori di calore adatti che soddisfino i requisiti del progetto, sono state introdotte limitazioni pratiche. Queste limitazioni includevano la velocità minima del refrigerante necessaria per il ritorno dell'olio a carico nominale e parziale. Il processo di selezione è stato effettuato per ogni scambiatore di calore, e quelli scelti sono stati sviluppati e installati sulle rispettive unità.

Gli scambiatori di calore sono stati sottoposti a test sia nelle operazioni di riscaldamento che di raffreddamento. I dati teorici ottenuti in precedenza sono stati confrontati con i dati pratici delle prove.

Come osservato, il software tende a sopravvalutare le temperature di evaporazione e sottovalutare le temperature di condensazione e le cadute di pressione in tutti i test. Modificare le correlazioni utilizzate nel software per la perdita di carico e il trasferimento di calore si è rivelato difficile. Tuttavia, introducendo fattori correttivi che agiscono sulla correlazione, è stato possibile regolare i parametri che si discostavano dal caso reale, rendendo così lo scambiatore di calore simulato più accurato.

Un processo iterativo è stato impiegato per perfezionare i coefficienti, ottenendo infine un'eccellente simulazione dei parametri con una deviazione inferiore al 15%. Grazie a questo cambiamento, l'errore software è stato ridotto dal 46.9% all'10.7% sulla caduta di pressione, dal 42.7% al 5.4% sulla temperatura di evaporazione e dal 5.9% allo 0.3% sulla temperatura di condensazione.

Questo lavoro faciliterà il dimensionamento dei futuri scambiatori di calore per alette e tubi funzionanti con R290. Le conoscenze acquisite da questo studio forniranno, in anticipo, informazioni sui valori effettivi di caduta di pressione, evaporazione e temperature di condensazione, semplificando il processo di dimensionamento.

Abstract

This thesis project focuses on optimizing the performance of fin and tube heat exchangers when operating with propane. The optimization is carried out through theoretical sizing and a comparison with experimental tests.

Initially, three heat pumps were operating using R32 and R513A, but the heat exchangers were designed to work with propane instead.

Using the Coildesigner software, a theoretical analysis of the fixed geometry exchangers was conducted. Various factors were analysed as the number of circuits changed, including pressure drops, refrigerant charge, condensation and evaporation temperatures. These factors were simulated and compared for each unit during both heating and cooling operations.

To select suitable heat exchangers that meet the project's requirements, practical limitations were introduced. These limitations included the minimum refrigerant speed necessary for the return of oil at nominal and partial load. The selection process was carried out for each heat exchanger, and the chosen ones were developed and installed on their respective units.

The heat exchangers underwent testing in both heating and cooling operations. The theoretical data obtained earlier was compared with the practical data from the tests.

As observed, the software tends to overestimate evaporating temperatures and underestimate condensing temperatures and pressure drops in all the tests. Modifying the correlations used in the software for pressure drop and heat transfer proved challenging. However, by introducing corrective factors that act on correlation, it was possible to adjust the parameters that were deviating from the real case, thereby making the simulated heat exchanger more accurate.

An iterative process was employed to fine-tune the coefficients, eventually achieving excellent parameter simulation with a deviation of less than 15%. Thanks to this change, the software error was reduced from 46.9% to 10.7% on the pressure drop, from 42.7% to 5.4% on the evaporation temperature and from 5.9% to 0.3% on the condensation temperature.

This work will facilitate the sizing of future fin and tube heat exchangers operating with R290. The knowledge gained from this study will provide, in advance, information about the actual values of pressure drop, evaporation, and condensation temperatures, simplifying the sizing process.

Acknowledgements

First of all, I would like to express my sincere thanks to my rapporteur Professor Anna Stoppato for the valuable suggestions, the fundamental indications and the constant support she has given me throughout this period.

I would also like to thank Dario and all my colleagues who, with their deep knowledge and patience, have supported me in these months.

Moreover, I would like to thanks girlfriend, mum, sister and Andrea for supporting me all these years during my studies.

Sommario

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Nomenclature

Abbreviation

F-gas	fluorinated gases
HCF	hydrofluorocarbon
GWP	global warming potential
ODP	ozone depletion potential
PFAS	per- and polyfluoroalkyl substances
HVAC	heating, ventilation and air conditioning
COP	coefficient of performance
H.E.	heat exchanger
DHW	domestic hot water
ATEX	appareils destinés à être utilisés en ATmosphères EXplosibles (French for "Equipment intended for use in explosive atmospheres")
SC	subcooling
SH	superheating
Rps	rotate per second

Symbols

T	temperature [°C], [K]
P	power [kW]
p	pressure [bar] [kPa]
Δ	difference between two terms
$\dot{m}$	mass flow rate [kg/s]
h	enthalpy [kJ/kg]
Nu	Nusselt number
Re	Reynolds number
Pr	Prandtl number
ρ	density [kg/m^3]
μ	viscosity [m^2/s]

λ	thermal conductivity [W/mK]
v	velocity [m/s]
S	surface [m^2]
cp	specific heat [J/kgK]
L	length of tube [m]

Subscripts

cond	condensing
evap	evaporating
ext	external
exch	exchanged
disch	discharge
fan	fan
air	air fluid
max	maximum

Chapter 1: Propane as new refrigerant

1.1 Introduction

Climate change is one of the greatest challenges our world has ever faced. The increase in average global temperatures is in fact affecting our climate and if not contrasted with concrete actions, are likely to worsen in the coming years. Due to ozone depletion leading to the effects of global warming, environmentally friendly refrigerants with ozone depletion (ODP) and low global warming potential (GWP) are needed for the new generation of refrigerators and air conditioners.

In this chapter, the main refrigerant alternatives will be analyse with a particular attention to propane R290 that is the refrigerant fulcrum of this thesis.

1.2 EU legislation to control F-gases and PFAS

1.2.1 F-gas Regulation from 2015

Fluorinated gases (F-gases) are a family of human-made gases used in a range of everyday products as well as industrial applications. The EU is taking regulatory action to control F-gases as part of its policy to combat climate change.

F-gases are often used as substitutes for ozone-depleting substances, because they do not damage the atmospheric ozone layer. However, F-gases are powerful greenhouse gases, with a global warming effect up to 25 000 times greater than carbon dioxide (CO2).

The emissions of F-gases in the EU doubled from 1990 to 2014 – in contrast to emissions of all other greenhouse gases, which were reduced. However, thanks to EU legislation on fluorinated gases, F-gas emissions have been falling every year since 2015 (EEA data).

Hydrofluorocarbons (HFCs) are by far the most relevant F-gas group from a climate perspective, although they are relatively short-lived. The other two F-gas groups, perfluorocarbons (PFCs) and sulphur hexafluoride (SF6), can remain in the atmosphere for thousands of years.

To control emissions from fluorinated greenhouse gases (F-gases), including hydrofluorocarbons (HFCs), the European Union has adopted two legislative acts: the F-gas Regulation and the MAC Directive.

The current F-gas regulation, which applies since 1 January 2015, replaces the original F-gas Regulation adopted in 2006. The implementing Regulations of the original Regulation remain in force and continue to apply until new acts are adopted.

The current regulation has reinforced the previous measures and introduced far-reaching changes by:

- Limiting the total amount of the most important F-gases (HFCs) that can be sold in the EU from 2015 onwards and phasing them down in steps to one-fifth of 2014 sales in 2030.
- Banning the use of F-gases in many new type of equipment where less harmful alternative are widely available
- Preventing emissions of F-gases from existing equipment by requiring checks, proper servicing and recovery of the gases at the end of the equipment's life

Thanks to the F-gas Regulation, the EU's F-gas emissions will be cut by two-thirds by 2030 compared to 2014 levels.

1.2.2 Review of the EU F-gas Regulation and the new Commission proposal

On 5 April 2022, the Commission made a legislative proposal to update Regulation (EU) No 517/2014 (the 'F-gas regulation'). This proposal will now be negotiated by the co-legislators in the European Parliament and the Council. The Commission is proposing to align the F-gas Regulation with the European Green Deal and the European Climate Law, try to follow the Montreal Protocol.

The review is intended, in particular, to:

- Deliver higher ambition e.g. through a tighter quota system for HFCs: reduce the amount of HFCs placed on the market by 98% by 2050 (compared to 2015). New restrictions on the use of F-gases in equipment are also included
- Ensure compliance with the Montreal Protocol, e.g. making phase-down steps also after 2030
- Improve enforcement and implementation, a quota price will be introduced, and penalties will become more stringent and more homogenous across the EU
- Achieve more comprehensive monitoring, e.g. by covering a broader range of substances and activities and improving the procedures for reporting and verifying data.

On this basis, the Commission has chosen a package of measures that will prevent emissions amounting to 40 $MtCO_2$ by 2030 and 310 $MtCO_2$ by 2050.

The substantial information emerging from the proposal on refrigerants can be seen in the table below:

Product	Range	Requirement	When
Self Contained	All	GWP max 150	1/01/2025
Split	<12 kW	GWP max 150	1/01/2027
	>12 kW	GWP max 750	

Table 1: Proposal resume

1.2.3 PFAS proposal regulation

In addition to the regulatory proposal for F-gases, a proposal has also been presented that aims to reduce PFAS emissions to the environment and to make products and processes safe for people.

Per- and polyfluoroalkyl substances (PFASs) are a large class of thousands of synthetic chemicals that are used throughout society. However, they are increasingly detected as environmental pollutants and some are linked to negative effects on human health. They all contain carbon-fluorine bonds, which are one of the strongest chemical bonds in organic chemistry. This means that they resist degradation when used and also in the environment. Most PFASs are also easily transported in the environment covering long distances away from the source of their release. PFASs have been frequently observed to contaminate groundwater, surface water and soil. Cleaning up polluted sites is technically difficult and costly. If releases continue, they will continue to accumulate in the environment, drinking water and food. The majority of PFASs are persistent in the environment.

PFAS substances include a range of fluorinated gases, including some HFCs and HFOs used in HVAC&R applications. In addition, trifluoroacetic acid (TFA), which is a PFAS, is a product of atmospheric degradation of HFO-1234yf and HFC-134a

Among the HFCs that have been identified as PFAS are: R32, R134a, R125, R143a and R152a. Some of the HFOs include R1234yf, R1234ze(E) and R1233zd(E). These HFCs and HFOs fall within the scope of the PFAS substances if they contain at least one CF2 (perfluorinated methyl group) or one CF3 (perfluorinated methyl group) in the molecular structure.

There is no certainty that this proposal will be accepted, but in any case it is advisable to develop technologies that do not include any type of refrigerant that could be banned in the coming years and thus nullify the efforts and investments used on that type of refrigerant.

Analysing the proposal, we can find a table where the restriction for the refrigerant is described. It's also describes the transition to the final usage of the PFAS refrigerants:

Use sector	Proposed derogation	Duration of derogation period	Cost impact of 5 and 12 year derogation periods
Refrigeration	Given the sufficiently strong evidence that technically and economically feasible alternatives are not available at EiF, derogations are proposed for:	Ban with a transition period of 18 months and a 5-year derogation for refrigerants in low temperature refrigeration below -50°C, because stakeholders indicated that low temperature refrigeration below -50 °C	Ban with a transition period of 18 months and a 5-year derogation: For low temperature refrigeration below -50°C, a 5-year derogation would permit a longer period for R&D

	- Refrigerants in low temperature refrigeration below -50°C - Refrigerants in laboratory test and measurement equipment - Refrigerants in refrigerated centrifuges	in large capacities is expected to still depend on fluorinated gases for 10 years from 2022 [sufficiently strong evidence base]. Ban with a transition period of 18 months and a 12-year derogation for refrigerants in laboratory test and measurement	and would reduce costs for producers whilst maintaining production rates and quality. This would also limit potential impacts on consumers and the risk of job losses. Ban with a transition period of 18 months and a 12-year derogation:

Table 2: PFAS restrictions

1.3 Usable fluids with GWP less than 150 conformed to the restriction

Propane (R290) is one of the refrigerants that has the ideal characteristics to be used in accordance with the proposed law. In fact, the R290 has a GWP of 3 that is well below the threshold of 150 set, which places it in the elite class of refrigerants with minimal environmental impact. Obviously, it is not the only refrigerant that can be used in these conditions, in fact we also find:

Refrigerant	Type	Chemical	GWP
R290	Natural	Propane(C_3H_8)	3
R717	Natural	Ammonia (NH_3)	0
R744	Natural	Carbon Dioxide (CO_2)	1
R1270	HC/Natural	Propylene (C_3H_6)	2
R600a	HC/Natural	Isobutane (C_4H_{10})	3
R1234yf	HFO	CF3CF=CH2	<1

Table 3: Low GWP refrigerant

1.3.1 R717

Ammonia is highly toxic with moderate flammability, it is easily available at low cost. It is mainly used in industrial refrigeration applications where I do not have the presence of people, sometimes it can be used in medium capacity air conditioning systems.

1.3.2 R744

Among these fluids, carbon dioxide is certainly the least flammable and most non-toxic fluid, easily available at almost no cost. It is used in applications of industrial refrigeration and heat pumps, excellent both for low temperature (e.g. refrigeration counters) and for high temperature (eg for heat pumps for domestic hot water production). In Europe, in fact, CO2 heat pumps are only 3% of the total heat pumps installed for domestic use, while in other countries such as Japan and China the use is growing with an annual production of about 10000 units. Negative properties include high working pressure, a negative effect on the human body, the leakage of the CO2 is almost no noticeable using the sensory organs. Working with CO2 requires training in industrial safety, but since R744 belongs to the same group as the widely used freons, this aspect is not given sufficient attention.

1.3.3 R1270

Propylene is highly flammable, easily available and an average cost of 3-5 Euro per kilogram value given by Nippon gases refrigerant. It can be used in industrial, commercial, cold storage, air conditioning and small/medium sized chiller applications.

1.3.4 R600a

The isobutane as the propylene is highly flammable, the cost is variable but it can be attested inside to 10 euros per kilogram. The main areas of use are for small size refrigeration systems because they are systems with low amount of refrigerant charge (a few decide grams) and therefore not dangerous and still used in completely isolated systems.

1.3.5 R1234yf

This type of refrigerant is slightly flammable in fact the proposed safety class is A2L, as availability and costs are extremely variable from country to country. It was introduced as a replacement for R134a for applications in car air conditioners having a GWP that meets the standard. Norway has proposed a revision of REACH (registration, evaluation, authorisation and restriction of chemicals) where a restriction is introduced to ban the use of PFAS substances, including HCF and HFO refrigerants. Among them the R1234yf fluid is included and therefore, if the proposal passed, it would no longer be usable in heat pumps.

The main characteristics of the different fluids have been briefly reported below:

Refrigerant	Safety Class ASHRAE 34-2016	Cost/ Availability	Main application	Issues
R717	B2	Low/High	Industrial application/ no people	High toxicity, low inflammable
R744	A1	Zero/High	Industrial application and HP	High pressure required
R1270	A3	Medium/Medium	Industrial and commercial refrigeration/ chiller	High inflammable
R600a	A3	Medium/Medium	Small refrigerant system	High inflammable
R1234yf	A2L	High/Medium	Car air conditioner	High inflammable, possible ban use from REACH

Table 4: Main characteristic of low GWP refrigerants

From the analysis conducted, it can be noted that each fluid has its own peculiarities of application and technologies more or less mature.

1.4 R290

Propane is a hydrocarbon (C_3H_8). The molecule is reported below:

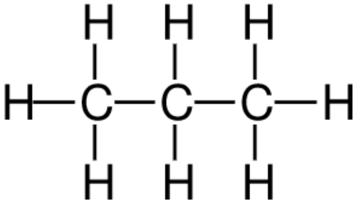


Figure 1: Propane molecule

It is a gas at standard temperature and pressure, but compressible to a transportable liquid. It can be classified as liquefied petroleum gas, LP-gas or LPG. Propane is produced from both natural gas processing and crude oil refining, in roughly equal amounts from each source. Is a nontoxic, colourless and odourless.

1.4.1 Pure fluid

Propane is a pure fluid, whereas most of alternative refrigerants are HFCs and HFOs. When the components of a mixture have different evaporation and condensation temperature, the mixture is zeotropic and the refrigerant has a value of glide. High value of glide influence the design and operation of the heat exchangers, using a pure fluid as refrigerant avoids all the inconveniences related to glide.

1.4.2 Low GWP

As we have already said propane is a natural fluid that has a GWP of three and ODP equal to zero, it is highly flammable, therefore it is classified as an A3.

1.4.3 Low cost

The low price of natural refrigerants is not the main reason to choose and use it, but the difference with respect to synthetic refrigerants is so big that this advantage must not to be ignored. Currently, the approximate price of propane in Italy (with no big differences in the rest of Europe) is 5€/kg. On the other hand, the prices of HFO refrigerant mixtures such as R455A or R454C is about 34 €/kg, whereas the prices of pure HFO components vary considerably among different substances. For example, the price of R-1234yf is about 77 €/kg. Considering the quantity of refrigerant used in all the units of one series, these differences of price can be definitively significant.

On this matter, another important aspect to take into account is the stability of prices. Indeed, the trend of prices during the last three years has shown that the only refrigerants with practically constant prices have been natural ones, such as propane.

1.4.4 Expected increase in its use

The field of applicability is extremely wide and expanding. It has been widely used domestic applications such as refrigerator and freezer, but has not been used often in medium and large heat pump applications. This is due to charging restrictions imposed by regulations. The safety systems required and the components needed to work with propane have always been more expensive. And also users and manufacturers are concerned about the flammability of hydrocarbon refrigerants. This characteristic is actually harmless if the refrigerant is used with the correct procedure. Hydrocarbons are flammable in the presence of two other compounds called the fire triangle: hydrocarbons, air and ignition. If one of the three factors is not available, there is no fire or explosion.

1.4.5 High efficiency

Propane's performance characteristic are similar to those of the now-outlawed R22, which was phased out because of its high ozone depletion potential. When compared with hydro chlorofluorocarbons (HCFCs) and hydrofluorocarbons (HFCs), propane will have a lower system pressure drop and a higher heat transfer performance. Since its thermodynamic properties are well suited to the temperatures typically encountered in building services engineering, the refrigeration cycle coefficient of performance (COP) is comparatively good.

A further study will be carried out in Chapter 3 where the three main fluids of interest of this thesis will be compared.

Chapter 2: Description of the units concerned

2.1 Introduction

We will begin this chapter by describing the general operation of a heat pump, and then move on to the units analysed in this thesis.

The units described in this chapter will be three:

- VMC air to air heat pump
- Scroll air to water heat pump
- Screw air to water heat pump

Each of which will have a different field of applicability. Below will be described the characteristic of each unit. These heat pumps currently are designed to work with a refrigerant fluid different from propane. These units are the starting point from which we then went to size those to R290. In the following chapters, comparisons will be made to see the operating differences analytically.

2.2 Heat pump general scheme

The energy, both thermal and electrical, used in residential and commercial application for space heating, conditioning and domestic water, represents a large part of the total energy consumed. Almost all of this energy is produced by fuels, liquids and gases, with which causes damage to the environment and harmful effects on human health. The new requirements linked to the improved standard of living require the use of air conditioning not only in the workplace but also in the house, with consequent increases in energy consumption. The heat pump represents a means to improve the level of comfort of living and working environments.

The unit consists of a closed circuit, run by a specific fluid (refrigerant) that, depending on the temperature and pressure conditions in which it is located, assumes the state of liquid or steam. The closed circuit consists of:

- Compressor
- Condenser
- Expansion device
- Evaporator

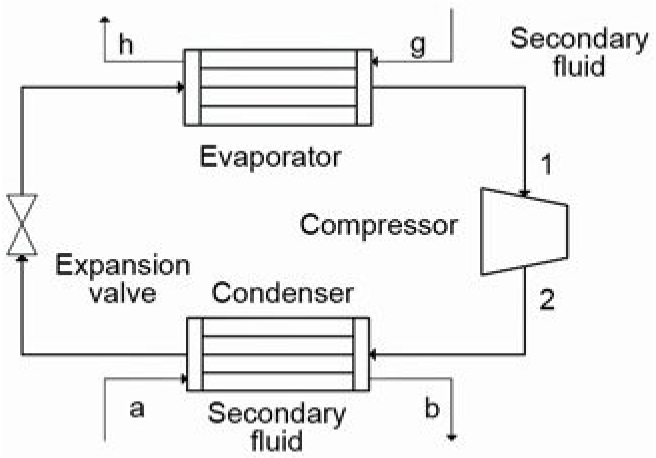


Figure 2: Basic heat pump scheme

The condenser and evaporator consist of heat exchangers, that is, tubes or plate placed in contact with a service fluid (which can be water or air) in which the refrigerant flows. This gives heat to the condenser and subtracts it to the evaporator. In operation, the refrigerant inside the circuit undergoes the following transformations:

- Compression: the refrigerant at low pressure and in the gas phase, coming from the evaporator, is carried at high pressure; in the compression, it is heated by absorbing a certain amount of heat.
- Condensation: the refrigerant, coming from the compressor, passes from the gaseous state to the liquid one yielding heat to the outside
- Expansion: by passing through the expansion valve, the liquid refrigerant is partially transformed into steam and cools
- Evaporation: the cooling fluid absorbs heat from the outside and evaporates completely.

The combination of these transformations constitutes the heat pump cycle: by supplying energy with the compressor, to the refrigerant, this, in the evaporator, absorbs heat from the surrounding medium and, through the condenser, yields it to the medium to be heated.

The efficiency of a heat pump is measured by its Coefficient of Performance (COP), defined as the ratio of thermal energy supplied to the heat transfer fluid (air or water) and electricity absorbed. A COP (for example) equal to 3 means that for each kWh of electricity consumed, the heat pump will make 3 kWh of thermal energy to the environment to be heated.

The possible availability of electricity from renewable sources makes the use of the heat pump particularly interesting in terms of energy saving and respect for the environment. The main sources of low temperature thermal energy required for its operation are air, water and ground (geothermal heat pump). The fluid to be heated can be directly the indoor air (direct expansion system) or water that subsequently distributes the heat to the environment (hydronic system) through appropriate local terminals.

2.3 VMC heat pump (A-A)

2.3.1 Description

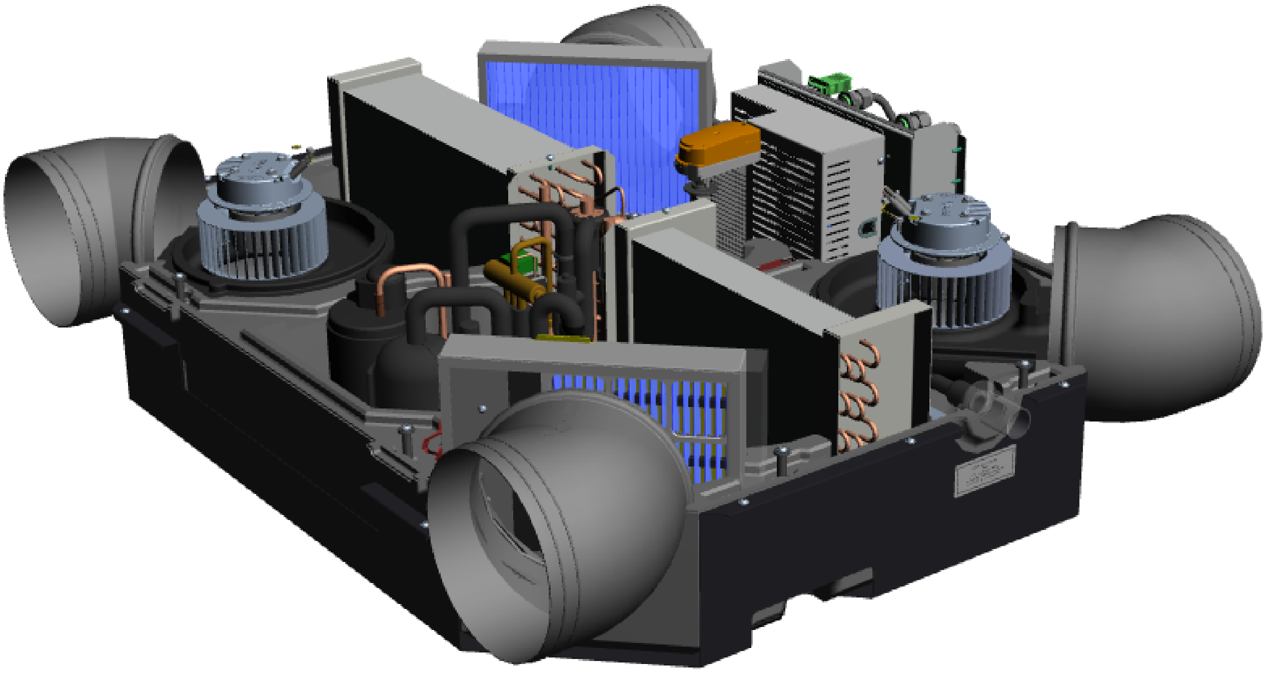


Figure 3: VMC heat pump (A-A)

The VMC heat pump (A-A) is a heat pump with an air renewal and purification system, with active thermodynamic recovery and R32 refrigerant, ideal for homes and offices from 90 to 250 m^2. It is usually inserted in false ceilings having a height of only 290mm.

The three typical operating cases are:

- Winter season: VMC heat pump (A-A) recovers the heat from the exhaust air to heat the incoming air. For most of the winter, it provides most of the building's energy needs.

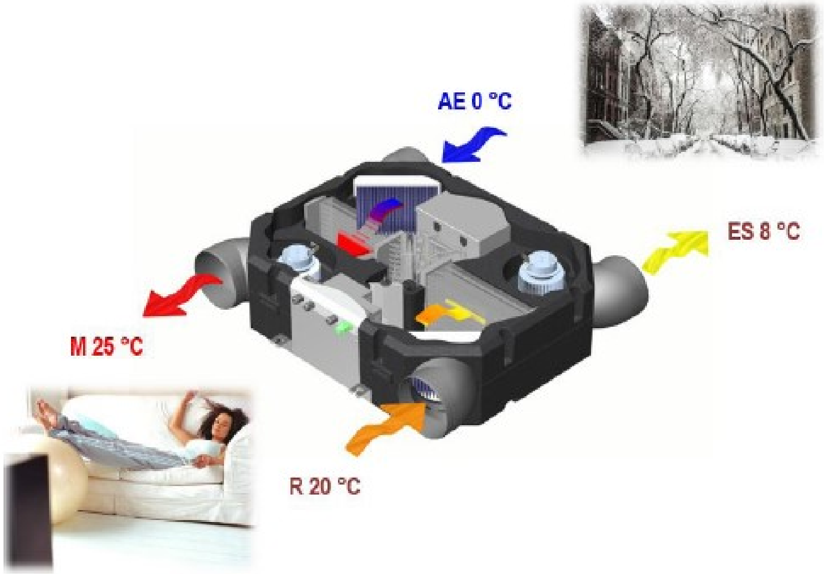


Figure 4: Winter operation VMC heat pump

- Summer season: VMC heat pump (A-A) cools the air entering in the environment by transferring heat to the expelled air. Even in the summer it reduces the use of the main air conditioning system. In cooling operation controls the humidity in the inlet air.

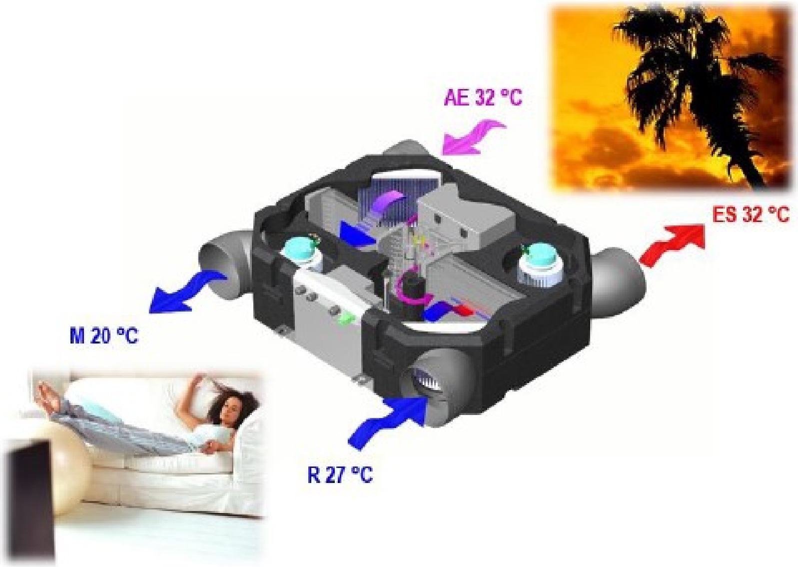


Figure 5: Summer operation VMC heat pump

- Half seasons: VMC heat pump (A-A) introduce external air in the environment without heating or cooling (free-cooling). The units can fully replace the air conditioning systems.

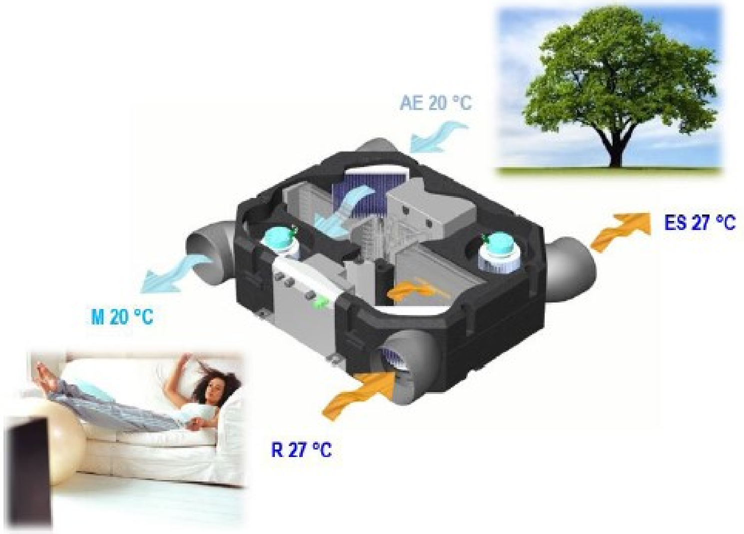


Figure 6: Half season operation VMC heat pmup

A. Inlet air filter
B. Outlet air filter
C. Inlet fan
D. Outlet fan
E. Rotary compressor with inverter
F. Renewal exchanger
G. Ejection exchanger
H. Adjustable air connections
I. Electrical panel

2.3.2 Functional Scheme

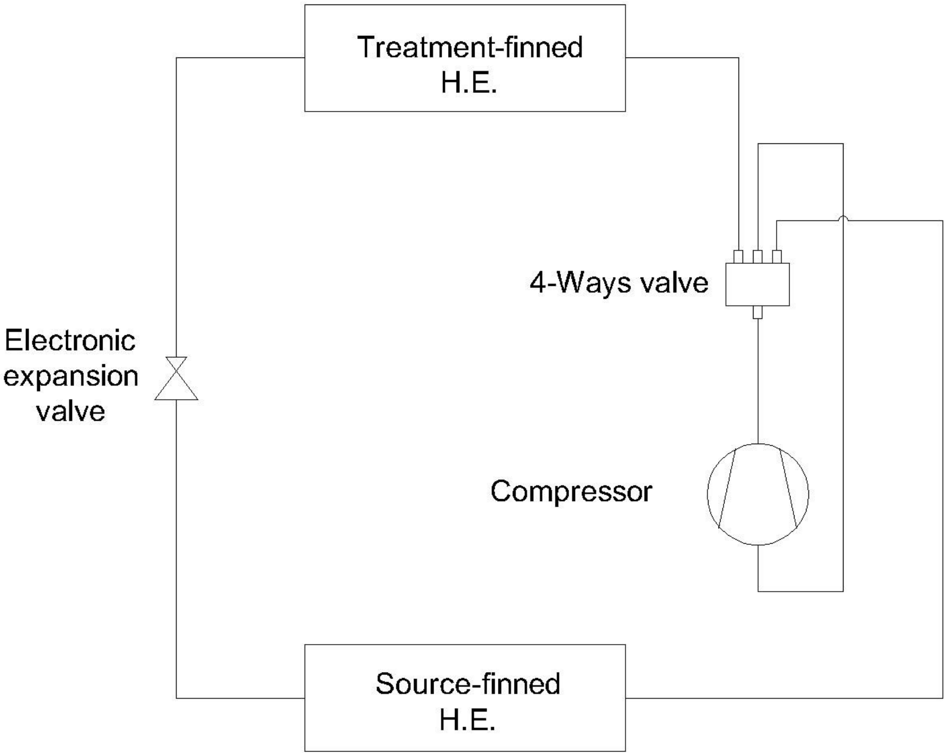


Figure 7: Simplified VMC scheme

In heating operation, the cycle is run clockwise, that is:

- The refrigerant enter the compressor where the compression of the fluid occurs with the respective pressure increase
- It enter the 4-way valve, which is switched so that the fluid is sent into the treatment finned coil which, in the diagram above, is on the left. Here the fluid releases heat to the incoming air from the outside by heating it to the desired temperature.
- The fluid is in the liquid phase and is sent to the electronic expansion valve where it undergoes a pressure reduction until it reaches the evaporation pressure conditions.
- The fluid enters the source finned coil where it evaporates, absorbing heat from the exhaust air coming out of the internal environment, passes through the 4-way valve and is sent to the compressor.

In the cooling conditions, a cycle reversal takes place: the 4-way valve is switched in such a way that, after the compression, the refrigerant fluid is sent into the source finned coil where the condensation takes place, giving heat to the air that is in the exhaust conditions and the cycle is then run in the opposite direction to the previous one. The dampers are used in particular cooling conditions where to reduce the T_cond, bypassing a part of the inlet flow to increase the flow of air to the condensing exchanger and therefore to have lower T_cond.

In free-cooling conditions, the compressor is switched off, the air is filtered at the inlet and is sent into the internal environment thanks to the supply fan.

2.3.3 Performance

Supply airflow	m^3/h	125	150	210	270	320
A7						
Heating capacity	kW	1,42	1,55	1,86	2.05	2,49
Total power input	kW	0,46	0,42	0,45	0,42	0,54
COP	-	3,09	3,69	4,13	4,93	4,61
A5						
Heating capacity	kW	1,97	2,10	2,21	2,37	2,45
Total power input	kW	0,40	0,52	0,47	0,37	0,32
COP	-	4,93	4,04	4,70	6,50	7,66
A30						
Cooling capacity	kW	0,92	1,38	1,47	1,72	2,07
Total power input	kW	0,36	0,52	0,48	0,54	0,81
EER	-	2,56	2,65	3,06	3,21	2,56
A35						
Cooling capacity	kW	1,57	1,64	1,73	1,92	2,23
Total power input	kW	0,36	0,52	0,53	0,55	0,81
EER		4,34	3,15	3,26	3,50	2,77
Rated static pressure supply fan	Pa	50	50	50	50	50
Max static pressure supply fan	Pa	120	120	120	120	120
Min inlet air temperature (D.B.)	°C	-15	-15	-15	-15	-15
Sound pressure level	db(A)	34	35	37	41	45

Table 5: Performance VMC heat pump

Where:

- A7 External air temperature7°C D.B/6°C W.B., Extracted air temperature 20°C D.B./13.7°CW.B.
- A5 External air temperature -5°C D.B/-5.4°C W.B., Extracted air temperature 20°C D.B./13.7°CW.B.
- A30 External air temperature 30°C D.B/22°C W.B., Extracted air temperature 27°C D.B./19°CW.B.
- A35 External air temperature 35°C D.B/24°C W.B., Extracted air temperature 27°C D.B./19°CW.B.

2.3.4 Constructive characteristics

Compressor		
Type of compressors	-	ROT
Refrigerant	-	R32
Number of compressors	Nr	1
Capacity scale	%	20-100
Oil charge	l	0,017
Refrigerant charge	kg	0,3
Refrigerant circuits	Nr	1
Treatment Area Fans (Supply)		
Type of fans		CFG
Number of fans		1
Fan diameter	mm	140
Type of motor	m^3/h	EC
Airflow		125-320
Treatment Area Fans (Ripresa)		
Type of fans		CFG

Number of fans		1
Fan diameter	mm	140
Type of motor		EC
Airflow		125-400
Connections		
Condensate drain	mm	32

Table 6: Contructive characteristic VMC heat pump

- ROT= rotary compressor

- CFG= centrifuge fan

- EC= EC Electronic switching motor

2.4 Scroll heat pump (A-W)

The Scroll heat pump (A-W) is available in various versions related to cooling and thermal capacity:

Size		18.2	20.1	25.2	30.2	35.2
Cooling capacity	kW	53,3	58,9	72,0	77,7	85,0
Heating capacity	kW	53,0	66,0	79,3	84,7	91,0

Table 7: Available size scroll heat pump

2.4.1 Description

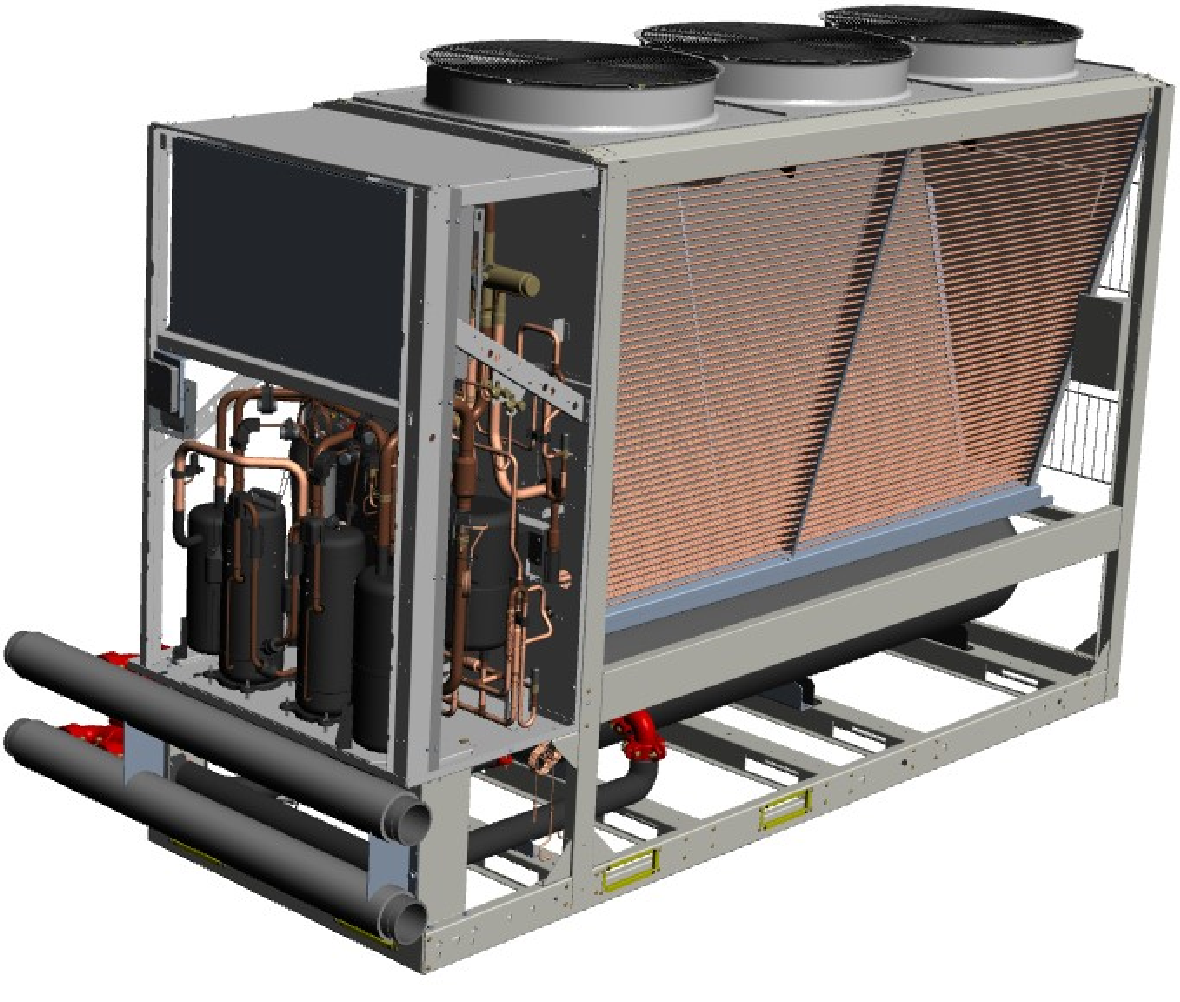


Figure 8: Scroll heat pump R32

The Scroll heat pump (A-W) is the an air condensed heat pump, equipped with Full DC Inverter technology and R32 refrigerant, for outdoor installation is able to manage and produce domestic hot water up to 55°C.

The unit is composed as follows:

- Compressor: it is a hermetic compressor Scroll with steam injection controlled by inverter. The unit consists of two compressors connected in tandem on a single cooling circuit and have a dedicated system for oil recovery.
- Treatment exchanger: is a direct expansion exchanger of the brazed stainless steel plate type, in a package without gaskets using copper as a brazing material.
- Source heat exchanger: it is a direct expansion fin&tube heat exchanger, made with copper tubes arranged on staggered and mechanically expanded rows to better adhere to the collar of the fins. The fins are made of aluminium with hydrophilic coating that allows the correct evacuation of condensate water and optimizes defrosting. The fins have a special corrugated surface and are adequately spaced to ensure maximum heat exchange efficiency.
- Fan: axial fans with four profiled blades in plastic material, directly coupled to the DC motor type "brushless" electronically controlled.
- 4-way cycle reversal valve: it is used for the transition from winter to summer operation by reversing the cycle and also for the defrosting cycle.
- Liquid receiver: is a device useful to contain the changes in refrigerant charge that occur between the two types of operation.
- Electronic expansion valve: it is a device capable of bringing the refrigerant from the condensation pressure to that of evaporation. Usually the refrigerant upstream of the valve is in a liquid state.
- Oil separator: Device useful for the separation of oil in the circuit, usually placed after the compressors.
- Suction liquid separator: device useful for the separation of the residual drops of liquid trapped in the vapour aspiration to the compressors. It is used mainly during defrosting and during transitional periods
- Heat exchanger economizer: Useful for heat recovery, is always a plate heat exchanger where the heat is recovered and used to pre-heating a fluid with some energy that, in other ways, would be lost.

2.4.2 Functional scheme

A simplified functional scheme for this unit is reported below:

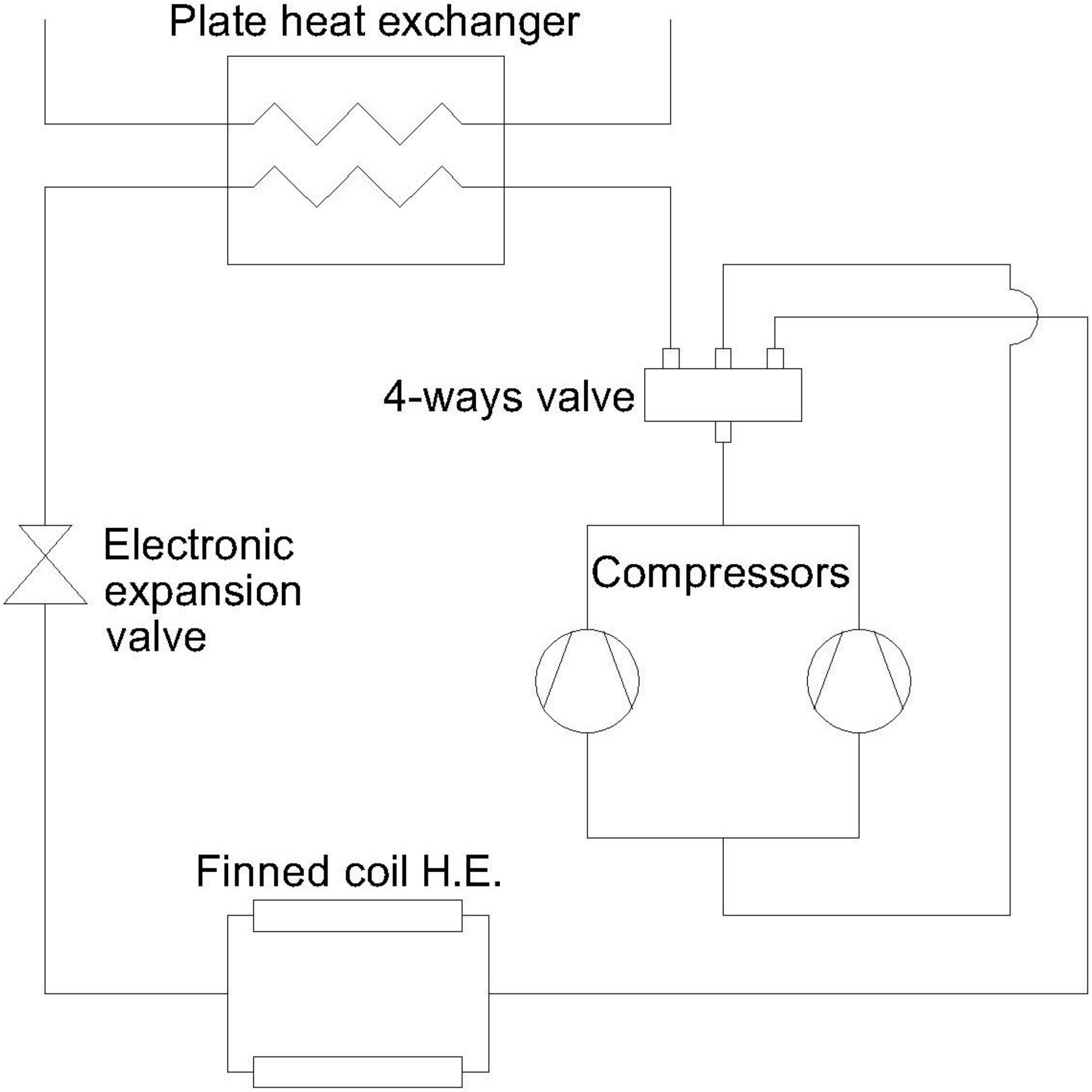


Figure 9: Simplified scroll heat pump scheme

The two compressors that are connected in parallel are responsible for compressing the refrigerant, and the 4-way valve is switched depending on the type of operation desired (heating or cooling).

During the heating process, the fluid passes through a plate heat exchanger, where it releases useful heat for heating eater. It is then condensed and directed to the finned coil heat exchangers, where it absorbs heat from the surrounding air.

On the other hand, during the cooling process, the 4-way valve s switched, and the refrigerant cycle runs in the opposite

2.4.3 Performance

Radiant pannels		Units	Value
Heating			
Heating capacity	1/8	kW	87,7
COP	2	-	4,15
SCOP	9	-	4,06
$\eta_{s,h}$	12	%	159
Cooling			
Cooling capacity	5/8	kW	107
EER	6	-	4,04
Water flowrate	5	l/s	4,80
Exchanger pressure drop side use	5	kPa	58,8
Terminal units			
Heating			
Heating capacity	3	kW	91,2
COP	2	-	3,25
SCOP		-	
$\eta_{s,h}$	7	kW	85,3
Cooling	6		2,91

Cooling capacity	9		4,50
EER	13	%	117
Water flowrate	7	l/s	4,12
Exchanger pressure drop side use	7	kPa	43,9
Radiatori			
Heating			
Heating capacity	4	kW	92,6
COP	2	-	2,53
Water flowrate	4	l/s	4,16
Exchanger pressure drop side use	4	kPa	44,7
AHRI data			
Cooling capacity	11	kW	84,6
Total power input	11	kW	28,9
COP	11	-	2,93
IPLV	11	-	4,79

Table 8: Performance scroll heat pump

1. Inlet/outlet water temperature on use side 30/35°C, air entering the external heat exchanger 7°C (U.R. = 85%)

2. COP (EN 14511:2018) heating performance coefficient. Ratio of heat output to power consumption according to EN 14511:2018.

3. Inlet/outlet water temperature on the use side 40/45°C, air entering the external heat exchanger 7°C (U.R. = 85%)

4. Inlet/outlet water temperature on the use side 50/55°C, air entering the external heat exchanger 7°C (U.R. = 85%)

5. Inlet/outlet water temperature on use side 23/18°C, air entering the external heat exchanger 35°C

6. EER (EN 14511:2018) cooling performance coefficient. Ratio of the cooling capacity to the power consumption according to EN 14511:2018.

7. Inlet/outlet water temperature on the use side 12/7 °C, air entering the external heat exchanger 35°C

8. Data for operating units with inverter frequency optimized for this application

9. Data calculated in accordance with EN 14825:2016

10. Seasonal Energy Efficiency Class of Space Heating according to Commission Delegated Regulation (EU) No. 811/2013, W = Water Outlet Temperature (°C)

11. Data calculated in accordance with AHRI 550/590 under the following conditions: Internal water exchanger 6,7°C. Air entering the exchanger external 35°C. Evaporator incrustation factor = 0.18 x10 (-4) m2 K/W

12. Seasonal energy efficiency in heating EN 14825:2018.

13. Seasonal energy efficiency in cooling EN 14825:2018.

2.4.4 Constructive characteristics

Compressor			
Compressor type			Scroll Inverter
Refrigerant			R32
No. Compressor		Nr	2
Oil charge		L	6
Refrigerant charge		kg	21,0
No. Circuits		Nr	1
User side exchanger			
Type of internal exchanger	1		BPHE
Water content		l	7,8
External Section Fans			
Fans type			Brushless DC Motor

No. fans		Nr	3
Standard air flow		l/s	10333
Installed unit power		kW	0,9
Water circuit			
Maximum water side pressure		MPa	1
Minimum circuit water volume in heating	2	l	620
Minimum circuit water volume in cooling	3	l	200
Total internal water volume		l	8,0
Power supply			
Standard power supply			400/3N/50

Table 9: Constructive characteristics scroll heat pump

1. BPHE = plate heat exchanger

2. Entering/leaving water temperature user side 25/30 °C, external exchanger entering air 2°C (U.R. = 85%) - Minimum water volume that does not consider the volume of water inside the unit.

3. Entering/leaving water temperature user side 15/10 °C, external exchanger entering air 25°C (U.R. = 85%) - Minimum water volume that does not consider the volume of water inside the unit.

2.5 Screw heat pump (A-W)

The Screw heat pump (A-W) is available in various versions related to cooling and thermal capacity:

Size	220.2	240.2	260.2	280.2	320.2	340.2	420.2
Cooling capacity [kW]	523	545	575	634	722	792	990
Heating capacity [kW]	503	508	537	631	697	776	907

Table 10: Available size screw heat pump

Figure 10: Screw heat pump R513A

2.5.1 Description

The Screw heat pump (A-W) unit is a heat pump with air-cooled inverter technology with simultaneous hot/cold production for outdoor installation. As we can see it comes in various versions that operate with different refrigerants, in particular it will be analysed version 280.2 that operates with R513a.

The unit is composed as follows:

- Compressor: compact semi-hermetic compressors with double helical screw with integrated oil separator. Starting with limited power consumption is achieved by progressively accelerating the compressor with the inverter. The power supply of the electric motor and inverter is three-phase. At the exhaust of the compressor is provided a non-return valve to prevent the counter rotation in stop.

- Cold Side Heat Exchanger: Heat exchanger of the tube type independent refrigerant side for each compressor. The exchanger consists of a mantle made of carbon steel. The tubes, anchored to the tube plate by mechanical spindle, are made of copper, internally lined to optimize the heat exchange.
- Heat exchanger: independent heat exchanger for each compressor. The exchanger consists of a mantle made of carbon steel. The tubes, anchored to the tube plate by mechanical spindle, are made of copper, internally lined to optimize the heat exchange.
- External air exchanger: finned pack heat exchanger, made with copper tubes arranged on staggered and mechanically expanded rows to better adhere to the collar of the fins. Hydrophilic aluminium fins adequately spaced to ensure maximum heat exchange efficiency.
- Fan: high performance helical fans, with PP-coated aluminium plate blades, three-phase electric motor directly coupled to the external rotor.
- 4-way cycle reversing valve:
- Electronic expansion valve:
- Suction liquid separator

2.5.2 Functional scheme

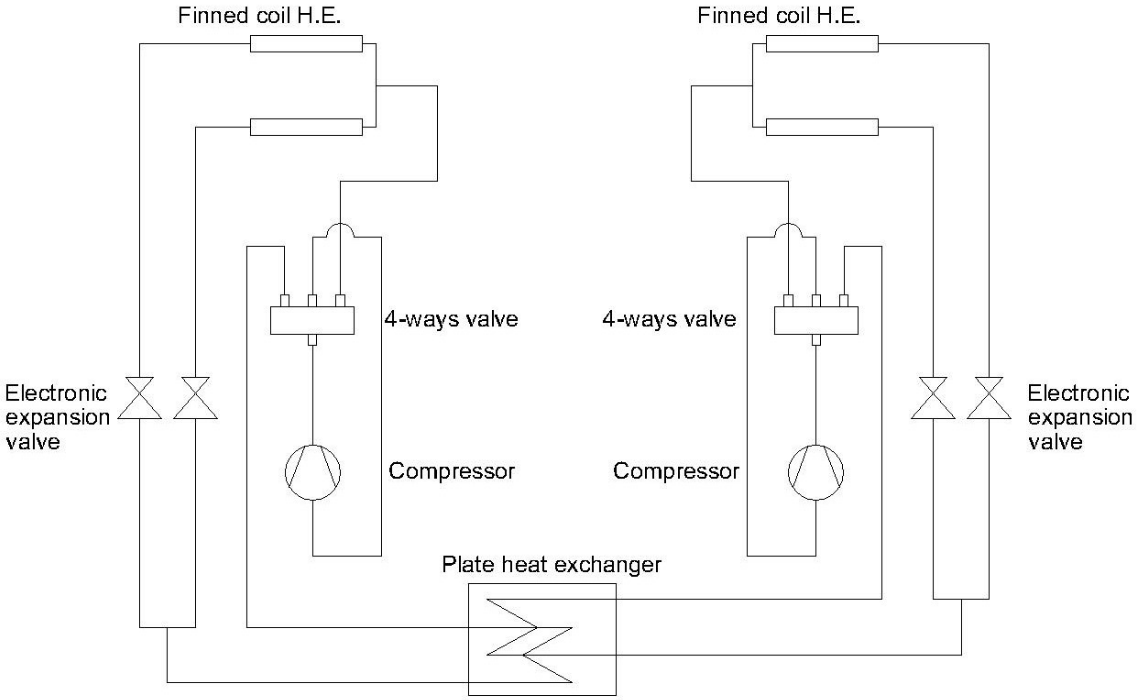


Figure 11: Simplified screw heat pump scheme

2.5.3 Performance

Cooling 100% - Heating 0%			
Cooling capacity	1	kW	634
Compressor power input	1	kW	185
Total power input	2	kW	204
EER	1	-	3,11
Water flowrate cold side	1	l/s	30,3
Cold side pressure drops	1	kPa	59
Cooling capacity (EN14511:2018)	3	kW	634
Total power input (EN14511:2018)	3	kW	206
EER	3	-	3,07
SEER	6	-	5,17
SEPR	6	-	5,80
Cooling capacity (AHRI 550/590)	4	kW	628
Total power input (AHRI 550/590)	4	kW	203
COP	4	-	3,09
IPLV	4	-	5,36
Cooling 0% - Heating 100%			
Heating capacity	7	kW	631
Compressor power input	7	kW	184
Total power input	2	kW	203
COP	7	-	3,11
Water flowrate hot side	7	l/s	30,2
Hot side pressure drops	7	kPa	36

Heating capacity (EN14511:2018)	8	kW	632
Total power input (EN14511:2018)	8	kW	205
COP	8	-	3,09
SCOP	5	-	-
Cooling 100% - Heating 100%			
Cooling capacity (EN14511:2018)	9	kW	633
Heating capacity (EN14511:2018)	9	kW	805
Total power input (EN14511:2018)	9	kW	192
TER (EN14511:2018)	10	-	7,48
Water flowrate cold side	9	l/s	30,3
Cold side pressure drops	9	kPa	59
Water flowrate hot side	9	l/s	38,4
Hot side pressure drops	9	kPa	57

Table 11: Performance screw heat pump

The Product is compliant with the Erp (Energy Related Products) European Directive, It includes the Commission delegated Regulation (UE) N. 813/2013 Commission (nominal heating capacity ≤400 kW at specified reference conditions) and the Commission delegated Regulation (EU) No 2016/2281, also known as Ecodesign LOT21

Contains fluorinated greenhouse gases (GWP 631)

1. Data referred to the following conditions: internal exchanger water temperature = 12/7 °C. Entering external exchanger air temperature = 35°C. Evaporator fouling factor = 0.44 x 10^(-4) m2 K/W.

2. The Total Power Input value does not take into account the part related to the pumps and required to overcome the pressure drops for the circulation of the solution inside the exchangers.

3. Data compliant to Standard EN 14511:2018 referred to the following conditions: Cold side exchanger water temperature = 12/7 °C. Entering external exchanger air temperature = 35°C

4. Data compliant to Standard AHRI 550/590 referred to the following conditions: Cold side exchanger water temperature = 6,7°C. Water flow-rate 0,043 l/s per kW. Entering external exchanger air temperature = 35°C. Evaporator fouling factor = 0.18 x 10^(-4) m2 K/W

5. Data compliant according to EU regulation 813/2013

6. Data compliant according to EU regulation 2016/2281

7. Data referred to the following conditions: Hot side exchanger water temperature = 40/45 °C. Entering external exchanger air temperature = 7°C D.B./6°C W.B. Evaporator fouling factor = 0.44 x 10^(-4) m² K/W

8. Data compliant to Standard EN 14825:2018 referred to the following conditions: Hot side exchanger water temperature = 40/45 °C. Entering external exchanger air temperature = 7°C D.B./6°C W.B.

9. Data compliant to Standard EN 14511:2018: Cold side exchanger water temperature = 12/7 °C. Hot side exchanger water temperature = 40/45 °C Evaporator fouling factor = 0.44 x 10^(-4) m² K/W

10. TER = (Cooling capacity + Heating capacity) / Total power input

2.5.4 Constructive characteristics

Compressor			
Type of compressors	1		ISW
Refrigerant			R513A
No. compressor		Nr	2
Rated power(C1)		HP	125
Rated power(C2)		HP	125
Std Capacity control steps			STEPLESS
Oil charge (C1)		l	18
Oil charge (C2)		l	18
Refrigerant charge(C1)		kg	174
Refrigerant charge(C2)		kg	174
Refrigeration circuits		Nr	2
Cold side exchanger			
Type of internal exchanger	2		S&T

No. of internal exchanger		Nr	1
Water content		l	496
Hot side exchanger			
Type of internal exchanger	2		S&T
No. of internal exchanger		Nr	2
Water content		l	62
External exchanger			
Type of external exchanger	3		CCHY
Number of coils		Nr	4
External section fans			
Type of fans	4		AX
Number of fans		Nr	14
Type of motor	5		EC
Standard airflow		l/s	78724
Connections			
Cold side water fittings			6’’
Hot side water fittings			6’’

Table 12: Constructive characteristic screw heat pump

Chapter 3: Comparative analysis of R32, R513A and R290

3.1 Introduction

This chapter describes the main characteristics of the three cooling fluids under consideration.

Afterwards, the behaviour of the three units' native refrigerants will be compared with the behaviour of propane under the same operating conditions.

3.1.1 Properties of Refrigerants R32, R513A and R290

Selection of a refrigerant is a complex process involving detailed analysis of environmental, thermos-physical and safety properties.

3.1.2 Environmental properties

Refrigerant	Atmospheric Life in years	GWP	ODP
R32	<5	675	0
R513A	14	631	0
R290	0.041	3	0

Table 13: Environmetal properties of R32,R513A and R290

Ozone depletion potential (ODP) and global warming potential (GWP) and atmospheric life are the significant factors demonstrating environmental impact of refrigerant when released to the surroundings. As we can see from the table, all three fluids have zero ODP, R290 have the lowest value of GWP and is under 150 that is the maximum limit fixed for the future application. The atmospheric lifetime is an indicator of the average existence of the refrigerant released into the atmosphere, until it decomposes: R513a have a high lifetime, while R290 have the lowest. R290 has very low or no direct effect on global heating and indirect effect also expected to be lower of its excellent thermo-physical properties.

3.1.3 Physical and thermophysical properties

Refrigerant	Molecular weight	Normal boiling point	Critical temperature	Critical pressure	Latent heat of evaporation
	[kg/kmol]	[°C]	[°C]	[MPa]	[kJ/kg]
R32	52.02	-51.65	78.10	5.782	299
R513A	108.40	-29.20	96.50	3.766	194.8
R290	44.09	-42.11	96.74	4.251	425.4

Table 14: Physical and therophysical properties of R32, R513A and R290

Latent heat of R290 is higher than R32 by 42% and R513A of 118%. The higher latent heat of evaporation indicate a lower refrigerant mass requirement with equal circuit and output power. Thermo-physical proprieties of refrigerant determine the energy performance of the refrigeration system. The table below shows thermos-physical properties of the refrigerant R322, R513A and R290, at evaporating temperature of 10°C and condensing temperature of 45°C.

Property	Temperature [°C]	State	R32	R513A	R290
Saturation pressure [MPa]	10	Liquid	1.107	0.453	0.637
	45	Vapour	2.795	1.218	1.534
Density [kg/m^3]	10	Liquid	1019.7	1188.4	514.73
	45	Vapour	84.86	66.318	34.15
Viscosity [µPa-s]	10	Liquid	134.63	119.15	113.55
	45	Vapour	14.243	12.577	8.963
Thermal conductivity [W/mK]	10	Liquid	0.137	0.075	0.101
	45	Vapour	0.021	0.016	0.022
Specific heat [kJ/kgK]	10	Liquid	1.806	1.349	2.573
	45	Vapour	2.206	1.229	2.371

Table 15: Properties at 10 - 45°C for R32, R513A and R290

The lower liquid density of R290 reflects the lower requirement of refrigerant mass resulting in lower friction and better heat transfer coefficients in evaporator and condenser. Refrigerant viscosity is the major source of irreversibility and influences condensation and boiling heat transfer coefficients. R290 has a good combination of low viscosity and high thermal conductivity, which improves the performance of condenser and evaporator. The higher specific heat of R290 gives lower discharge temperature.

3.1.4 Safety characteristics

According to ASHRAE standard, R290 is classified as A3 class refrigerant, which means that, is a nontoxic and highly flammable refrigerant. R32 is classified as A2L and 513A as A1.

The toxicity of a refrigerant depends on effects related to a precise reference concentration (400ppm). Group A indicates non-toxic refrigerants for contractions of less than 400 ppm, group B refers to refrigerants that are toxic for contractions of less than 400 ppm.

The flammability of a refrigerant depends on the lower limit of flammability and the heat of combustion related to a condition of 60 °C and 1 atm. Class 1 indicates refrigerants that have no flame propagation, class 2 indicates moderately flammable refrigerants with a limit of flammability lower than 0,10 kg/m^3 and combustion heat lower than 19,000 kJ/kg. If the flame propagation is less than 10 cm/s the class becomes 2L and class 3 indicates highly flammable refrigerants with a lower flammability limit of more than 0,10 kg/m^3 and a combustion heat of less than 19,000 kJ/kg.

Below are the value of lower flammability limit by mass and volume, combustion heat, ignition temperature, ignition energy and propane toxicity.

LFL by mass	LFL by volume	Heat of combustion	Combustion velocity	Ignition temperature	Ignition energy	Toxicity	Safety class
kg/m^3	%	kJ/kg	m/s	°C	mJ	ppm	/
0,075	2,1	50500	0,4	466	0,25	1000	A3

Table 16: Safety parameter of R290

Flammability of R290 is a major concern. However, it should be remembered that it does not combust spontaneously in contact with air. Necessary preconditions for any accidents would be the release and the mixing of the refrigerant with air in correct proportion (1% to 10%) and the presence of an ignition source with energy greater than 2.5 x 10^{-4}kJ or a surface with a temperature higher than 440°C. Safe use of R290 is quite possible with few precautions.

3.2 Comparative analysis of refrigerant performance

A theoretical investigation of the performance of the vapour compression cycle with refrigerant R290, R32 and R513A was carried out. Performance parameter such as pressure ratio, discharge temperature, refrigerant mass flow rate, volumetric refrigeration capacity and COP were investigated for various temperature ranging between -30°C to 15°C, at a constant condensing temperature of 45°C.

3.2.1 Simulation conditions

For the performance analysis of refrigerant R290, R32 and R513A, a standard vapour compression cycle with a cooling capacity of 1 TR was considered where TR means ton of refrigerant. Remember that 1 TR is equal to 3,517 kW.

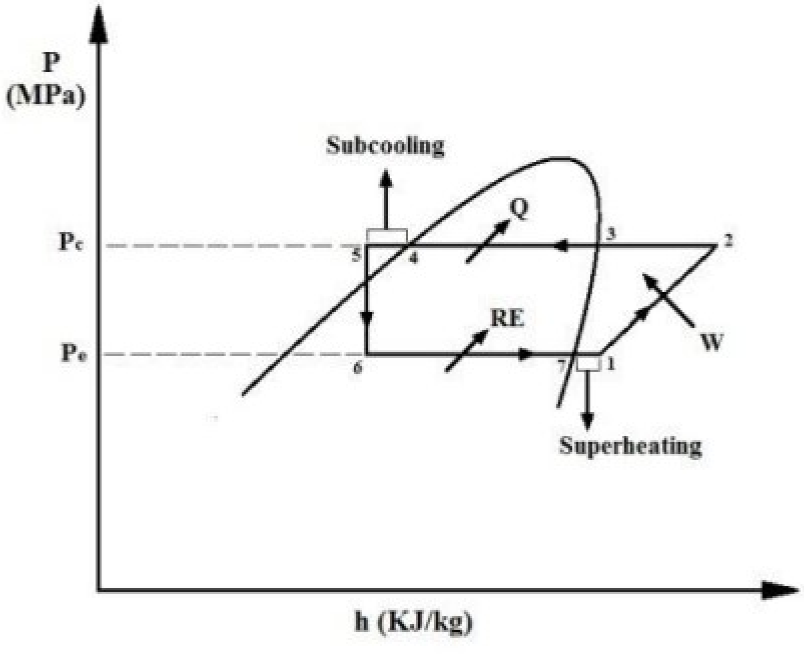


Figure 12: Reference cycle for simulation

Important assumptions made were:

- Negligible pressure drop and negligible heat loss to the surrounding
- Isentropic compression and isenthalpic expansion
- Superheating of 5°C and subcooling of 5°C.

Properties of refrigerants were calculated using REFPOP.

Thermodynamic properties at each state were determined and performance parameters were determine by using well-known equation.

$$Pressure\ ratio = \frac{\mathrm{p}_2}{p_1}$$

$$Isoentropic\ compression\ work, P_{comp} = h_2 - h_1$$

$$Refrigerating\ effect\ (RE) = h_1 - h_6$$

$$Power\ per\ ton\ of\ refrigeration = 3.5\frac{P_{comp}}{RE}$$

$$Volumetric\ refrigeration\ capacity = d\ RE$$

$$Coefficient\ of\ performance\ (COP) = \frac{RE}{P_{comp}}$$

3.3 Results of analysis

Applying the different equation for the refrigeration cycle under predetermine condition of operation and using REFPROP, performance parameters for R290, R32 and R513A like pressure ratio, discharge temperature, volumetric refrigeration capacity, refrigerant mass flow rate power and COP were obtained and plotted against evaporating temperature as shown in the figures below.

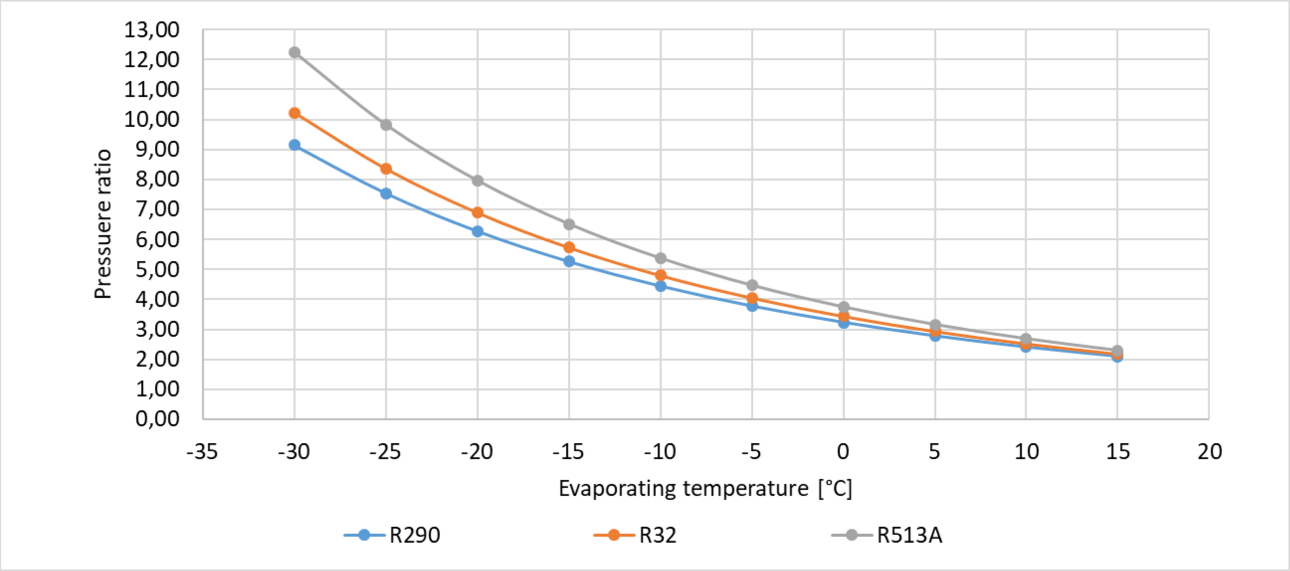


Figure 13: Pressure - evaporating temperature comparison R32, R513A and R290

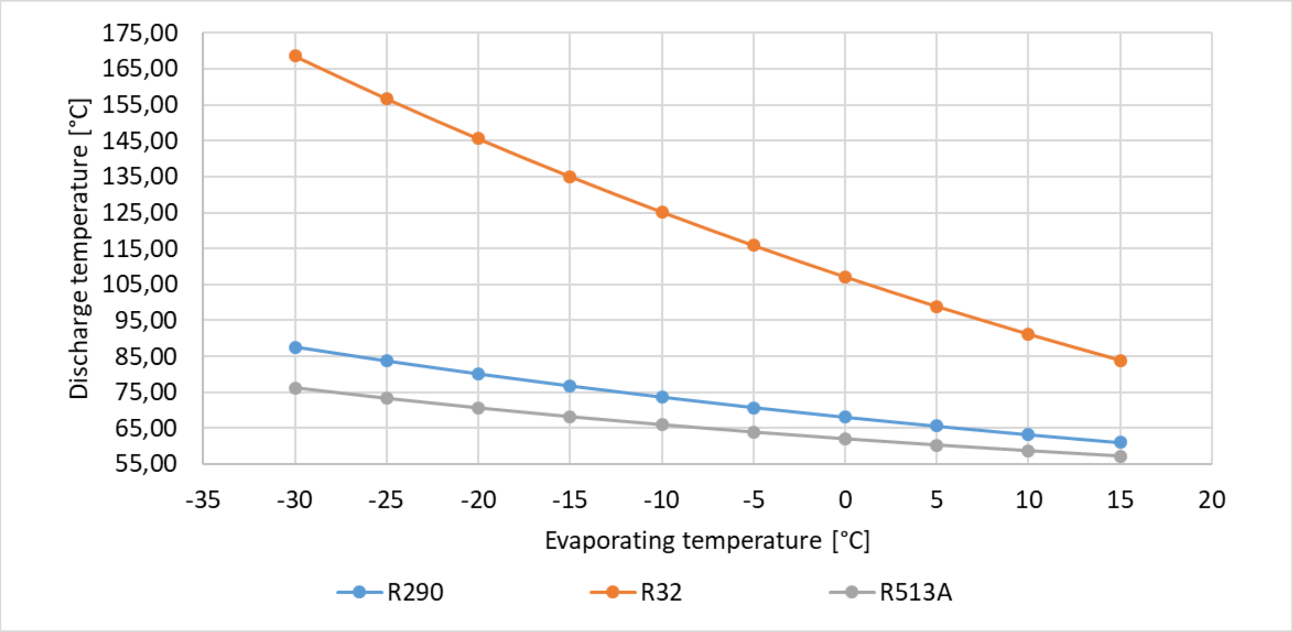


Figure 14: Discharge temperature - evaporating temperature comparison R32, R513A and R290

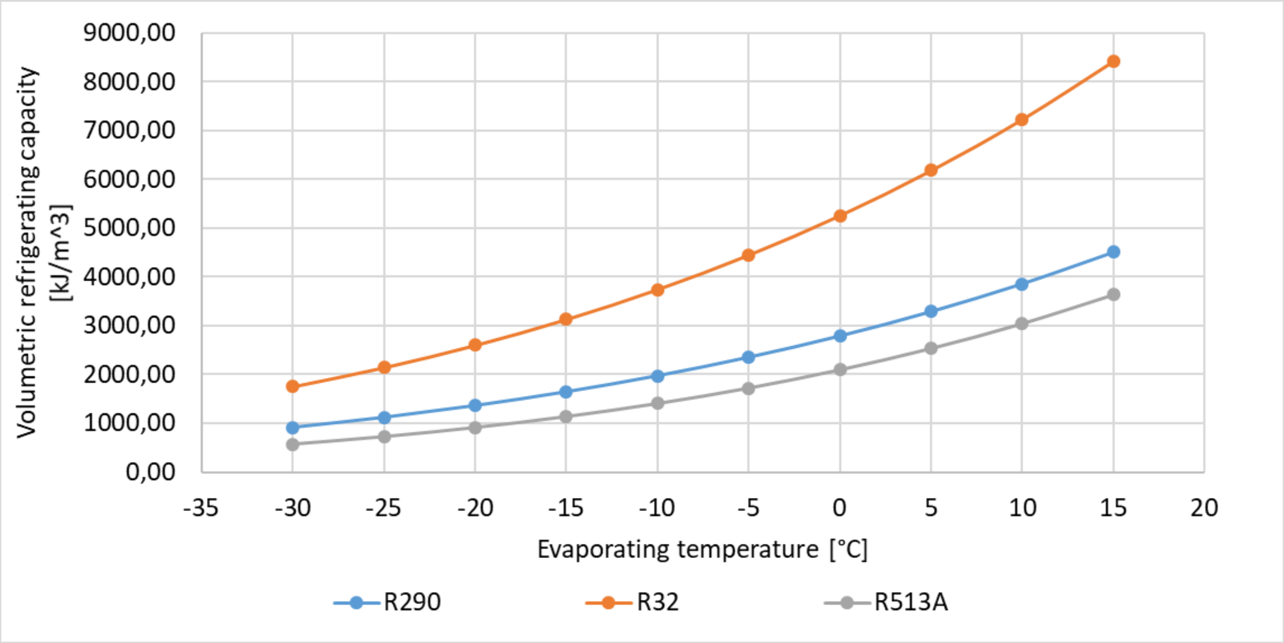


Figure 15: Volumetric refrigerating capacity - evaporating temperature comparison R32, R513A and R290

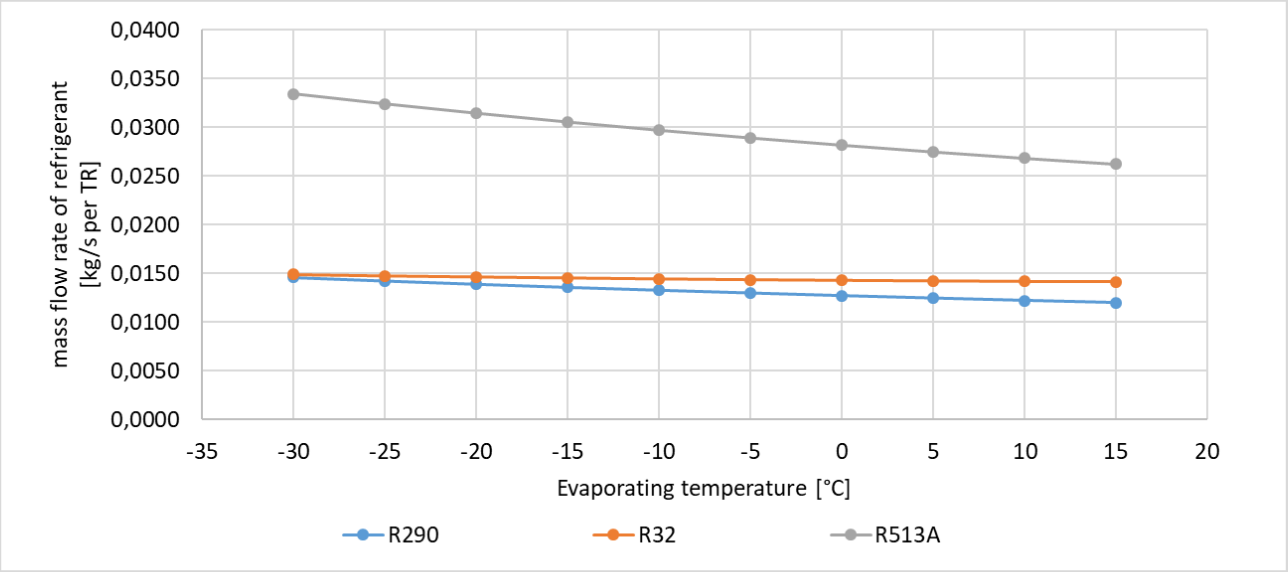


Figure 16: Mass flow rate - evaporating temperature comparison R32, R513A and R290

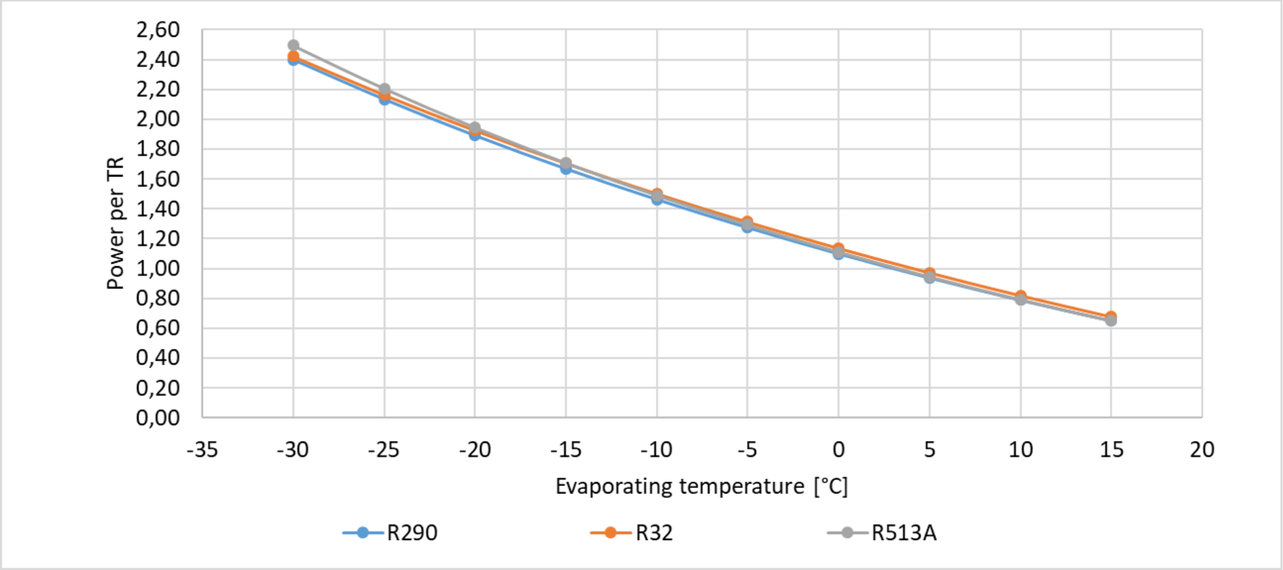


Figure 17: Power per TR - evaporating temperature comparison R32, R513A and R290

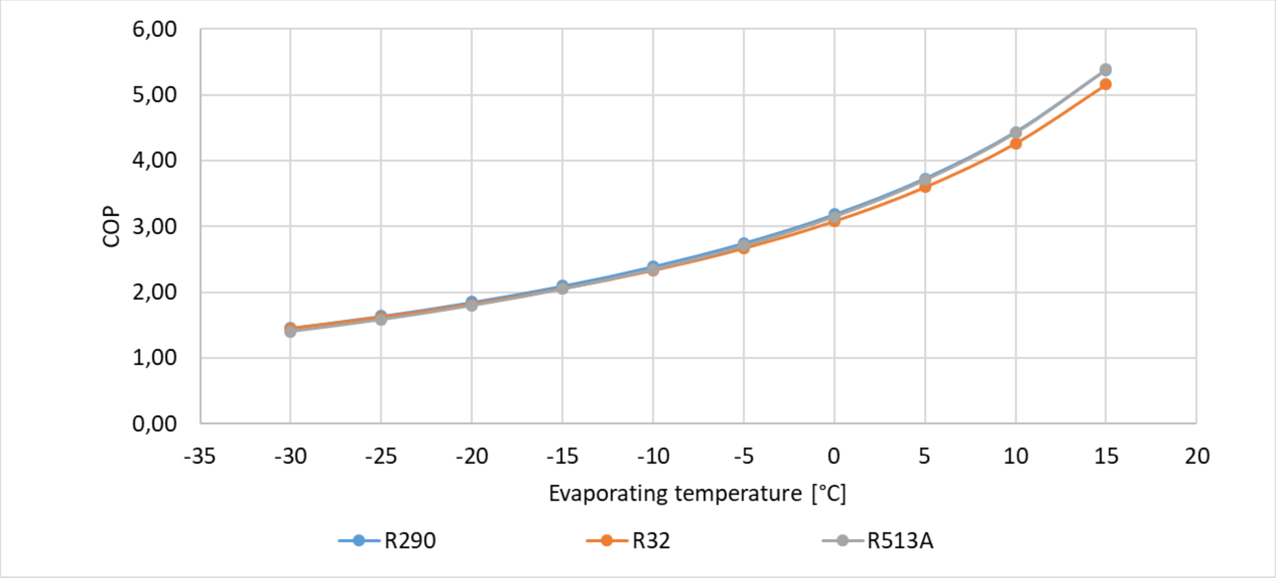


Figure 18: COP - evaporating temperature comparison R32, R513A and R290

Lower pressure ratio is desirable as the volumetric and isentropic efficiencies are expected to increase with decrease in pressure ratio influencing the COP of the system. Compressor discharge temperature is an important characteristic of an alternative refrigerant selection. Lower discharge temperature improve the life of the compressor by improving stability of the refrigerant and the lubricant but on the other hand, must be know that if the target is the production on domestic hot water (DHW) the discharge temperature must be at least 55°C. As show in the figure, the discharge temperature with R290 is much lower than that of R32 and similar with R513A in the condition simulated. A particular note must be done regarding these considerations: discharge temperature of R290 is lower than R32 but here we do not consider that applicability limits of the compressor that regarding the compressors operating at propane reach temperatures significantly higher than the other two and this allows the possibility of generation of DHW.

Volumetric refrigerating capacity is a measure of the size of the compressor required for particular operating condition. Similar value require no change in compressor size. As we can see in the figure, R290 gives lower volumetric refrigerating capacity value compared to R32 but slightly higher value compared to R513A.

Mass flow rate of R290 and R32 are very similar the flow of the fluid is detached from the other passed the -15°C, however R513A have it doubled.

The coefficient of performance (COP) is the measure of energy efficiency of a refrigeration system. As shown in the figure, COP varies also with evaporating temperature. COP values with R290 are almost equal to the other two refrigerants.

In the figure below are shown a comparison between the three refrigerant simulated at the same condition (T_cond=45°C, T_evap= -5°C, SH=SC=5 K).

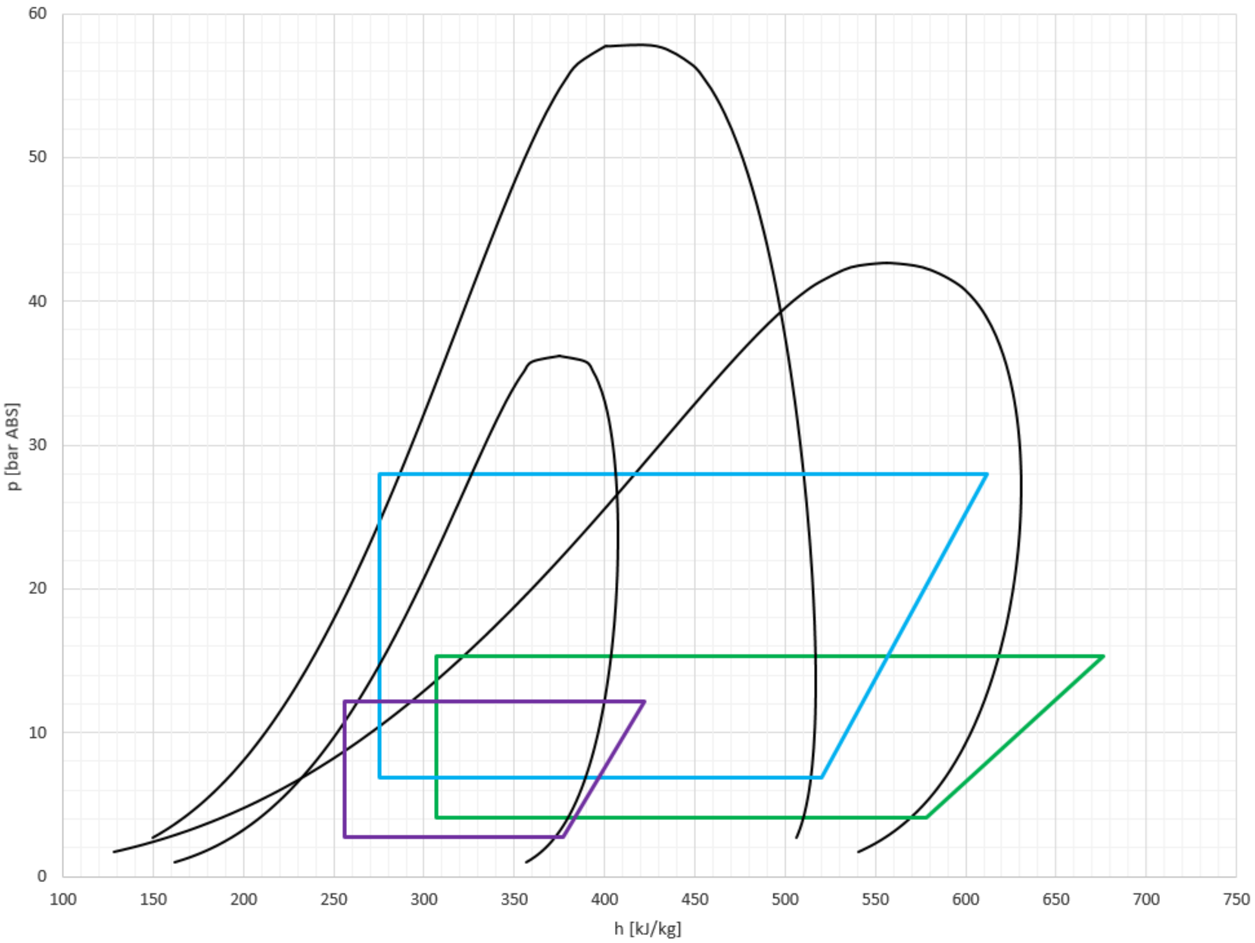


Figure 19: Refrigerant cycle comparison: blue is the R32 cycle, purple is the R513A cycle and green is the R290 cycle

we can see that the enthalpy difference between input/output condenser (as well as for the evaporator) are very different between the three cycles, particularly in the R290 cycle this difference is high. If we write the heat exchange as Q=ṁ*(Δh) we see that the heat exchanged at the condenser is higher than in the other cycles at the same flow rate. This confirms that to exchange the same power of the other two, for R290 will need a smaller amount of refrigerant flow.

Chapter 4: Component compatibility with R290 refrigerant

4.1 Introduction

We have already see in Chapter 2 the main characteristics of the three units, in this chapter the components compatibility with R290 will be analyses.

The components that will be analysed are:

- Compressor
- Heat exchangers
- Electronic expansion valve valve

For all three units, the compressors previously used are not compatible with R290 as there are substantial differences between the three fluids as we have already seen in Chapter 3. The main causes that lead to choosing a dedicated compressor for propane are discussed briefly below.

4.2 Compressor

Figure 20: From left to right: rotary compressor, scroll compressor and screw compressor

4.2.1 Different densities between fluids

As already analysed in Chapter 3, the differences in density in the three fluids is considerable, in fact propane has densities that are about halved compared to the other two refrigerants. This therefore also involves other aspects such as lower speeds, lower pressures and finally higher volumes under the same conditions. This involves a different type of compressor operation, in fact, with the same power, usually the propane compressors process volumes of about 20% greater than the others.

4.2.2 Higher achievable temperatures of R290

In the literature, you can find many studies regarding the field of applicability of propane, in particular propane could work at higher temperatures than other fluids, in fact this also involves a possible interest in using propane for the production of domestic hot water. It is easily understood from the different cycle comparison conducted in Chapter 3, where the difference is substantial. It could also seen from the analysis of the cycle seen before, R290 can reach a temperature near to 75°C inside the compressor easily.

4.2.3 Compressor-oil-fluid compatibility

During past studies on other fluids compatibility, it was found that although theoretically the fluid under investigation was fully compatible with the compressor and the oil used, then in practice the units analysed suffered from a problem of dragging oil into the circuit. This involved various problems within the circuit such as:

A greater concentration (stagnation) of oil in the batteries that then led to a greater inefficiency in the heat exchange of the same being that the oil does not exchange heat as a refrigerant and then subtract heat exchange surface to the same. The oil could not make return in the compressor resulting in incorrect operation of the same that led the compressor to work with less oil than necessary and in cases limit the rupture of the same.

Usually compressor manufacturers provide or recommend, in addition to the compressors themselves, a list of oils that can be used with different refrigerants and their behaviour in different compressor load conditions at different temperatures both during daily operation and during heating-cooling seasons. In addition, the oil and refrigerant mixture must not damage the circuit seals.

Il propano viene classificato come fluido A3 e solitamente questo comporta delle restrizioni riguardanti le componenti con cui viene in contatto. Il compressore da utilizzare con il propano sarebbe meglio se fosse certificato ATEX. The ATEX 2014/34/EU legislation published on 29 March 2014, by the European Union, refers to the harmonisation of the laws of the various member states concerning equipment and protection systems intended for use in potentially explosive atmospheres. Propane units must have ATEX certified components, in particular the compressor must satisfy some conditions, which are:

- No ignition source: safety electrical panel and isolated electrical components
- No leakage source : usually no threads are present, the joints are thermo-brazed to ensure greater tightness and reduce the likelihood of refrigerant leakage
- Containment : no deflagration in the case of initiation

4.3 Heat exchanger

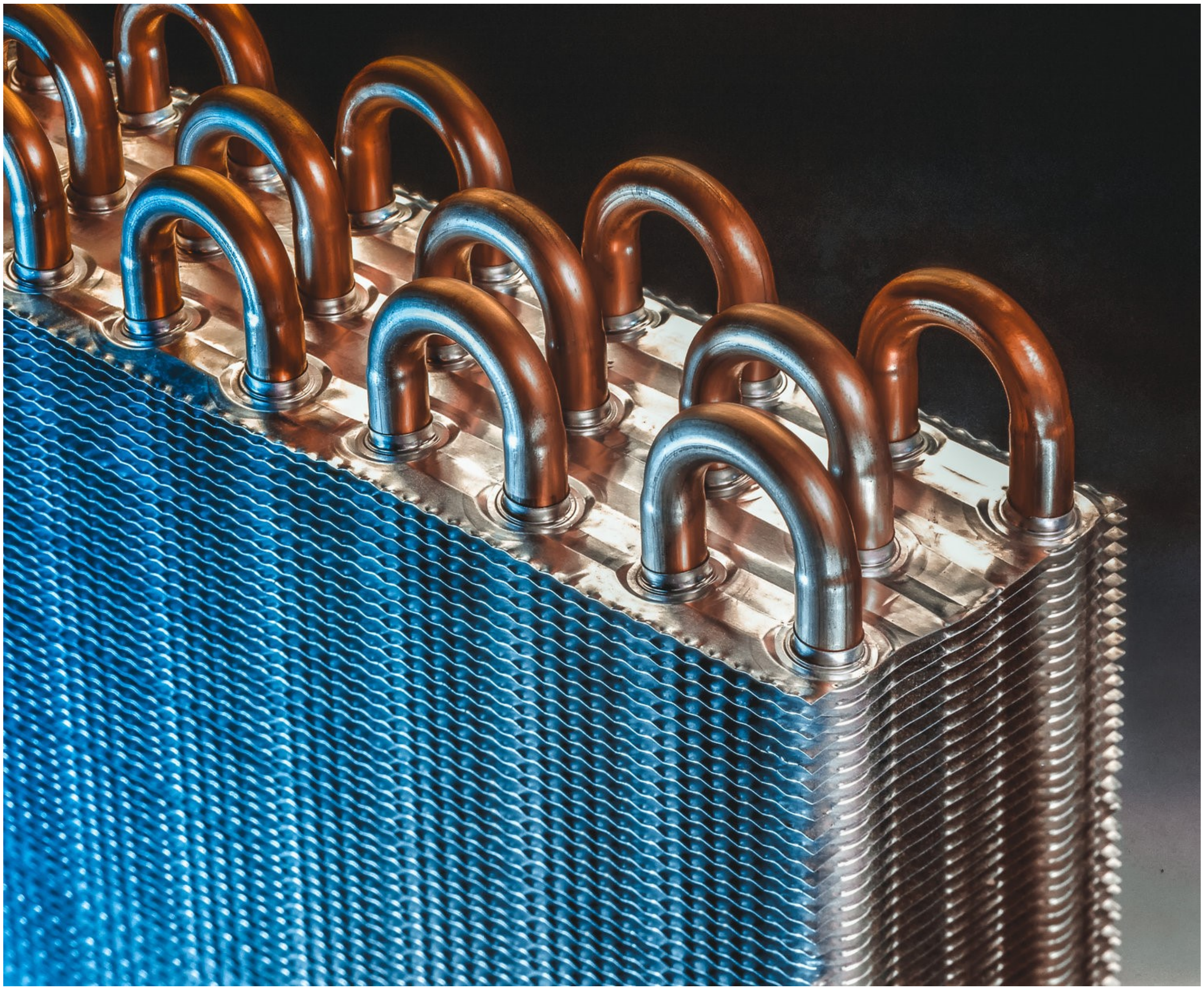


Figure 21: Typical view of fin and tube heat exchanger

Regarding the compatibility between refrigerant R290 and heat exchanger materials used in heat pumps at R32 and R513A, attention should be paid to the following aspects:

1. Corrosion: Propane (R290) is a highly flammable refrigerant, but is not corrosive. On the other hand, R32 is slightly corrosive, especially in the presence of moisture. R513A is a low toxicity, non-flammable refrigerant. So in general commonly used materials, such as stainless steel, aluminum and copper pipes, are generally compatible with the mentioned refrigerants.

2. Pressure: Propane (R290) has a higher operating pressure than R32 and R513A, so native heat exchangers must be able to withstand the higher pressures associated with R290.

In general, if a heat pump is designed to use refrigerant R32 or R513A, the materials should be compatible with refrigerant R290. However, to ensure the compatibility and optimum efficiency of the system, it is always advisable to consult the manufacturer's specifications and follow the manufacturer's instructions for the conversion or use of different refrigerants.

For all three units, the compatibility of the refrigerant fluid with the heat exchangers was analysed by trying to use propane from the existing circuits of the units under investigation. Several tests were carried out both software and practical and no particular compatibility complications were noted because, as we have already mentioned above, propane does not react with copper or even steel; materials present in heat exchangers. Obviously in this chapter we are referring only to the compatibility with the materials and not if the same exchanger operating at R32 is efficient with fluid R290.

4.4 Electronic expansion valve

Figure 22: Typical electonic expansion valve

As the heat load increases at the evaporator, the valve reduces the refrigerant flow to maintain constant evaporation pressure, equal to the valve calibration pressure and vice versa. In short, the valve underfeeds the evaporator when the thermal load is greater and supercharges it when the thermal load is lower. Please note that the presence of vapour in the liquid at the inlet of the valve will decrease the capacity of the valve itself since the steam, with the same weight, occupies a greater space than the liquid that generated it; consequently its presence, reducing the passage of the liquid, will adversely affect the capacity of the valve. When we think about the lamination valve, we can therefore reason in terms of valve opening. From the software of the company of the valve you can search for the compatibility of a certain valve with the operating cycle conditions and, entering all the necessary values, the configurator returns a percentage of valve opening. So, in this case, the compatibility is to understand if the valve can bring the refrigerant from the condensation pressure to that of evaporation.

Obviously, the expansion valve will have to operate at very different pressures with the various fluids, as seen in Chapter 3. Therefore, in conclusion, the lamination valve is compatible with propane, it simply has to be calibrated for the R290 operating cycle.

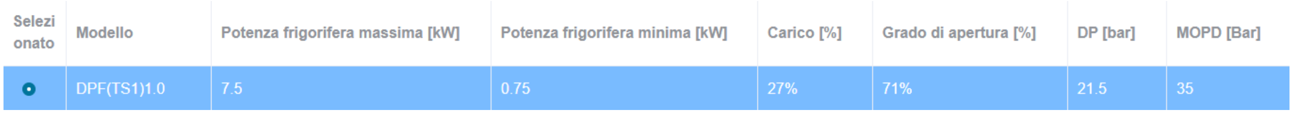
Example:

VMC heat pump (A-A) simulation lamination valve R32 and R290:

The simulation was performed in cooling condition, then with the following values:

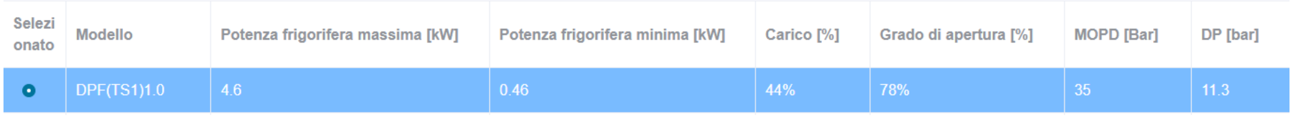
- Condensing temperature = 52 °C
- Evaporating temperature = 11 °C
- Heat load = 2 kW
- Superheating = Subcooling = 5 K
- Load percentage= 60%

You can select the same lamination valve for both fluids. Below we find the output values from the configurator.



Selezionato	Modello	Potenza frigorifera massima [kW]	Potenza frigorifera minima [kW]	Carico [%]	Grado di apertura [%]	DP [bar]	MOPD [Bar]
●	DPF(TS1)1.0	7.5	0.75	27%	71%	21.5	35

Figure 23: Electonic valve for R32



Selezionato	Modello	Potenza frigorifera massima [kW]	Potenza frigorifera minima [kW]	Carico [%]	Grado di apertura [%]	MOPD [Bar]	DP [bar]
●	DPF(TS1)1.0	4.6	0.46	44%	78%	35	11.3

Figure 24: Electonic valve for R290

As we can see from the two tables, under the same conditions, passing from R32 to R290, the valve opening increases and at the same time a decrease of the pressure losses, indicated with DP. Therefore, it is usually necessary to analyse the operating limit conditions to be sure that the expansion valve can work even in the worst conditions. By choosing a wrong expansion valve, for example smaller than necessary, the condenser that is located before the valve tends to flood and then increases the subcooling with consequent loss of exchange inside the exchanger. The system would require a greater opening to be able to balance the system, but being already all open this phenomenon is no longer controllable by the valve.

Chapter 5: Design of heat exchangers via software

5.1 Introduction

In the previous chapters, the main characteristics and behaviour of the fluid were evaluated. In this chapter, the heat exchangers of the three units will be analysed in different heating and cooling conditions through a software called Coildesigner. This chapter is fundamental because it will allow us to obtain the basis for a comparison between theoretical data and real data.

5.2 Simulation procedure

Software used for the simulation: Microsoft Excel, NIST REFPROP and Coildesigner.

The procedure of designing a heat exchanger consists of several steps, some initial parameters must be chosen:

- The heating and/or cooling power is fixed, decided at the design stage, since it depends on the thermal load required by the unit.
- Subcooling (SC), superheating (SH): The values of SH and SC have been set on the basis of experience: it has been seen that with a SC of 5 K usually the fluid comes out of the bell of the graph p-h. This allows to have all liquid entering the expansion valve and that is made because if the expansion device is crossed by some steam bubble could works badly. Also the value of SH is fixed at 5 K, to get out of the vapour saturation curve of the graph p-h and have only steam phase at the entrance of the compressor. If liquid refrigerant is present, the compressor may suffer considerable damage, or in the worst cases may lead to breakage. As shown in the graph below, these two conditions are fixed by the designer on the REFPROP.
- Evaporation temperature (T_evap) and condensation temperature (T_cond): condensation and evaporation temperatures are chosen to meet the minimum efficiency while keeping the unit costs acceptable

It is well known that during heating, the condenser must release heat to the fluid, so it is usually tried to have a condensation temperature as high as possible. One of the advantages of compressors operating with R290 is that they can operate in a higher temperature range than conventional refrigerants. This has led to considering also the possibility of generating domestic hot water, which is usually in the range of 35-75 °C.

The mass flow calculation is given by:

$$\dot{m} = \frac{P_{exchanged}}{\Delta h}$$

using REFPROP, you can get enthalpies related to the points of the graph in the desired conditions and so obtain the refrigerant flow rate.

In the simulation software, physical and geometric properties of the finned coil under consideration are defined:

- tubes length
- tubes per bank
- number of banks
- tube outer/inner diameter
- horizontal and vertical spacing between tubes
- fin spacing
- tube internal surface parameters

Then the type of fluid and the correlation coefficients are fixed in order to obtain the most likely real results, which are linked to the type of heat exchanger and type of heat exchange. Correlation coefficient are various and are based on research studies during the decades. Correlations were introduced in the calculations of the heat transfer coefficients when it was understood that from the theoretical calculation to the practical case there was a large deviation in the results. Over the years, many researchers have published their correlations classes of fluids, conditions of motion and geometric configurations. Just as an example, we report a famous correlation, called Dittus-Bölter correlation that provides the values of heat transfer coefficient for forced convection in the case of motion at relatively high speeds (turbulent motion) of fluids inside ducts:

$$Nu = 0.023\, Re^{0.8} Pr^{0.4} \quad Nu = \frac{h_c\, D}{k}; Re = \frac{\rho\, w\, D}{\mu}; Pr = \frac{\mu\, c_p}{k}$$

It can be noted that the three groups (Nu, Re, Pr) that appear in the above equation are dimensionless: they are respectively the numbers of Nusselt, Reynolds, and Prandtl.

This is just one of the many correlation coefficients used in the software, in particular four different coefficients are used to describe the heat exchange:

- Air side heat transfer
- Liquid side heat transfer refrigerant side
- Two phase refrigerant side
- Vapour phase refrigerant side

Four other pressure drop coefficients are also used:

- Air side pressure drop
- Liquid phase pressure drop refrigerant side
- Refrigerant Two phase pressure drop refrigerant side
- Refrigerant Vapour phase refrigerant side

Then we start to trace the connections between tubes: we go to identify the circuitry of the individual circuits that make up the coil, also indicating the fluid inlet and outlet. Remember that the more a heat exchanger has a symmetrical geometry, the more the heat exchange between the various circuits will be equal and there will be no evident imbalance between each other.

Another important consideration during the tracing of coil circuits is that usually to promote the evaporation of the fluid you want the fluid to enter from below and exit from above. This is done to promote the natural process of natural circulation and losses due to mass transport are reduced: when a fluid absorbs heat, the temperature increases and its density decreases. This will lead the fluid to be two-phase and a circuiting upwards will allow separating the liquid phase from the vapour phase, thus allowing the vapour phase to rise upwards and then exit the exchanger. It is not true for the condenser, because a circuiting down allows the natural flow of the liquid and an easier drag of the oil.

The ambient condition to fix during the simulation are the temperature and the relative humidity. These parameters depend on the behaviour condition considered:

- Installation inside or outside the building to be conditioned
- Geographical location of installation
- Heating or cooling condition

Standard points are defined by the standard EN 14511.

We also must consider the air flow rate of the fans and then the air flow rate that will impact the heat exchanger (given from data sheets of fan's manufacturers). To consider the real effect of air on the heat exchanger and also the fact that the air flow isn't uniform, you can use a polynomial distribution of air if you know it.

As already mentioned, during the study of heat exchangers, we must remember that they can have two main configurations: concurrent and counter current flow.

Concurrent → when the refrigerant inside the exchanger flows in the same direction of the fluid to be heated/ cooled (air or water) the exchange is define as concurrent flow.

Countercurrent → is defined countercurrent flow when the fluids pass through the exchanger in opposite directions

Remember that when the heat pump is designed to be able to reverse the cycle, one exchanger will be crossed by the refrigerant in reverse of the other operation. Which implies that if the heat exchanger is designed to run in counter current then, once the cycle is reversed, it will be in the cocurrent condition and in this condition it can be considered that the heat exchange is lower and therefore with worsening values compared to the previous case.

Special attention must be paid to the configuration in order to favour the heat exchange of the exchanger with greater importance. It is usually preferred to favour the counter current evaporator for the following reasons:

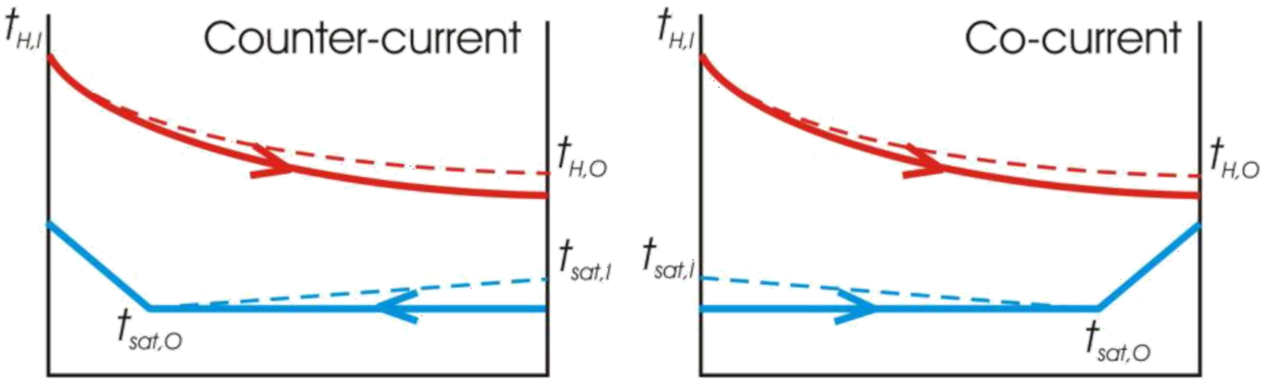


Figure 25: Counter flow and co-current flow configuration

In the case of dry expansion evaporators, the steam leaving them must be superheated by at least 5 K; The figure shows the temperature profiles in the two configurations during the evaporation and overheating process, admitting the presence of pressure drops only in the two-phase zone. The overheating of steam is characterized by low coefficients of heat exchange: superheated steam is a bad conductor of heat and has a low thermal conductivity, which means it will have a low coefficient of heat exchange. Saturated steam has a better thermal conductivity thanks to the mixture of particles of liquid and gas. Liquid water being a good heat conductor increases the overall heat exchange coefficient of saturated steam.

For this reasons it is advisable to associate to this area the highest values of temperature difference, this is possible only in the configuration in countercurrent. In concurrent, in fact, the profiles at the exit close and the heat exchanger would have an inefficient area from the point of view of heat exchange characterized by low coefficients of exchange and small differences in temperature. This example explains the rule universally followed in the design of dry expansion evaporators, which provides for the arrangement of fluids in countercurrent.

Enter inside the Fluid Inlet States the values of mass flow rate and vapour content of the refrigerant previously calculated with REFPROP.

We assume an initial pressure from which to start the simulations, this will be the only parameter that we will change (increase or decrease) until the achievement of the conditions of interest (example superheating or subcooling =5 K) keeping constant the mass fraction, the mass flow rate and so the heating or cooling power.

Finally, the results are exported to an Excel sheet and the parameters of interest for sizing purposes are identified.

Several simulations are then performed, for example by changing the number of circuits or the banks of the heat exchangers, keeping the temperatures and the refrigerant flow and the exchanged power equal to the previous ones.

Obviously if the number of circuits or the ranks increase/decrease, the geometry of the circuits could change significantly.

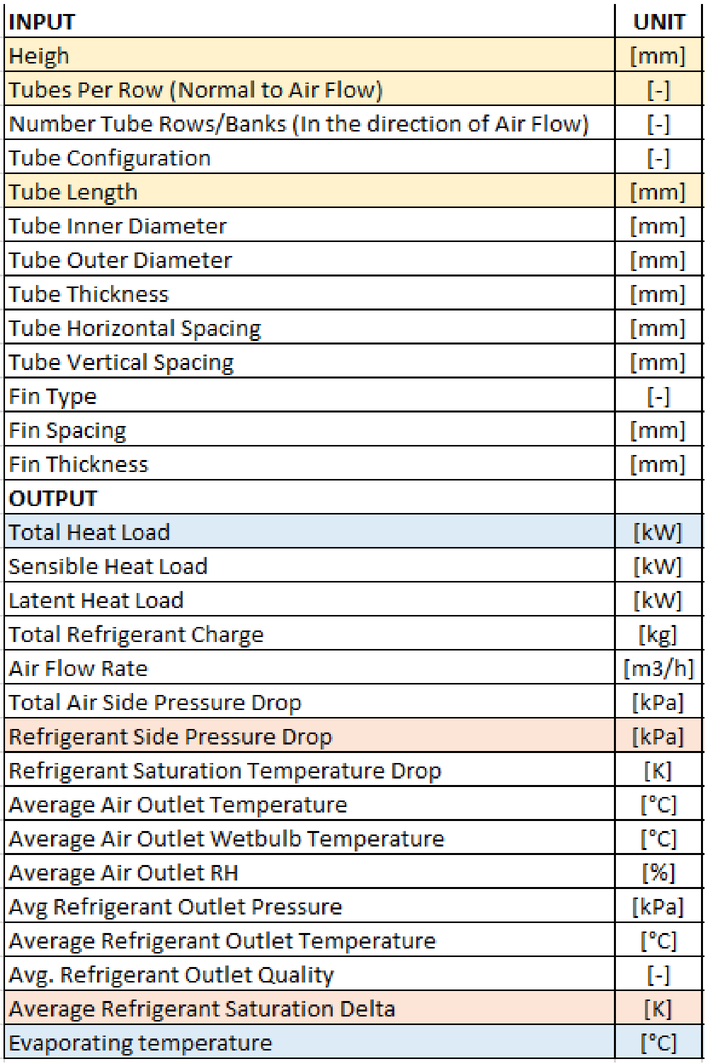
The more circuits/banks/pipe thickness combinations you make, the more likely you are to approach a coil with the characteristics we have set out. Usually a good finned coil heat exchanger meets the parameters that we set or that we are willing to accept such as the refrigerant side pressure drop, the refrigerant flow rate, or still the evaporation temperature as similar as possible to the fixed value initially.

The same finned coil can then be simulated under reverse operating conditions.

Please note that heat exchangers have a preferential operational working for which you give more weight in the design phase. Usually not optimal for heating/cooling, so a compromise is sought.

The results obtained by CoilDesigner are many, exporting them in an Excel file you can isolate the most important data for sizing and comparison. The table data is divided into input and output:

- Input: you can find all the main geometric characteristics of the exchanger. It is very important to see them visually because it allows us to understand if we are considering heat exchangers with the same geometries but with different piping patterns or if the geometry has changed for example by changing the fin spacing, or the number of ranks of the exchanger. In general, this section is useful to understand at a glance which exchanger we are considering.
- Output: this section is the core of the sizing. The most important data are coloured in different colours, such as refrigerant charge, refrigerant side pressure drop and ΔT. In addition to these, the temperature of evaporation or condensation are also reported: we are interested in the evaporation temperature in the case of study of an evaporative exchanger and vice versa in the case of condensing exchanger. From this section, we can understand and reason if the exchanger could be used in our unit or if it is unsuitable. All these arguments depend, as mentioned above, on the compromises that the designer is willing to make. Above a template is reported.



INPUT	UNIT
Heigh	[mm]
Tubes Per Row (Normal to Air Flow)	[-]
Number Tube Rows/Banks (In the direction of Air Flow)	[-]
Tube Configuration	[-]
Tube Length	[mm]
Tube Inner Diameter	[mm]
Tube Outer Diameter	[mm]
Tube Thickness	[mm]
Tube Horizontal Spacing	[mm]
Tube Vertical Spacing	[mm]
Fin Type	[-]
Fin Spacing	[mm]
Fin Thickness	[mm]
OUTPUT	
Total Heat Load	[kW]
Sensible Heat Load	[kW]
Latent Heat Load	[kW]
Total Refrigerant Charge	[kg]
Air Flow Rate	[m3/h]
Total Air Side Pressure Drop	[kPa]
Refrigerant Side Pressure Drop	[kPa]
Refrigerant Saturation Temperature Drop	[K]
Average Air Outlet Temperature	[°C]
Average Air Outlet Wetbulb Temperature	[°C]
Average Air Outlet RH	[%]
Avg Refrigerant Outlet Pressure	[kPa]
Average Refrigerant Outlet Temperature	[°C]
Avg. Refrigerant Outlet Quality	[-]
Average Refrigerant Saturation Delta	[K]
Evaporating temperature	[°C]

Figure 26: Excel template for the results

The "Total Heat Load" will be constant in all configurations for the same unit because it is one of the data that is fixed initially and on which depend the parameters of flow and mass fraction. So, this parameter is useful for checking if the conditions at which we want the exchanger to work are those designated.

Another important parameter is the "Total Refrigerant Charge" because when working with heat pumps and in particular with those operating with propane, there are legal limits of total charge to be respected as already explained in Chapter 1. Usually the choice of an exchanger is also based on this value: if a choice has to be made, other conditions being equal, an exchanger containing less refrigerant is preferred.

The "Refrigerant Side Pressure Drop" is one of the parameters where it is more difficult to explain the decision and choice process. It usually occurs that the pressure drop is in a decided range based on many aspects: gaining more experience with different units is easier to understand if the pressure drop can be acceptable, or too high. If the general behaviour of the refrigerant is known, it can be estimated whether there will be problems of oil dragging as a function of the pressure drop. In fact, we must remember that the pressure drop varies with the square of the velocity so, high pressure drop means high velocity and this can leads to drag of oil but also in vibration problem that we want to avoid.

The "Average Refrigerant Saturation Delta" is the value of ΔT sought during the software simulation performed above, through the repeated simulations we try to obtain a value of ΔT= ± 5 [K].

The "Evaporating Temperature" or "Condensing Temperature" is given by the difference between the outlet temperature of the refrigerant and the ΔT. This is a very important parameter to consider. In particular, when we consider an evaporator in heating condition and the evaporating temperature is lower than -2 °C we could run into the frosting phenomenon. Frosting could occurs when we are in then heating condition and the T_evap is lower than -2 °C and the external temperature is lower than 5°C, the humidity of the external air in contact with the metal of the evaporator frost and slowly covers it completely. In this condition the heat exchanger freezes and does not allow the fluid to exchange heat. Defrosting is an unavoidable phase of any heat pump machine that intervenes every time the evaporator is so frosted as to seriously affect efficiency. Therefore the less defrosting cycles the machine runs, the lower the efficiency loss.

It is not easy to explain the process of choosing an heat exchanger because a lot of it is based on the experience of the designer, but in the sections below we tried to better explain the reasoning performed on each individual unit from a theoretical point of view integrated with practical aspects.

5.3 VMC heat pump (A-A)

Simulations were performed on both heat exchangers for the VMC heat pump (A-A). Both exchangers, called source and treatment, exchange heat with the air fluid.

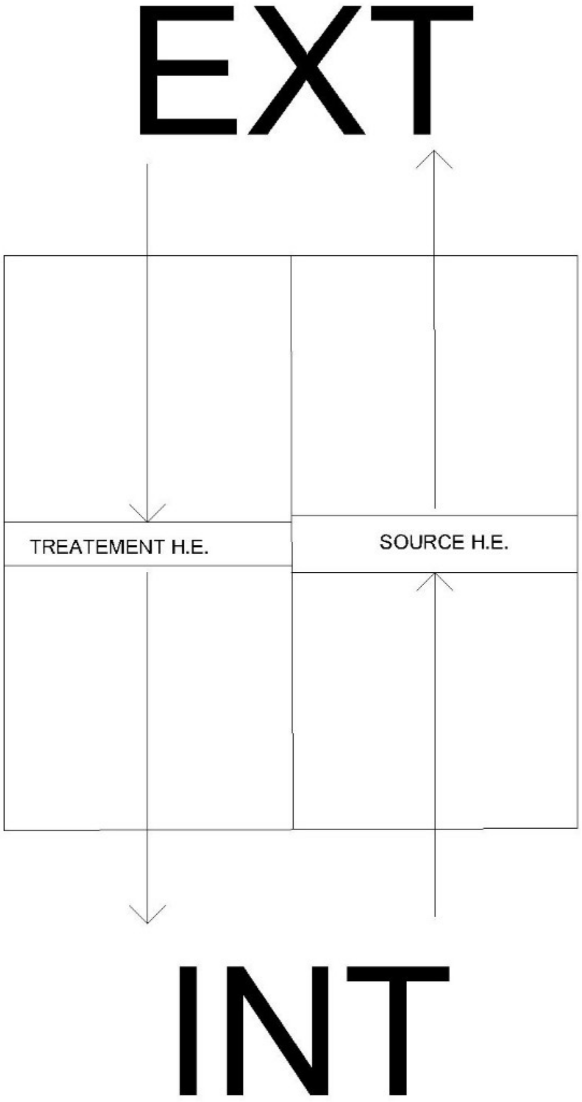


Figure 27: Simplified representation VMC heat pump with air flux

In the image, the treatment exchanger is located on the left and the airflow that passes through it is unidirectional: from the outside to the inside, while the source exchanger, on the right in the image, will have opposite airflow direction and then from inside the environment to the outside. It was decided to keep the external dimensions of the exchangers as from the original with the unit operating at R32 both for a matter of size and to keep the unit as unchanged as possible so as not to add costs to the construction of the new heat pump.

Treatment heat exchanger

	T_{cond} [°C]	T_{evap} [°C]	T_{ext} [°C]	RH [%]	P_{exch} [kW]
Cooling condition	52	11	35	40	2
Heating condition	30	5	7	87	1.9

Figure 28: Simulation condition treatment heat exchanger

Source heat exchanger

	T_{cond} [°C]	T_{evap} [°C]	T_{ext} [°C]	RH [%]	P_{exch} [kW]
Cooling condition	52	11	27	47	2
Heating condition	30	5	20	50	1.75

Figure 29: Simulation condition source heat exchanger

As already mentioned for the source exchanger we see that T_ext=T_ripresa and therefore its conditions will be equal to the conditions inside the environment to be conditioned.

That said, we insert in the Excel sheet integrated with REFPROP the above information to search for the operating points of the cycle to R290 and the flow of refrigerant; the information that interests us will depend on the type of operation of the exchanger:

If we are analysing the evaporator operation we will use the conditions of point 6 of the cycle with vapour content, refrigerant flow rate and corresponding pressure; if we analyse the condenser operation we will look at the refrigerant flow conditions, the discharge temperature at the compressor (and then entering the condensing exchanger) and the pressure.

From this simulation we want to understand how pressure losses on the refrigerant side, refrigerant charge and T_evap and T_cond change by varying the heat exchanger circuits, keeping all other conditions unchanged.

Using the Coildesigner software, the values of m_ref and titre or temperature (depending on operation) will be kept fixed, varying the pressure to obtain the desired power that we have fixed and aiming to obtain an overheating equal to 5K.

5.3.1 Starting configurations for heat exchangers

Treatment: exchanger 3 ranks, 11 tubes per rank. Designed to operate in countercurrent when it works as an evaporator and then in equicurrent when it works as a condenser.

Source: 6-rank heat exchanger, 11 tubes per rank. Designed to operate in equicurrent when the evaporator is running, in counter current when the condenser is running. Usually it is preferred to favour the evaporation with a counter current exchange, but for issues of geometry and dimensions the exchanger must be sized in these conditions.

The circuitry was designed following a fixed logic: if they size evaporator tried to make the fluid inlet start from the bottom and the output at the top, and then when it works as a condenser the flow will go down. We usually prefer to have the circuits as symmetrical as possible so that the circuits exchange the same heat each. Sometimes it is complicated to have the same circuitry in all circuits, so you prefer to place in the lower (or higher) circuits with more numbers of tubes.

The simulations and then the collected data will follow this logic:

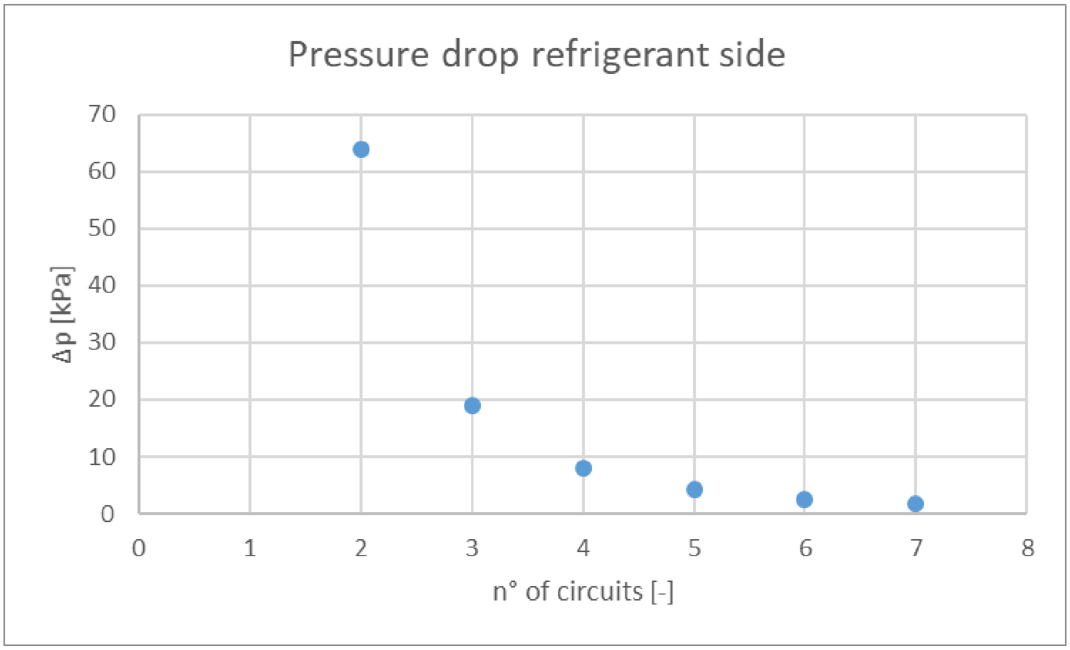
- Simulated battery treatment with a pipe diameter from 5mm varying the number of circuits from 1 to 8 in both heating and cooling conditions thus obtaining 16 simulations.
- Simulated source battery with a pipe diameter from 5mm varying the number of circuits from 1 to 8 in both heating and cooling conditions thus obtaining 16 simulations.
- Simulated battery treatment with a pipe diameter from 4mm varying the number of circuits from 1 to 8 in both heating and cooling conditions thus obtaining 16 simulations.
- Simulated source battery with a pipe diameter from 4mm varying the number of circuits from 1 to 8 in both heating and cooling conditions thus obtaining 16 simulations.

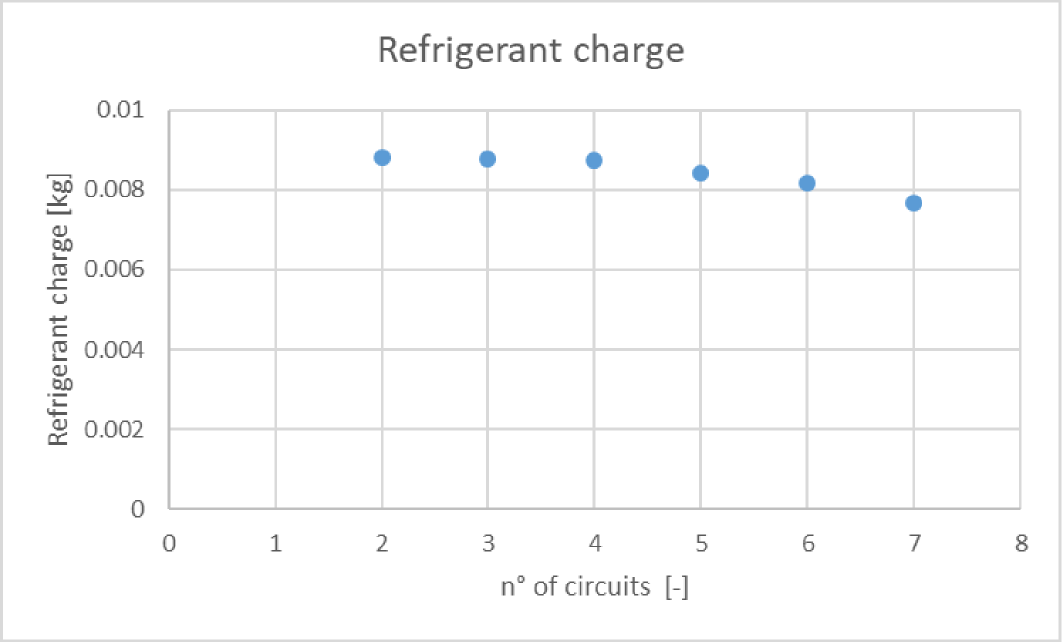
The output parameters returned by the software are many, but for this comparison and to be able to set the comparison with the real case, we would focus on: pressure drop refrigerant side, refrigerant charge and T_evap or T_cond.

During data collection, we noticed that some values at extremes (single circuits and 8 circuits) were atypical. One has to look for the motivation in the correlations for the refrigerant and for the pressure used in the program: as we know, they are based on the numbers of Nusselt, Prandtl and Reynolds and therefore depend on parameters such as fluid velocity, pipe diameter, etc. and apply only to certain fluid velocity ranges. They are therefore excluded from the graphed points as they would lead to a fallacious evaluation of the results. Below are the graphs of Δp, refrigerant charge and T_cond/T_evap for all the simulations described above.

Treatment heat exchanger 5mm

Cooling - Evaporator





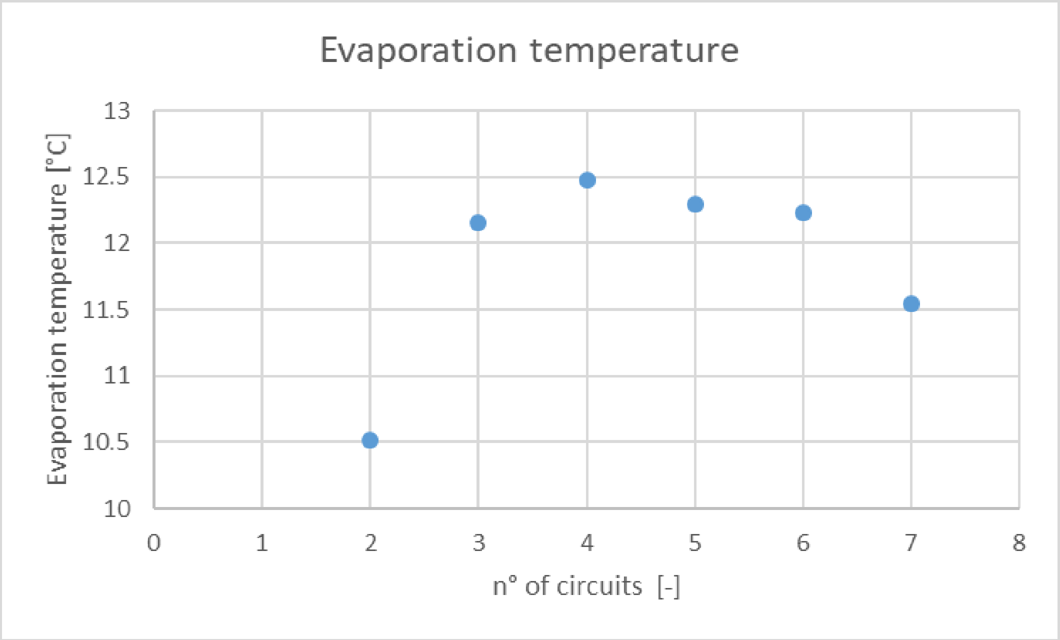


Figure 30: Pressure drop, refrigerant charge and evaporation temperature for different n° of circuits for 5mm treatment heat exchanger

As you can see in the graph the pressure drop decreases as a function of the number of circuits of the exchanger. This is due to the fact that the relationship between pressure losses and fluid velocity is given by:

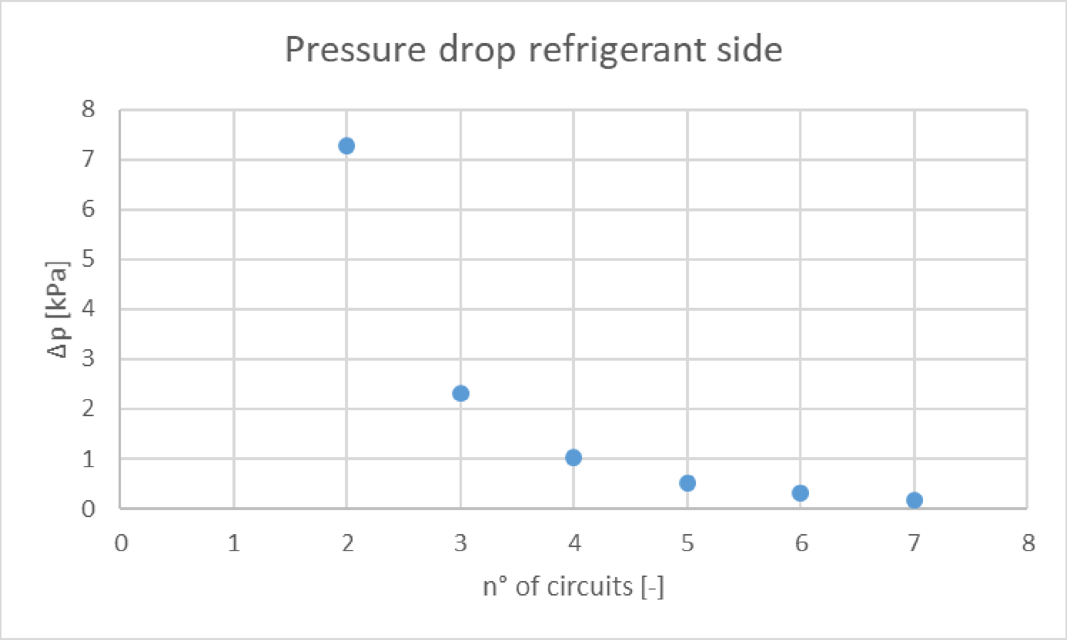
$$\Delta p = \frac{64}{Re}\frac{L}{D}\frac{w^2}{2}\rho$$

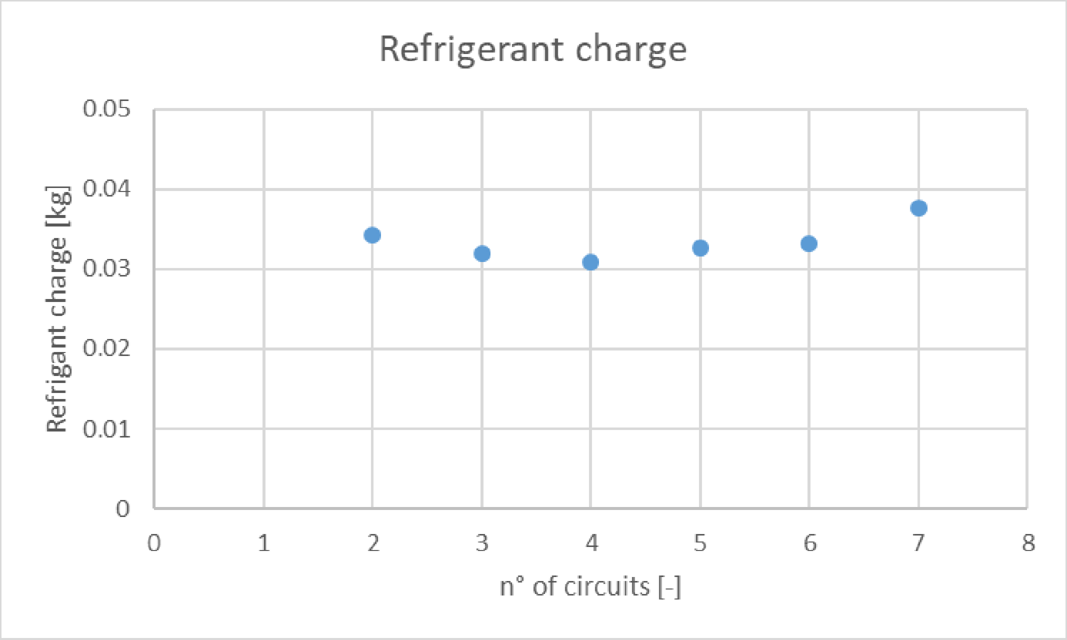
So as we can see, with the same refrigerant flow rate, if the number of circuits increases then the flow is divided over several circuits and the speed in the single circuit decreases and this in turn leads to a decrease in pressure drop. We see that the point that differs most from the others is that relating to the 2 circuits getting to a value three times higher than the next.

As for the refrigerant charge, we see it is very low in general, and tends to decrease slightly as circuits increase, in any case it remains in a range between 7grams and 8.8 grams. This minimum variation can also be justified by a non-completely homogeneous exchanger circuit.

The T_evap in this graph is not very homogeneous and as we can see it varies between 11.5 and 12.5°C, except for the value associated with the 2 circuits that goes down to 10.5°C. It is reached a maximum of 12.5°C, corresponding to the 4 circuits and then start a fall/ drop down as the circuits increase.

Heading-Condenser





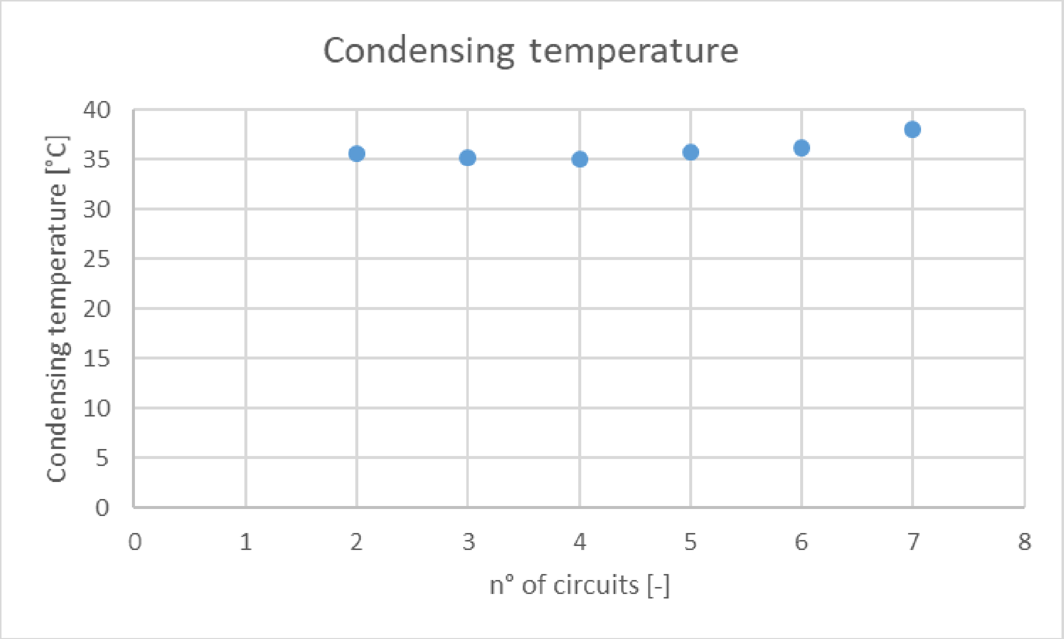


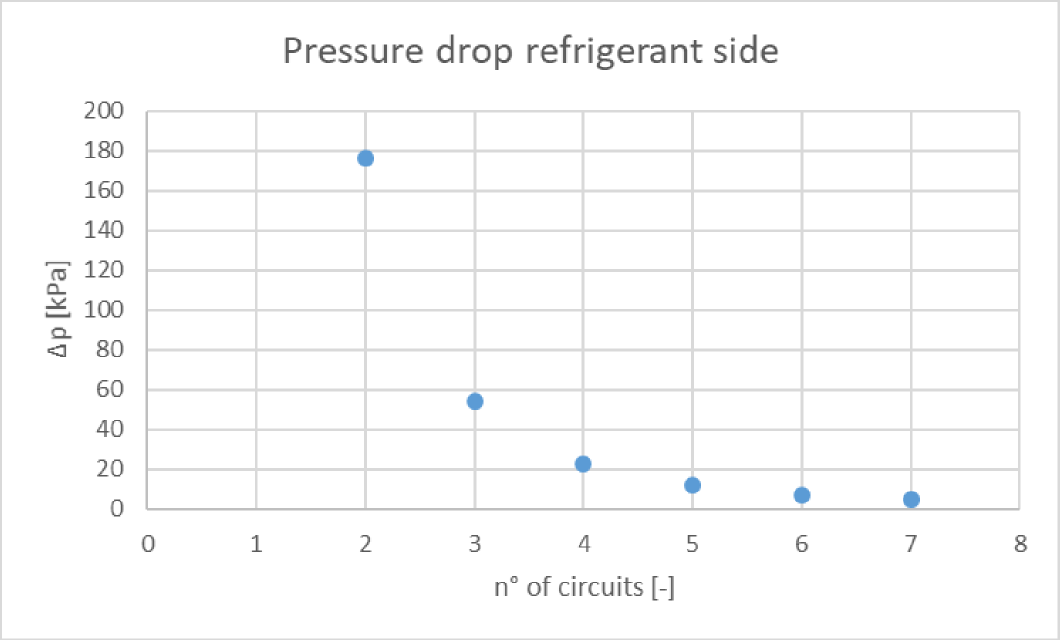
Figure 31: Pressure drop, refrigerant charge and condensing temperature for different n°of circuits for 5mm treatment heat exchanger

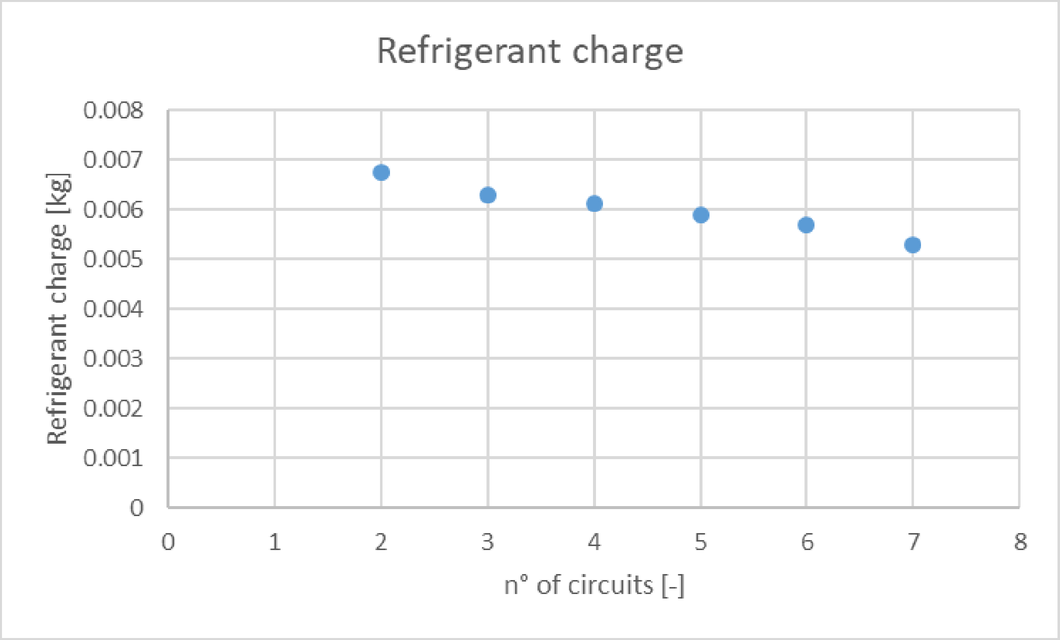
You can see the same trend of the pressure drop also in the operation of the condenser, but with different values, where the maximum is reached at 2 circuits with 7.3 kPa and then go to decrease with increasing circuits.

The refrigerant charge is almost constant and varies in a range of 7 grams (30-37g). The result is a constant trend of consumption with a small increase in the circuits.

Treatment heat exchanger 4mm

Evaporator





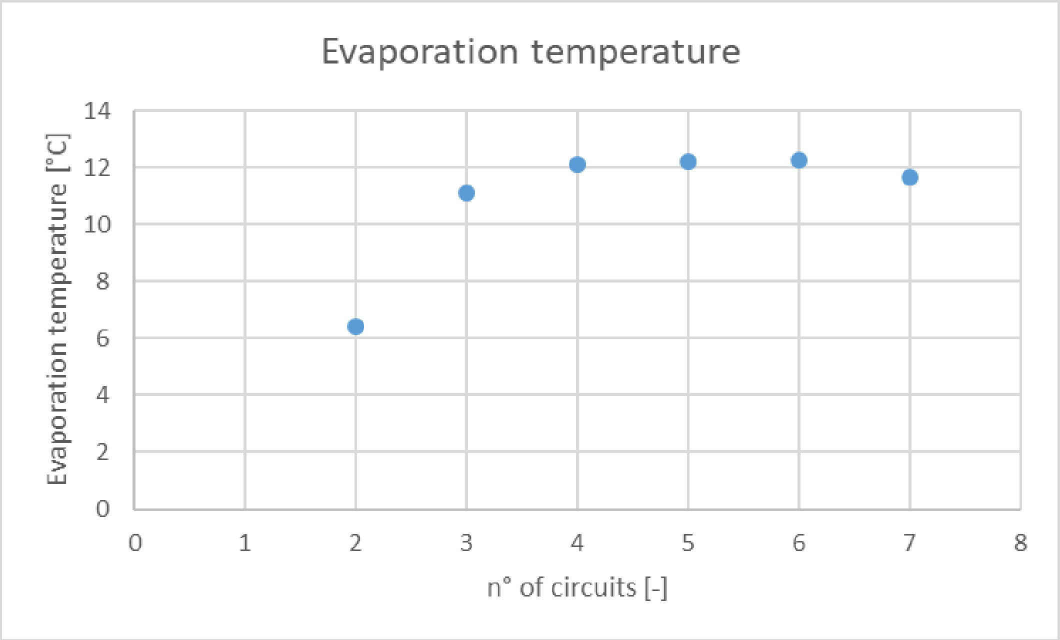
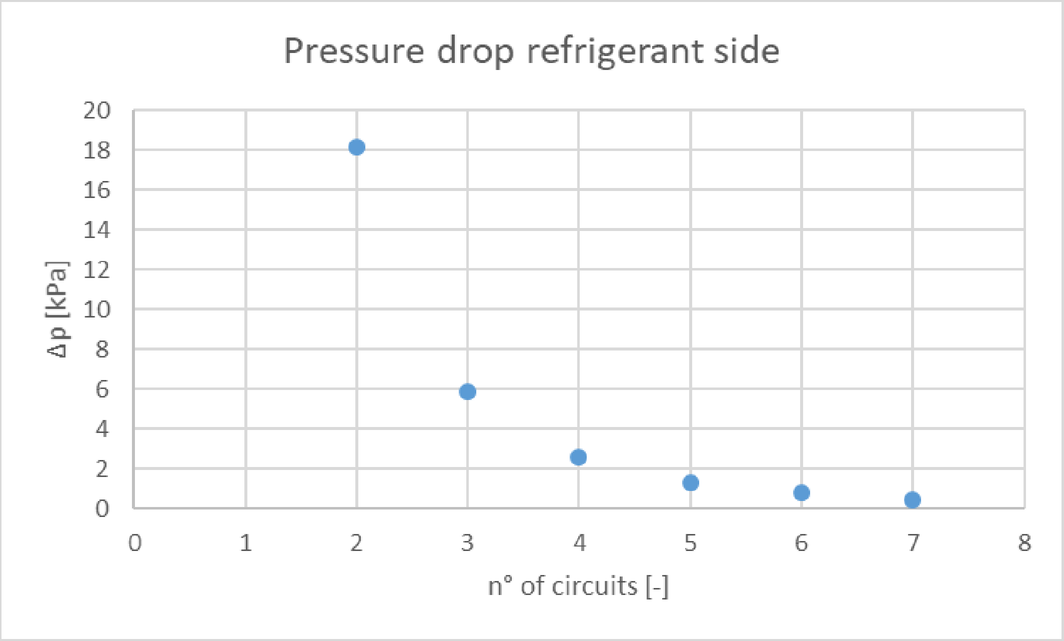
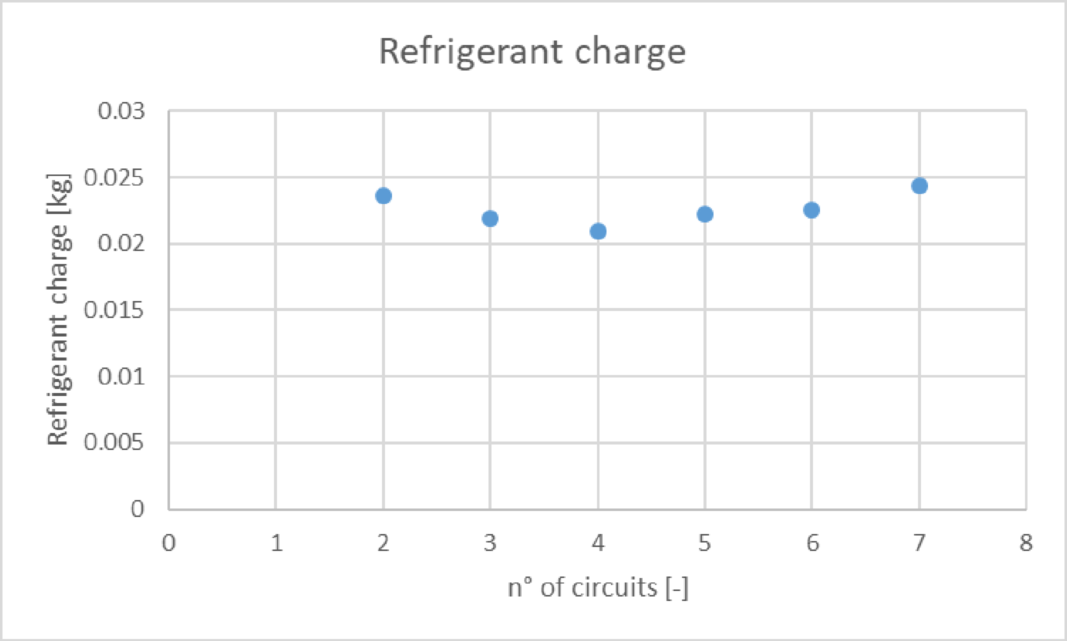


Figure 32: Pressure drop, refrigerant charge and evaporation temperature for different n° of circuits for 4mm treatmentheat exchanger

Condenser





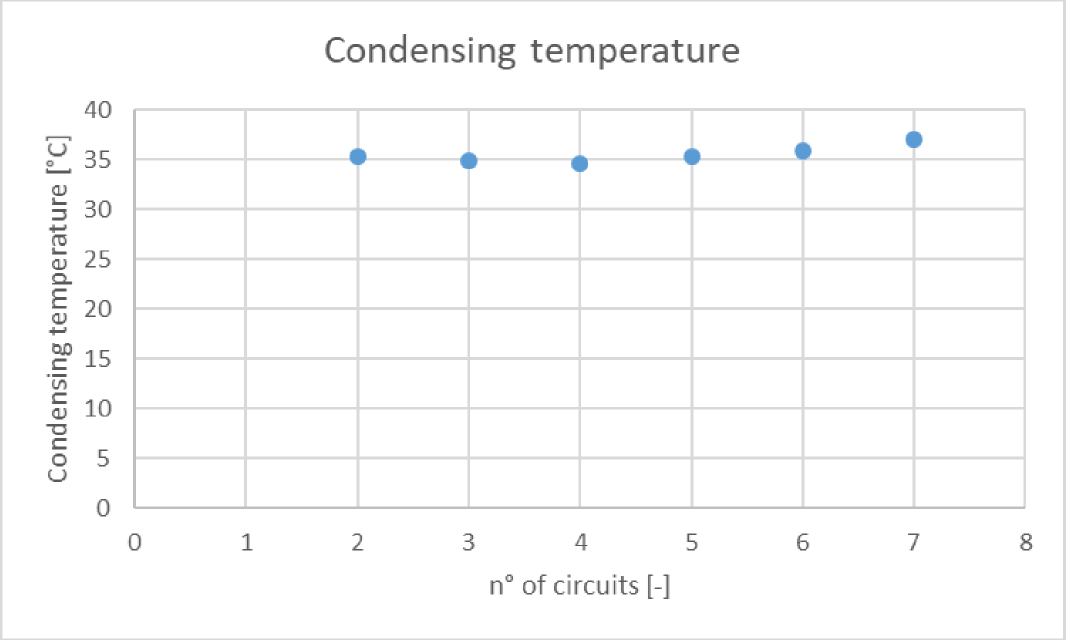


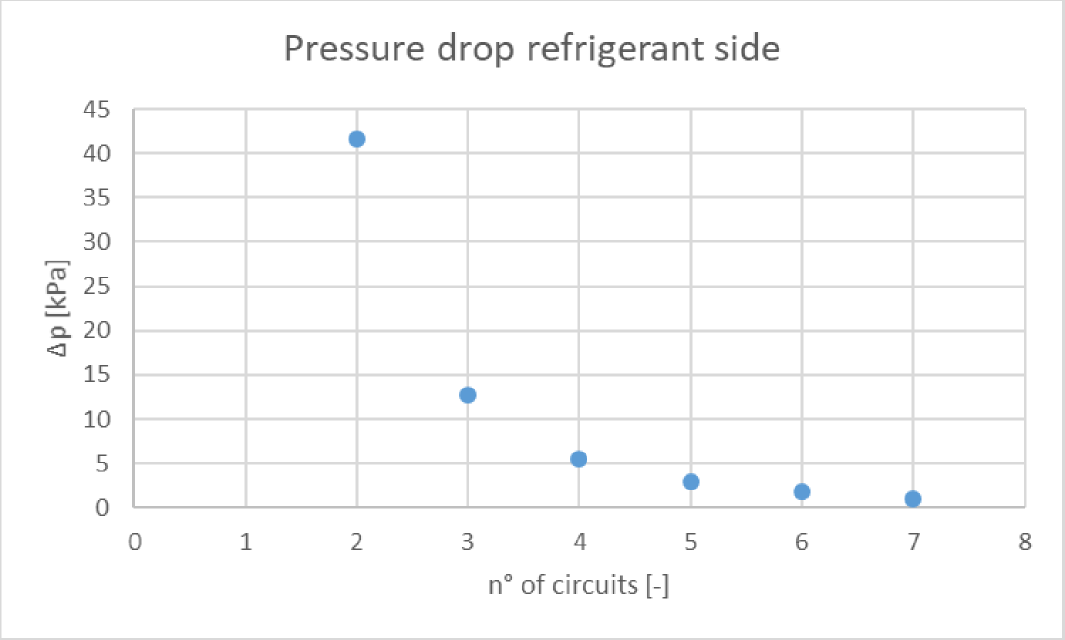
Figure 33: Pressure drop, refrigerant charge and condensing temperature for different n° of circuits for 4mm treatmentheat exchanger

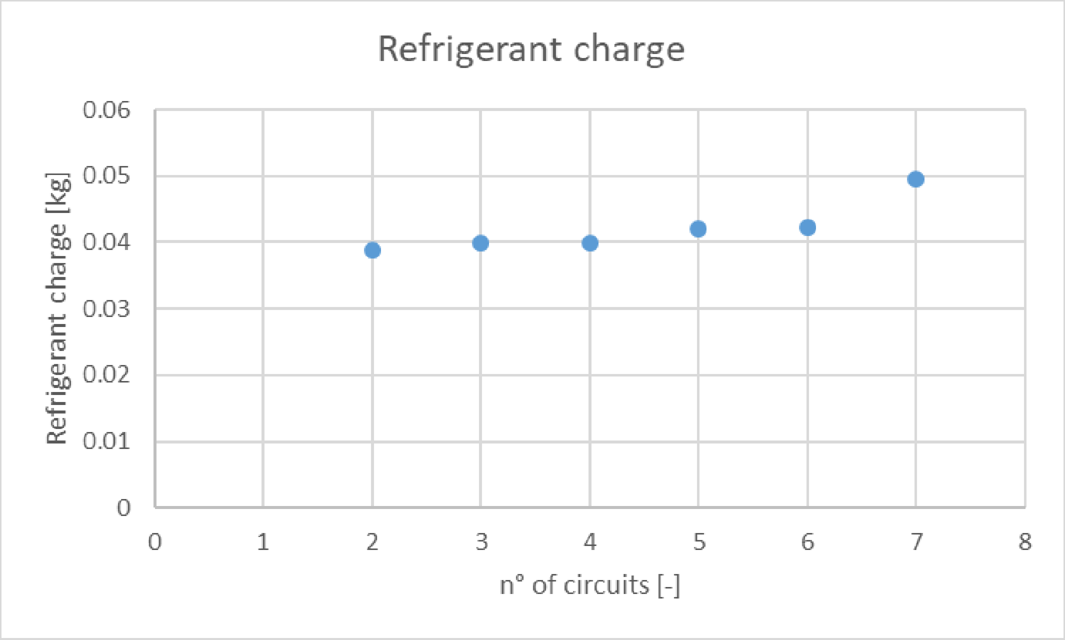
We can see that, although they are different values compared to the cases at 5mm, the trends are similar. Then in this chapter, we will analyse the differences between the 4mm and 5mm configurations.

Source heat exchanger 5mm

As mentioned before, the source exchanger will act as a condenser in cooling conditions and as an evaporator in heating conditions.

Cooling - Condenser





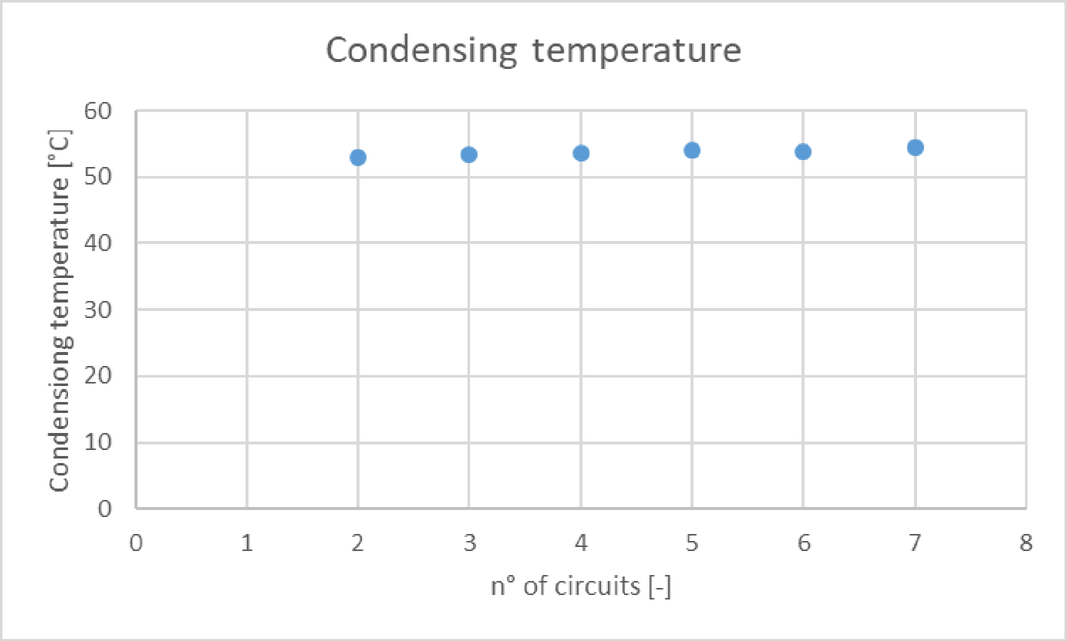


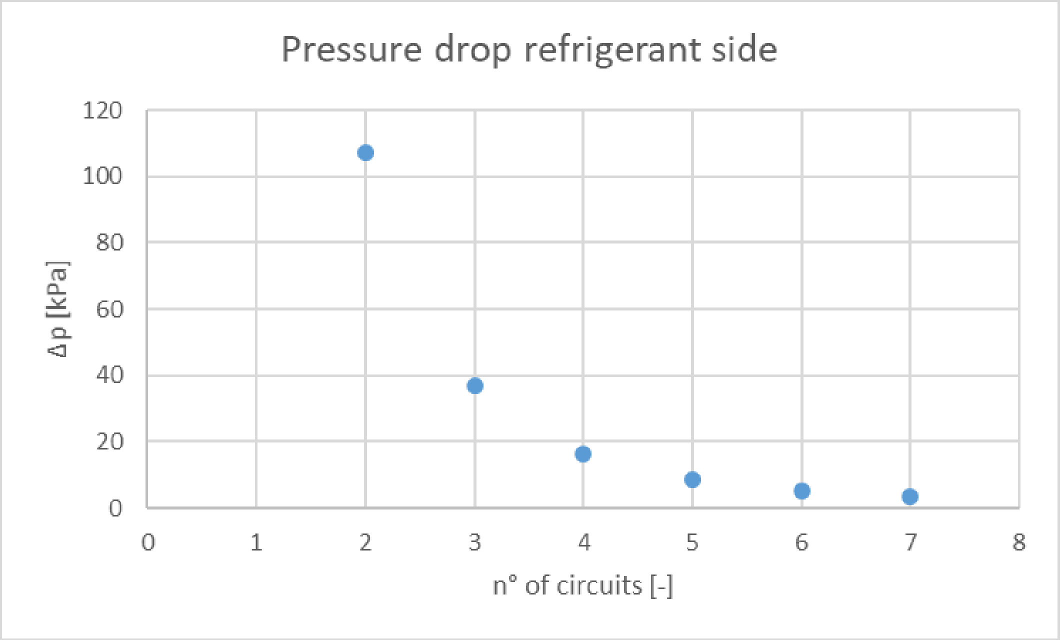
Figure 34: Pressure drop, refrigerant charge and condensing temperature for different n° of circuits for 5mm source heat exchanger

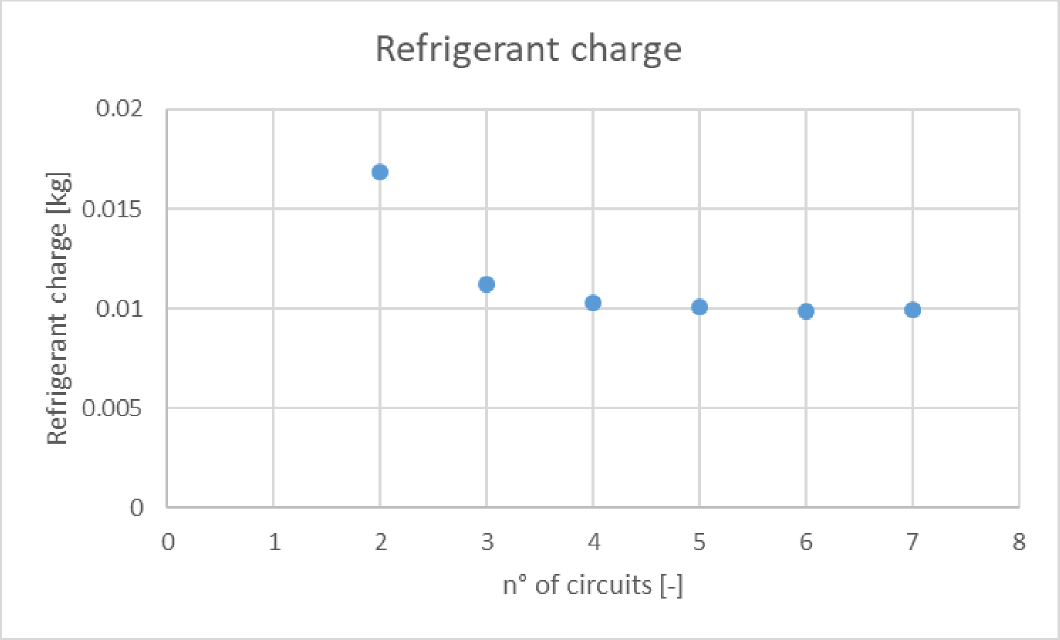
We note a similar trend to the previous ones analysed, with a maximum of 41.5 kPa in correspondence of the 2 circuits that goes to diminish with the increasing number of circuits.

Even the charge seems to have a more or less constant trend with a peak at the 7 circuits

T_cond remains constant in a range of 1.5 degrees (52.9°C and 54.3°C).

Heating - Evaporator





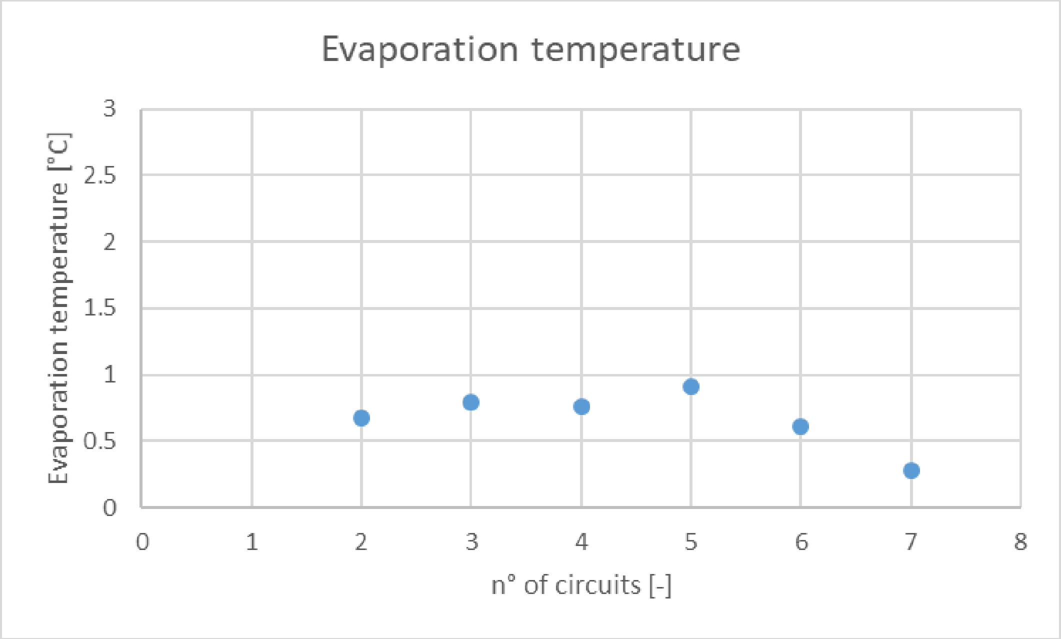


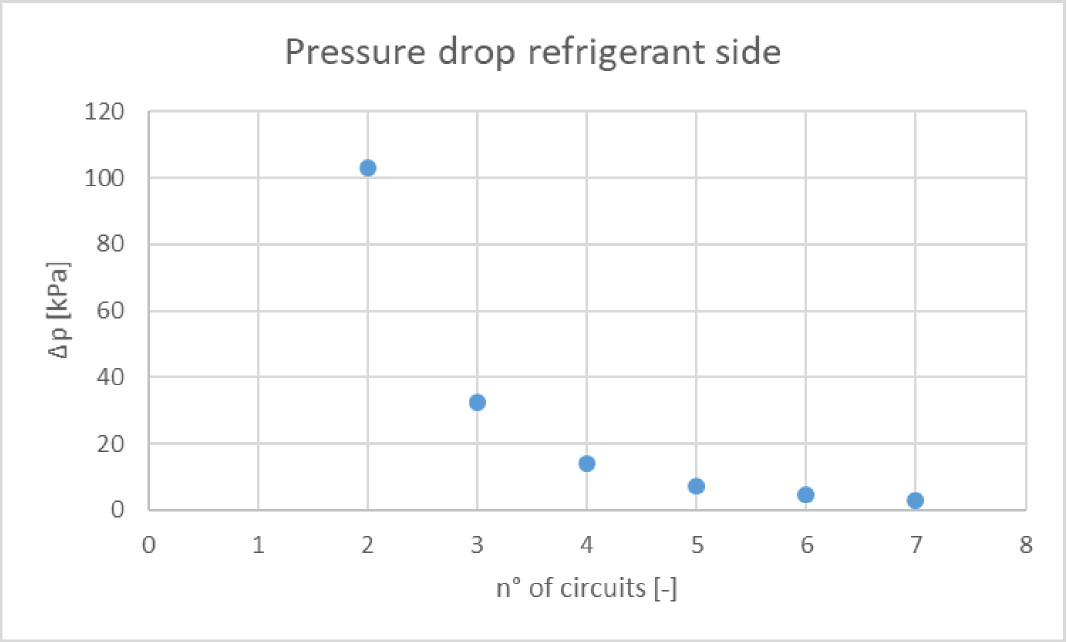
Figure 35:Pressure drop, refrigerant charge and evaporation temperature for different n° of circuits for 5mm source heat exchanger

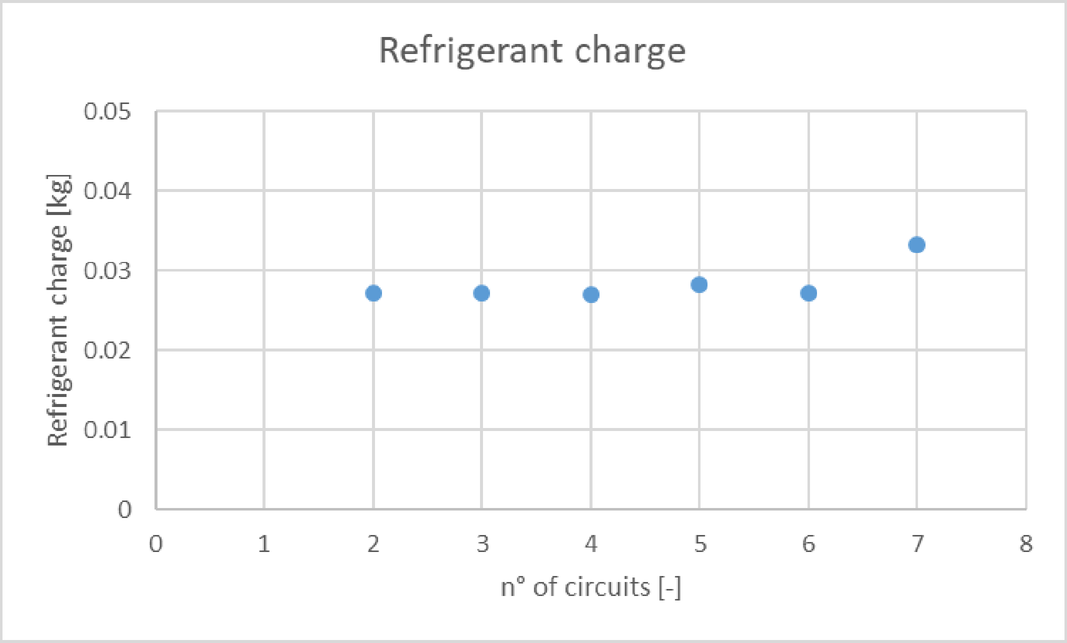
The pressure drop has a similar trend to those analysed above.

The refrigerant charge in this graph has a maximum in the 2 circuits and the minimum in the 7 circuits varying therefore between 16.8g and 9.9g. T_evap varies in the range of 0.3-0.9°C.

Source heat exchanger 4mm

Cooling-Condenser





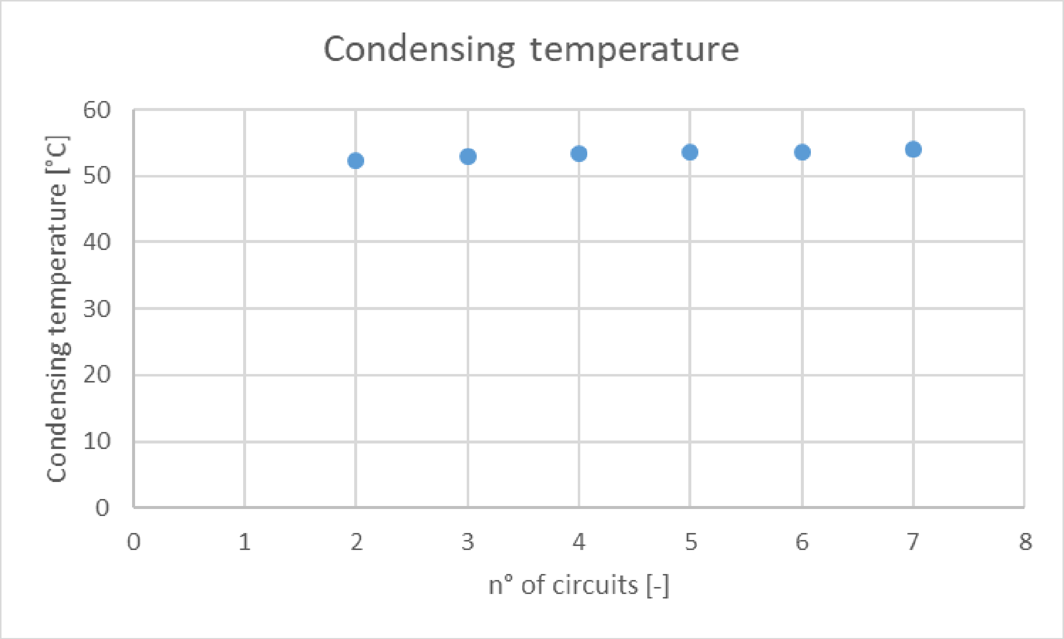
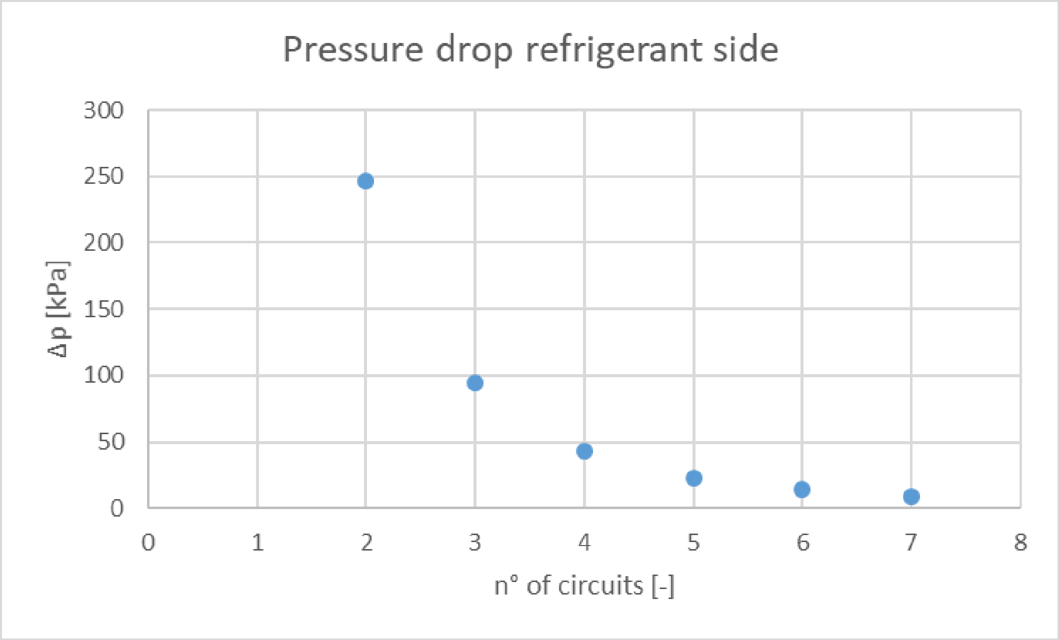
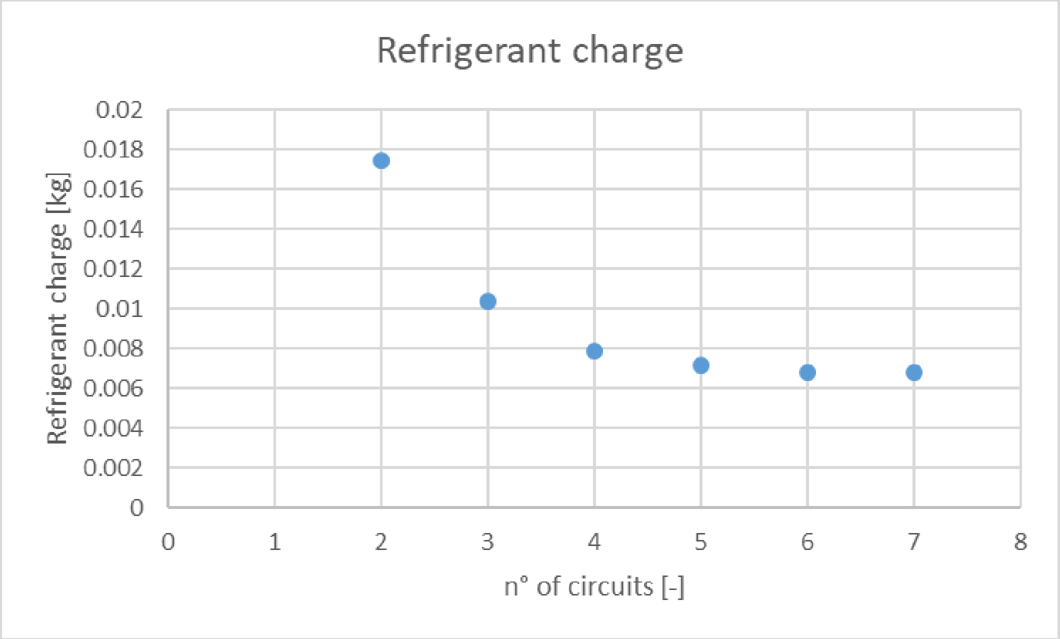


Figure 36: Pressure drop, refrigerant charge and condensing temperature for different n° of circuits for 4mm source heat exchanger

Heating-Evaporator





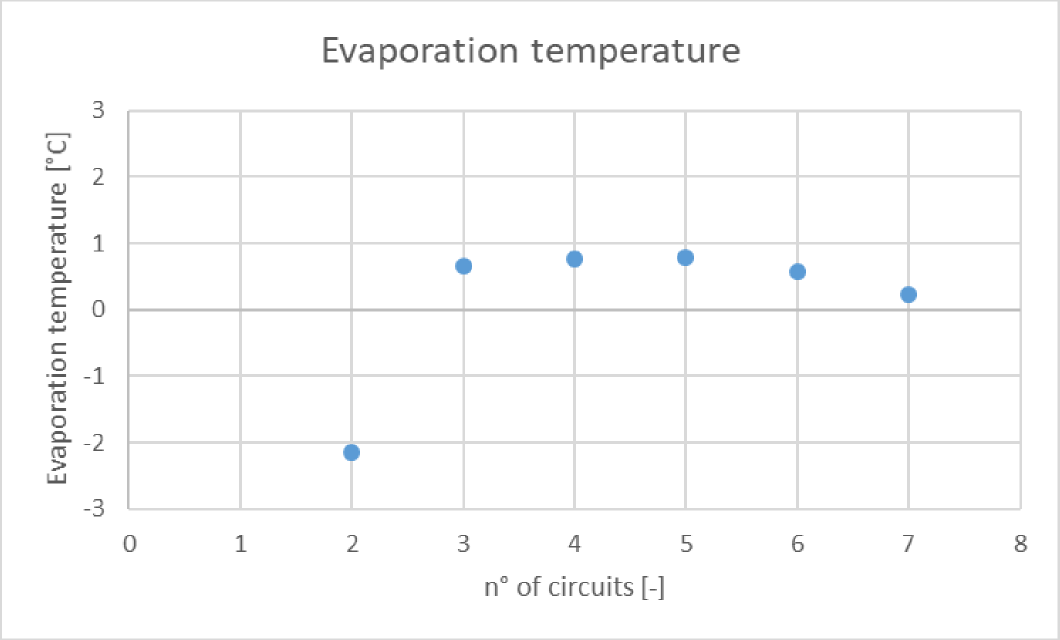
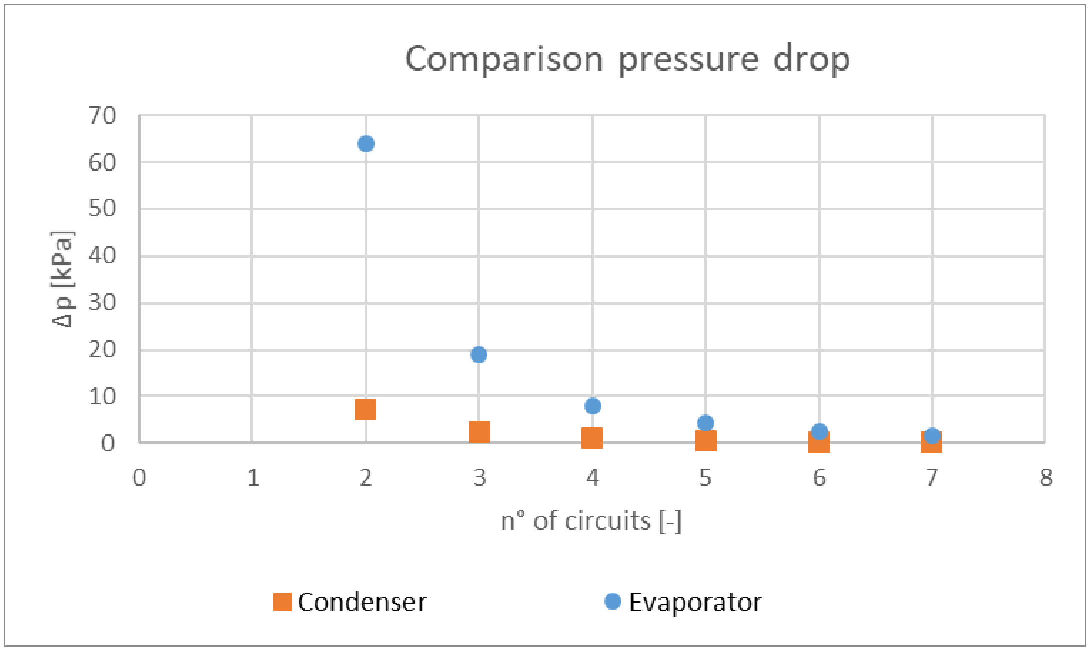


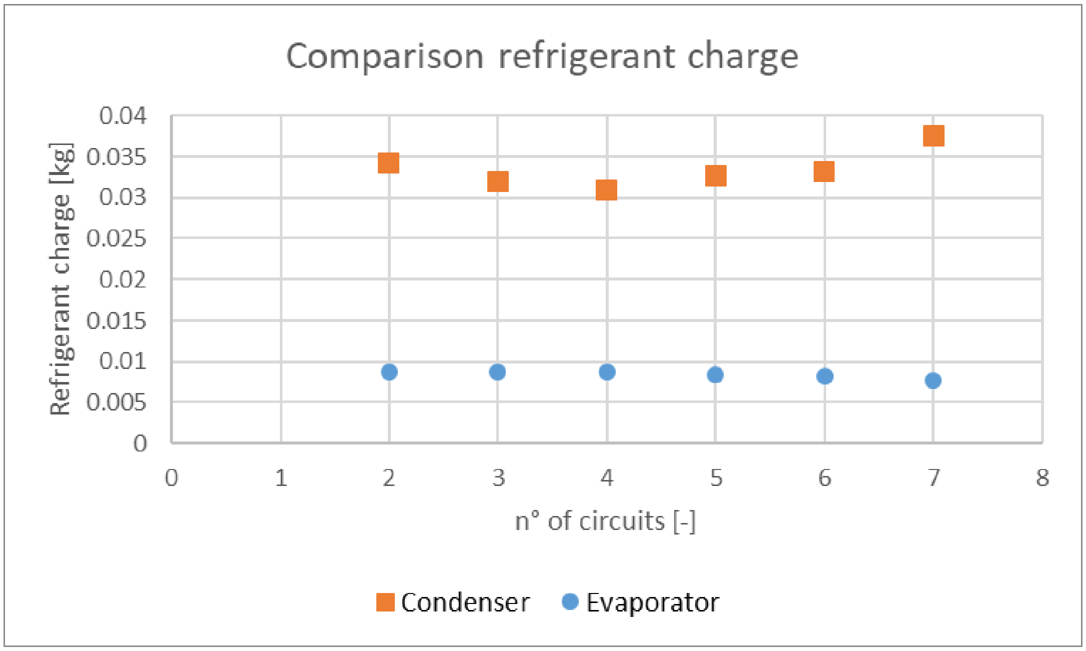
Figure 37: Pressure drop, refrigerant charge and evaporation temperature for different n° of circuits for 4mm source heat exchanger

We can see that, although they are different values compared to the cases at 5mm, the trends are similar. Then in this chapter, we will analyse the differences between the 4mm and 5mm configurations.

5.3.2 Evaporator and condenser comparison

<u>Treatment heat exchanger 5mm</u>





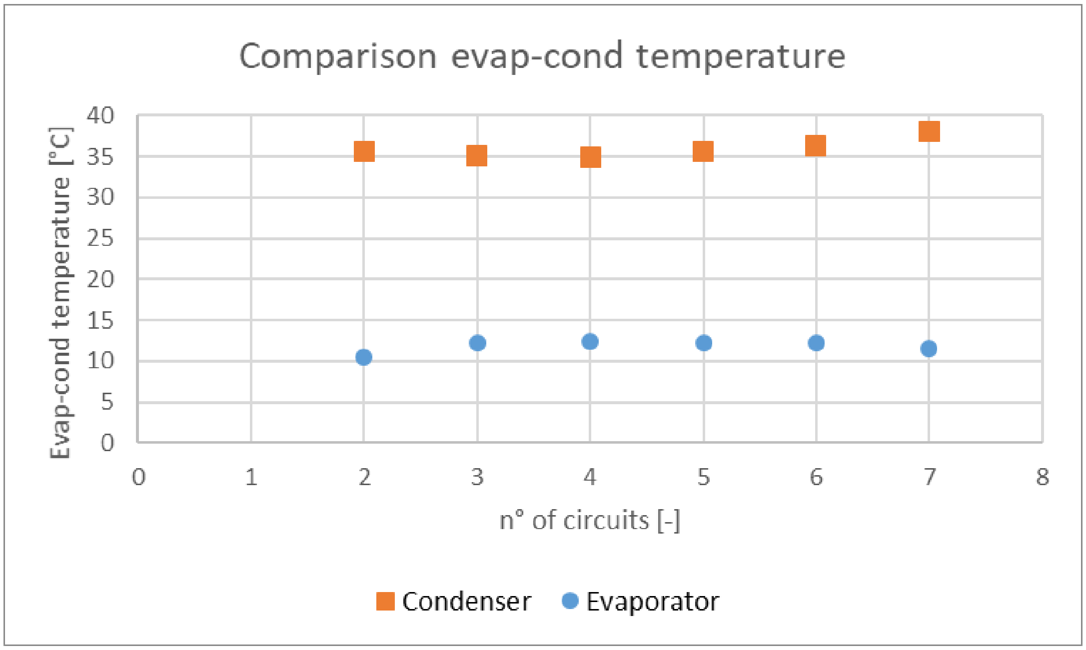


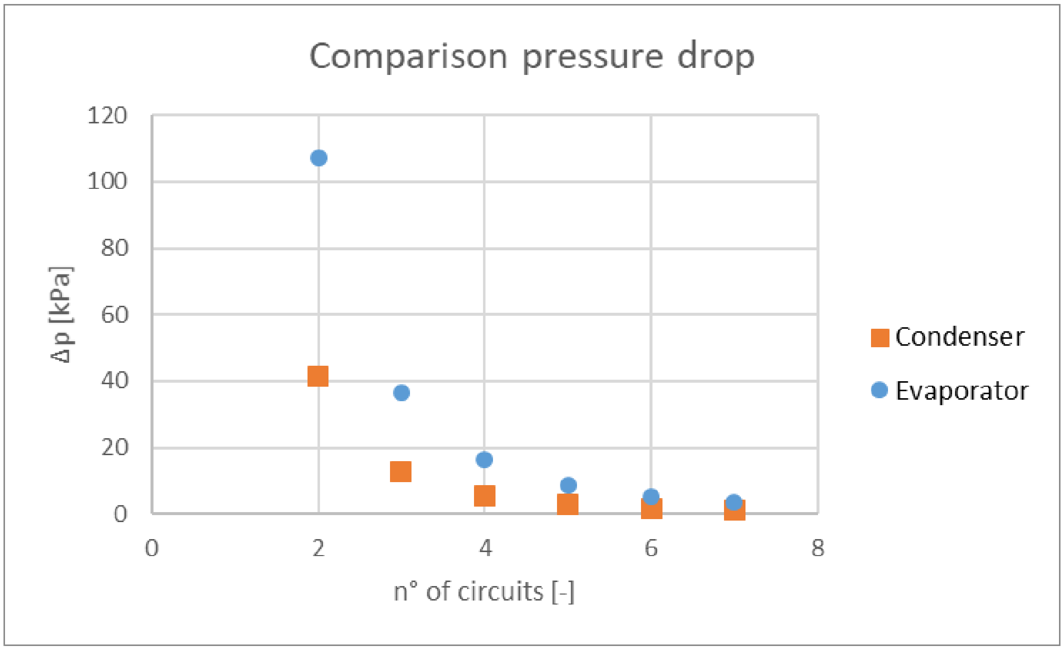
Figure 38: Condenser - evaporator 5mm treatment heat exchanger comparison

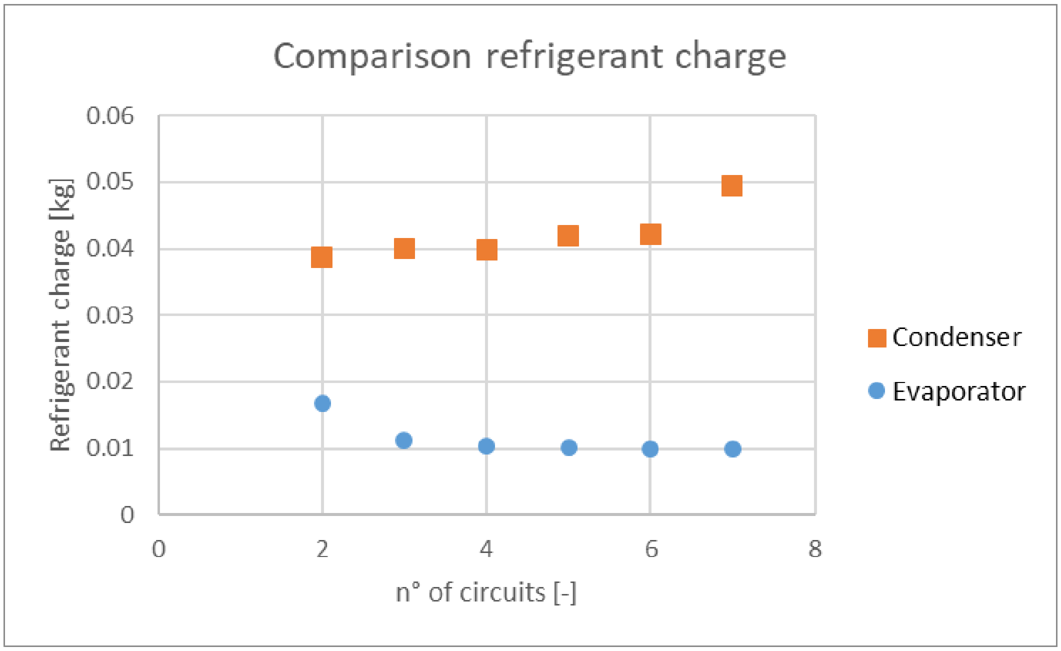
In these graphs, the points of the evaporator are represented in blue, while in orange the points related to the operation of the condenser for the heat exchanger treatment. As we can see the pressure drop in evaporator operation will be higher than in condenser operation. This is since during the evaporation phase the fluid is composed mainly of gas, which has a low density, which is linked to a high speed of flow and as already said the pressure is proportional to the square of the speed. Conversely, in condenser operation we know we have more liquid phase and this leads to a higher density and therefore lower speed and pressure drop than in the case of the evaporator. During sizing, priority was given to operation as an evaporator in both heat exchangers. The refrigerant side pressure losses on which a heat exchanger is sized are those in evaporation this because the evaporation phase of a heat exchanger is the most critical phase between the two: remember that in the circuit not only the refrigerant fluid flows but also the oil from the compressor. If the refrigerant cannot drag the oil to the circuit until the compressor is returned, wear and tear of the mechanical components of the compressor can occur until, in the worst case, the rupture of the compressor. In condensation, this problem usually does not exist because the liquid phase has no difficulty in dragging the oil due to its higher density than the gas phase. So I simulate both operations, but the decision is made on evaporating Δp values.

As we can see from the comparison on the refrigerant charge between evaporator and condenser, when I simulate the operation as condenser it will require more refrigerant charge, while in the operation by condenser will be less. The no-costant density of the fluid in the two operations can explain this deviation:

- In condensation, there will be more liquid phase, which implies having a medium density fluid higher. To parity of scope I will have therefore a greater charge
- In evaporation there will be more gas phase with lower average density than in the previous case and therefore, with the same refrigerant flow rate, less charge.

Source heat exchanger 5mm





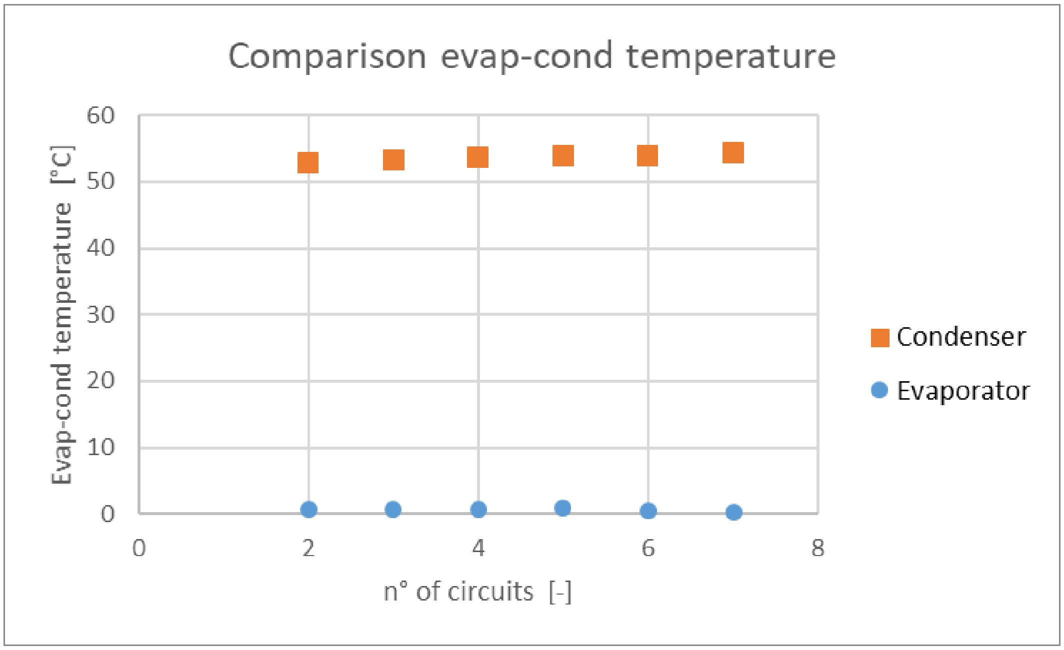


Figure 39: Condenser - evaporator 5mm source heat exchanger comparison

As we can see the reasoning conducted for the treatment heat exchanger is confirmed by the graphs of the source exchanger. The trends related to the source heat exchanger in evaporator and condenser operation are similar to those already analysed for treatment.

We can notice a big difference in the maximum values of Δp of this exchanger that reaches a value of 107 kPa in the configuration to 2 circuits while in the heat exchanger the maximum reached was internal to 64kPa. These pressure losses can be justified by the fact that the two heat exchanger are different from each other. In fact the source exchanger have twice the ranks (6 ranks against the 3 of the treatment) which implies an increase in the length of the tubes along which the fluid will have to flow, remember that the losses are also proportional to the length of the pipe according to the above formula and reported below:

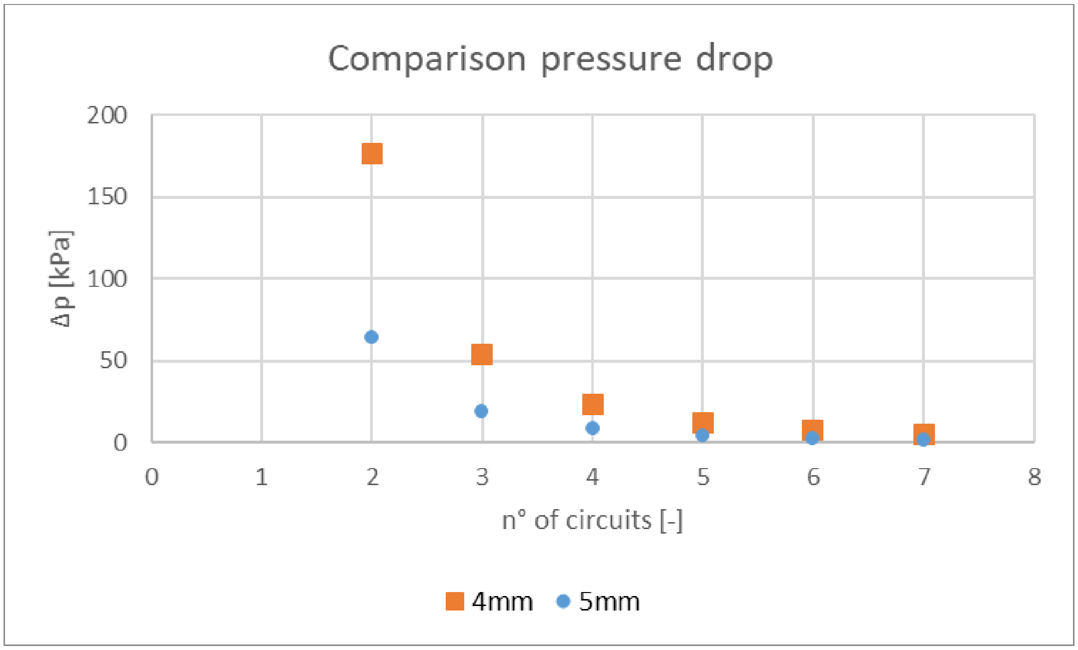
$$\Delta p = \frac{64}{Re}\frac{L}{D}\frac{w^2}{2}\rho$$

5.3.2 Comparison 5mm and 4mm geometries

The simulations performed on the two heat exchangers were compared in the configuration with a diameter of 5 mm and 4 mm.

Treatment heat exchanger

Evaporator



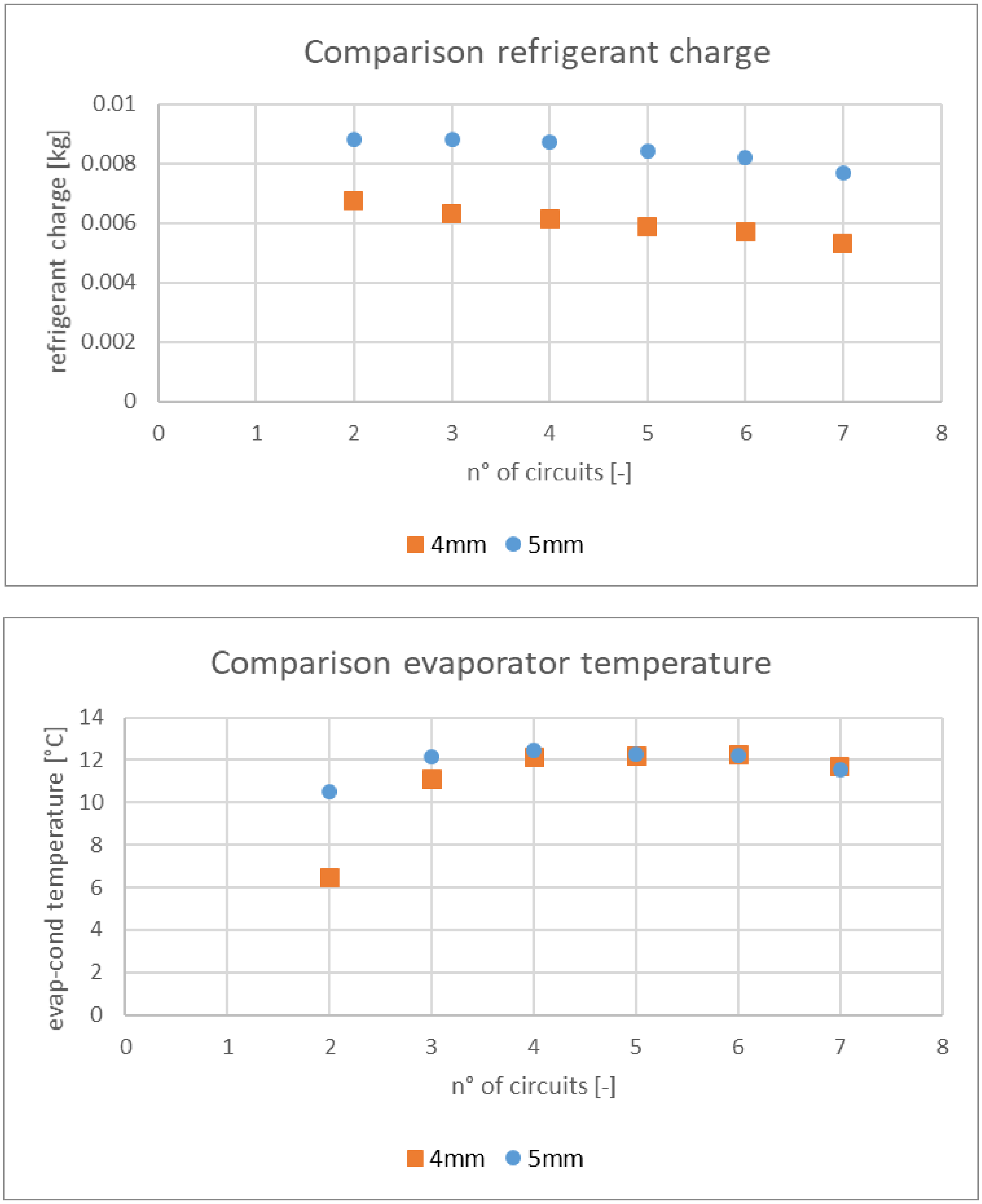


Figure 40: 5mm - 4mm evaporator treatment heat exchanger comparison

The Δp varies considerably between configurations with 5mm diameter and 4mm. The variation of Δp can be justified by the following reasoning:

At the same refrigerant flow rate (m_ref), a smaller pipe diameter results in a higher speed and therefore a higher pressure drop.

As for the refrigerant charge we can say that if the diameter of the pipes will decrease then also the charge will be smaller, in fact the volume of 4mm will be smaller than that of 5mm which involves:

$$if\ \rho_{5mm} = \rho_{4mm}\ \left[\frac{kg}{m^3}\right]\ e\ V_{5mm}\ > V_{4mm}\ [m^3]$$

And the refrigerant charge can be expressed as:

$$Refrigerant\ charge = \ \rho V$$

So:

$$(\ \rho V)_{5mm} > (\ \rho V)_{5mm}$$

So in general we can summarize in a table the main differences between 5 mm and 4 mm configuration that can help to easly understand the key concept:

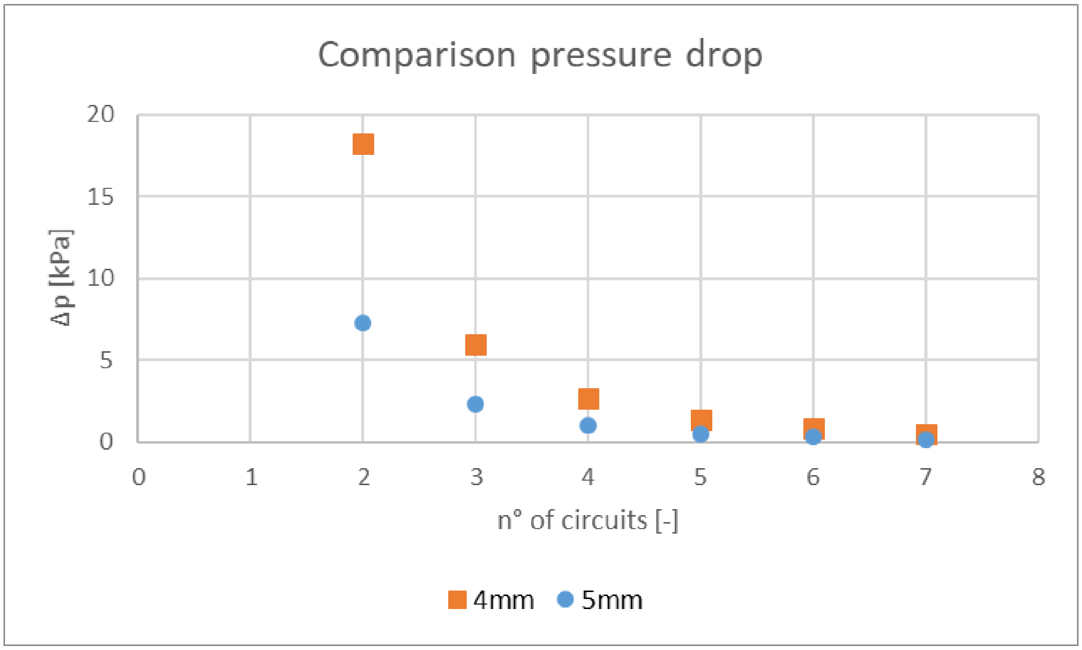
	5 mm	4 mm
Pressure drop refrigerant side	**-**	+
Refrigerant charge	+	**-**

Table 17: General comparison 5mm - 4mm

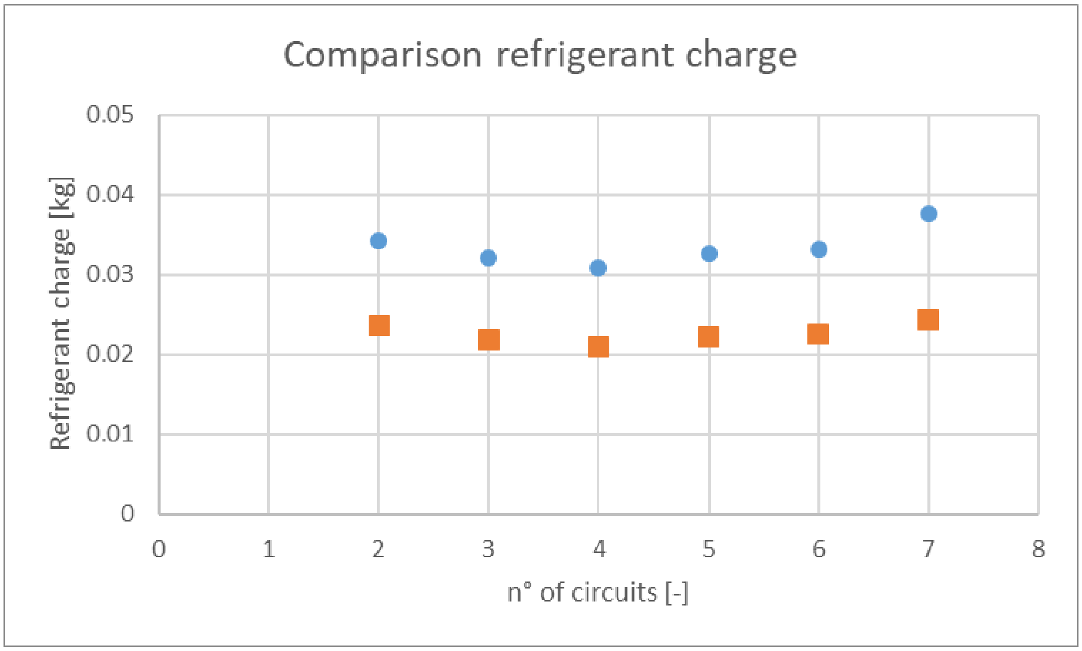
Where the "+" means greater than the other configuration and "-"means smaller.

As we can see, the differences arise mainly on the refrigerant side pressure drop and on the refrigerant charge. While the evaporation temperatures remain close to the same except for the point, relating to the 2 circuits where the deviation is considerable, of about 4 °C, and is relevant to the above considerations on the diameter and velocities of the fluid.

Condenser



Comparison pressure drop
Δp [kPa]
20
15
10
5
0
0
1
2
3
4
5
6
7
8
n° of circuits [-]
■ 4mm
● 5mm



Comparison refrigerant charge
Refrigerant charge [kg]
0.05
0.04
0.03
0.02
0.01
0
0
1
2
3
4
5
6
7
8
n° of circuits [-]

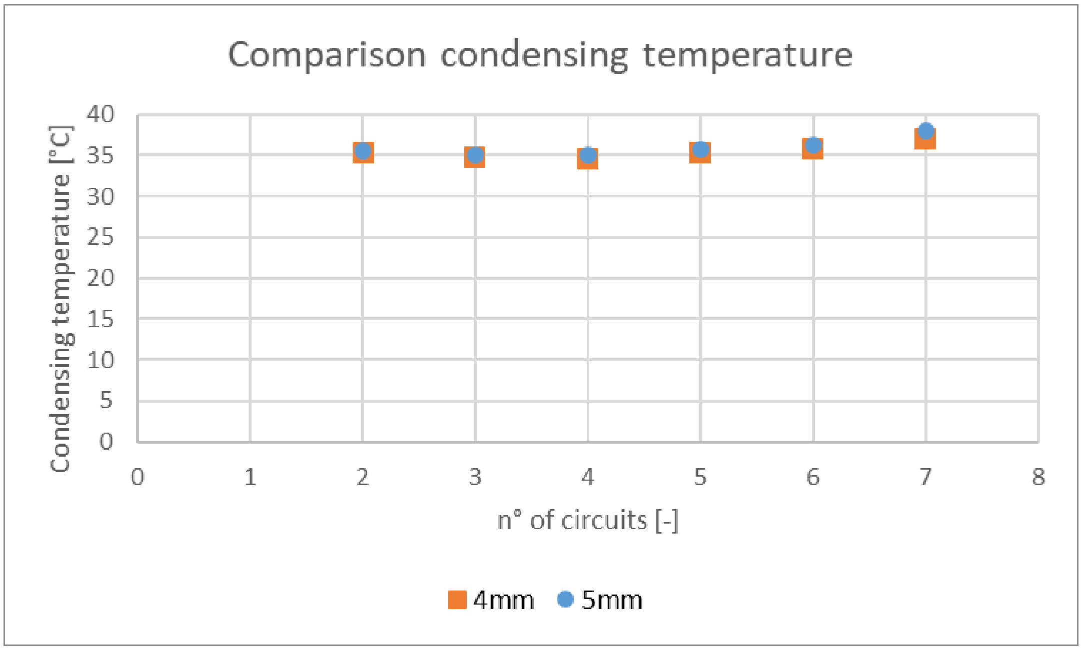
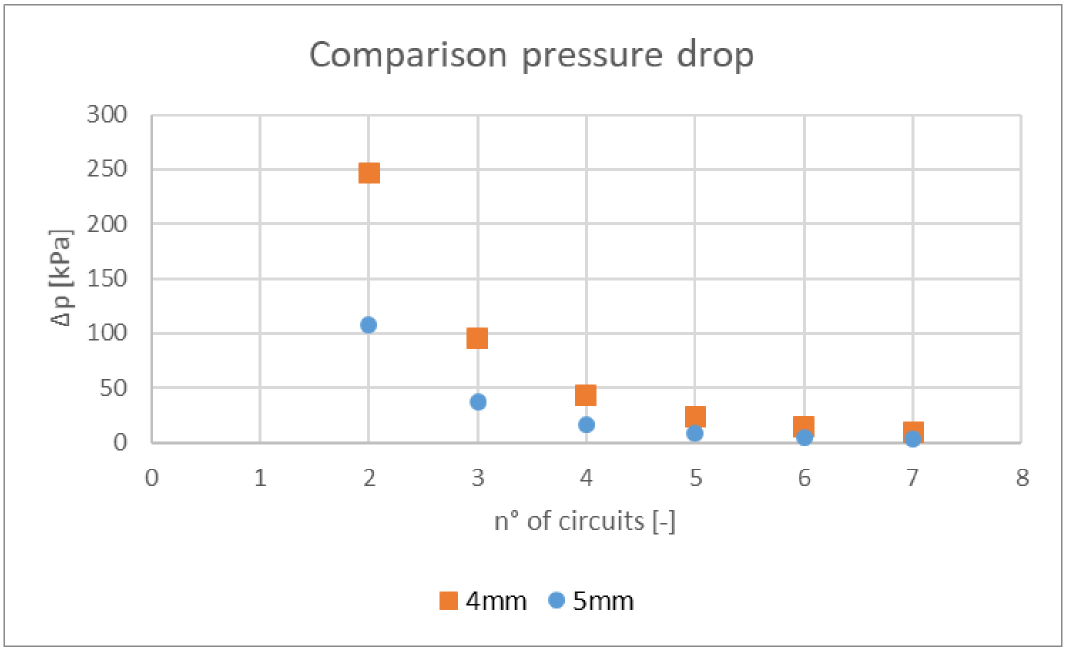


Figure 41: 5mm - 4mm condenser treatment heat exchanger comparison

As we can see from the above observed deviation on the evaporation during evaporator operation, in condensation temperatures it is not evident, in fact a fairly precise overlap of the points is noted. The trends of Δp and charge are similar and the above considerations apply.

Source heat exchanger

Evaporator



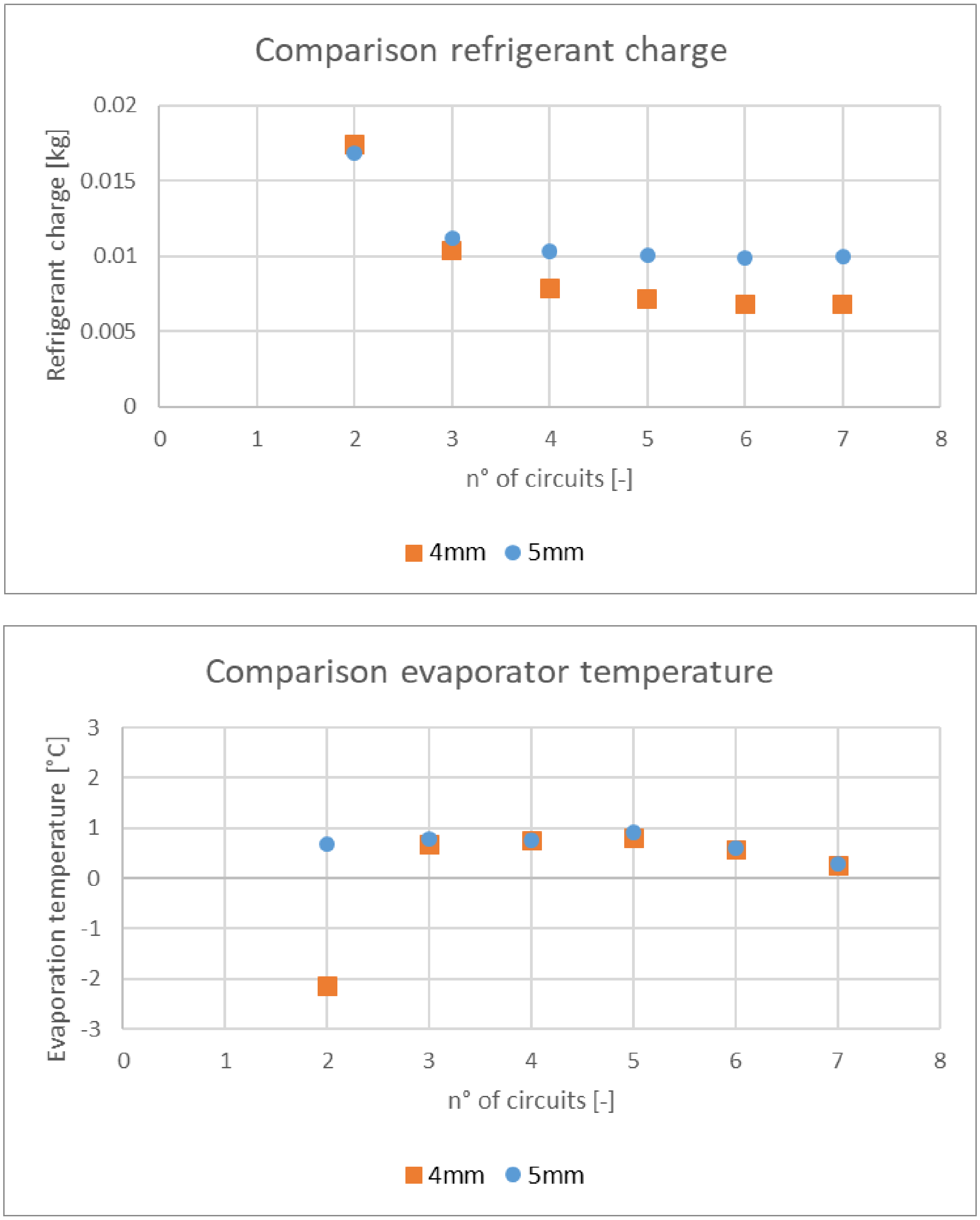
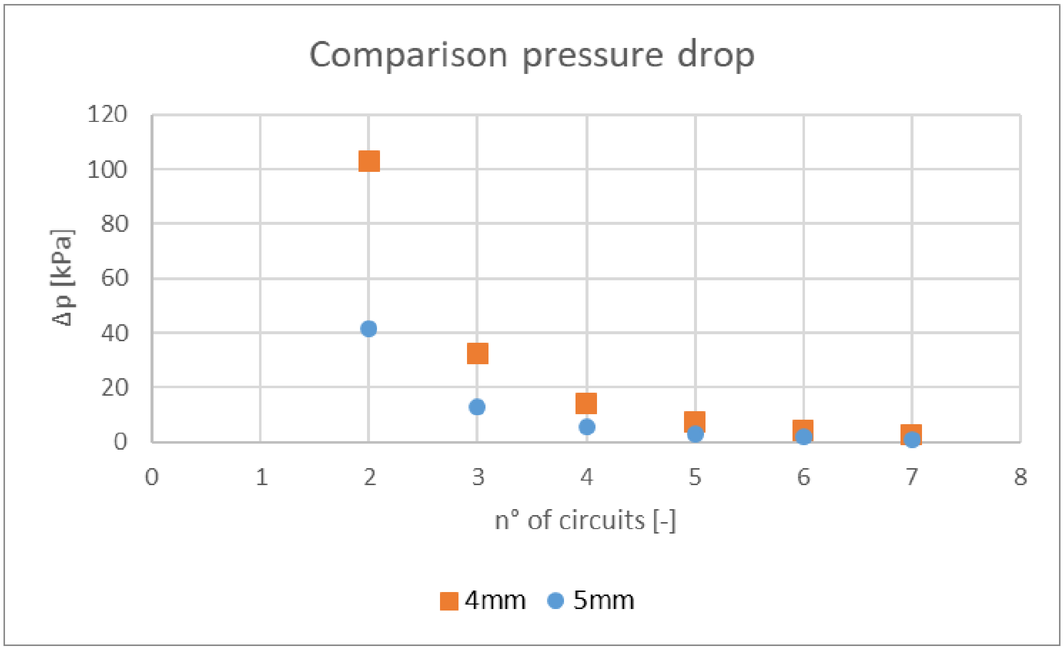
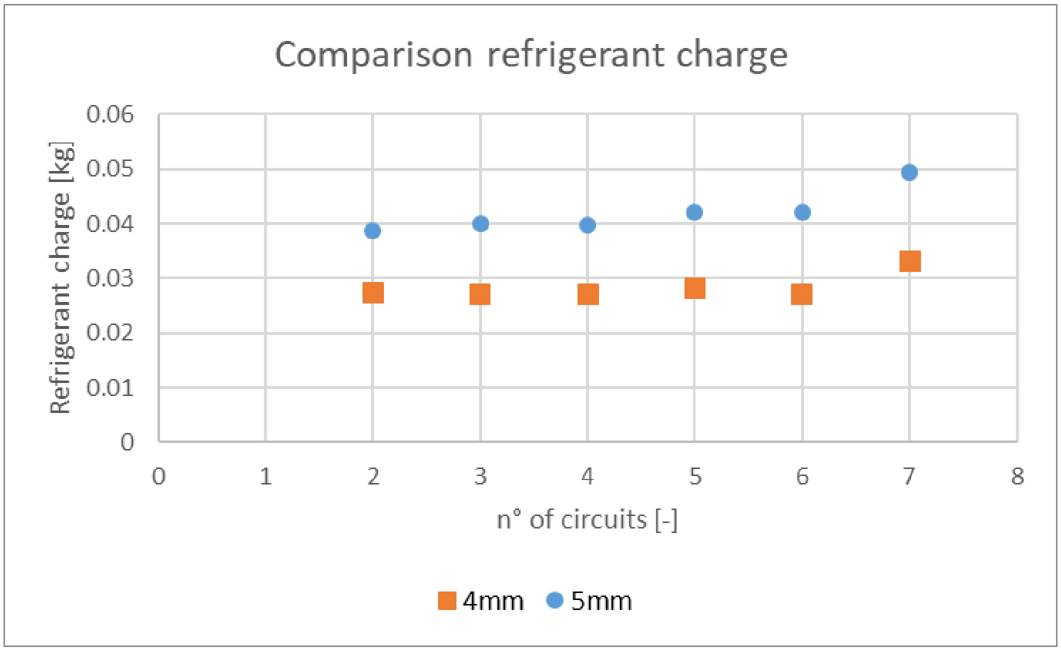


Figure 42: 5mm - 4mm evaporator source heat exchanger comparison

Again, we note that the point of the T_evap related to the configuration at 2 circuits 4mm differs from the 5 mm of about 3°C, while in the refrigerant charge graph we can see that the points relating to the 2 circuits and the 3 circuits are superimposed. This means that to have a benefit in the use of the 4 mm compared to the 5mm it will be necessary to select a heat exchanger with a number of circuits greater than three to obtain a reduction of refrigerant charge required.

Condenser


Comparison pressure drop
Δp [kPa]
120
100
80
60
40
20
0
0
1
2
3
4
5
6
7
8
n° of circuits [-]
4mm
5mm


Comparison refrigerant charge
Refrigerant charge [kg]
0.06
0.05
0.04
0.03
0.02
0.01
0
0
1
2
3
4
5
6
7
8
n° of circuits [-]
4mm
5mm

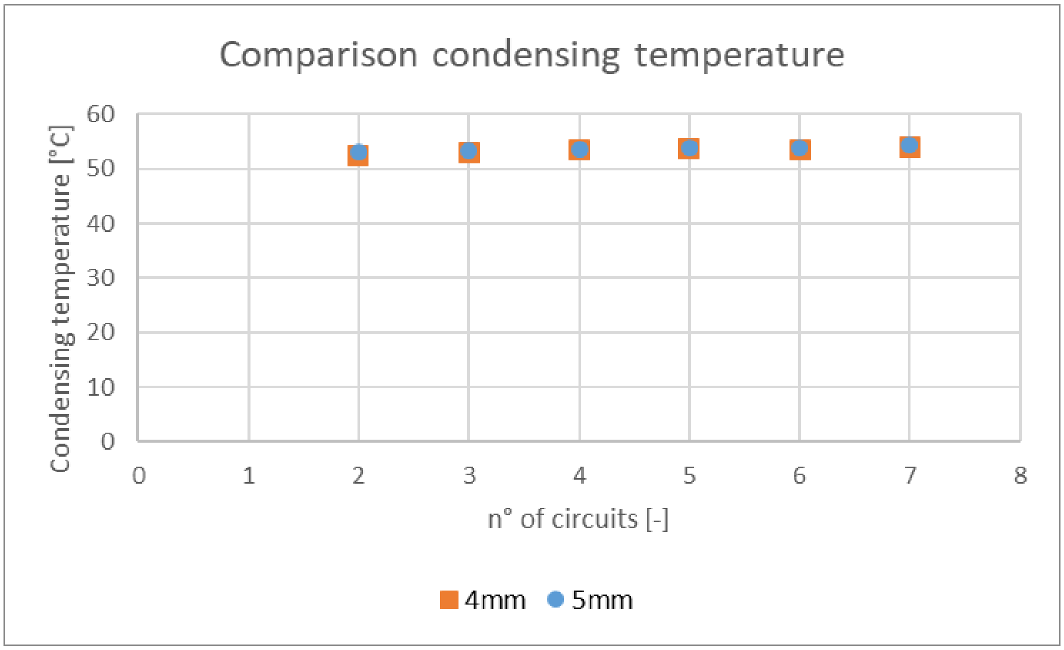


Figure 43: 5mm - 4mm condenser source heat exchanger comparison

As we can see, the same reasoning applies to the condenser operation of the treatment exchanger.

Summary observations of comparison analysis

From the two comparisons performed we can draw conclusions. In general, we can say that:

- Comparisons evaporator-condenser: the pressure drop will be greater in the evaporator operation because the amount of gas is greater, we have lower densities, higher speeds. The pressure drop is proportional to the square of the speed. In condenser operation, having more liquid phase, the fluid density will be higher, this results in lower speeds and therefore lower pressure drop. Regarding the refrigerant charge: the refrigerant charge can be expressed as the product of density and volume, in condenser operation, with the same volume of the pipes, the density is greater and therefore the refrigerant charge will be higher.
- Comparison 5-4mm: the pressure drop is higher in 4mm configurations because the pressure drop is inversely proportional to the diameter of the tubes. This means that the smaller the diameter, the greater the pressure drop at the same refrigerant flow rate. Regarding the refrigerant charge: at the same density of refrigerant fluid, the volume of the pipes is lower in configurations at 4mm and therefore the refrigerant charge will also be lower.

5.3.3 Practical limitation

From the graphs obtained from the simulations carried out through the software Coildesigner we can notice that the considerations executed are purely theoretical, In order to be able to compare the theoretical data with the practical data and the theoretical choice of heat exchangers, we must now introduce practical considerations that have hitherto been neglected.

5.3.3.1 Minimum speed for oil return

During the simulations, the heat exchangers were considered only as a separate element, disconnected from any other component, but they are connected to other components such as the compressor that uses an oil that, together with the refrigerant, circulates through the circuit. One of the most critical challenge in designing a unit is the return of oil to the compressor, with for other refrigerants having higher densities than propane the problem does not arise.

For R290 several studies have been carried out to obtain an experimental equation that allows the minimum speed of propane to be calculated so that the oil can return to the compressor, please note that these types of studies are been conducted by some companies and not from research lab:

$$V_{min,gas} = \left(9.8\ \phi_D\ \left[\frac{\rho_{oil} - \rho_{gas}}{\rho_{gas}}\right]\right)^{0.5} \left[\frac{m}{s}\right]$$

Where:

- ϕ_D is the inner tube diameter [m]
- ρ_{oil} is the oil density [kg/m^3]
- ρ_{gas} is the density of the refrigerant [kg/m^3]

For this equation, it is necessary to know the densities of the refrigerant fluid and the oil used in the compressor that be potentially different from compressor to compressor. Remember that during operation, the density of the fluid changes along the cycle as it undergoes several processes, in the condenser we know to have more liquid phase and therefore usually the density therefore it does not turn out to be a problem in fact there are no problems of oil dragging and moreover the geometry of the condenser, with the fluid inlets placed at the top and the outputs at the bottom, facilitate the natural flow of oil. In the evaporator, we have a large part of the fluid that is in the steam phase and this involves having a lower refrigerant density.

Let's calculate the density values of the refrigerant on Excel integrated with REFPROP for the previously simulated heating and cooling conditions.

- Heating: T_cond=30°C, T_evap=5°C, T_dish=46.2°C, P_evap=1.75kW, P_cond=1.9kW, SH=SC=5K
- Cooling: T_cond=52°C, T_evap=11°C, T_dish=71.6°C,P_cond=2.5kW,P_evap=2kW, SH=SC=5K

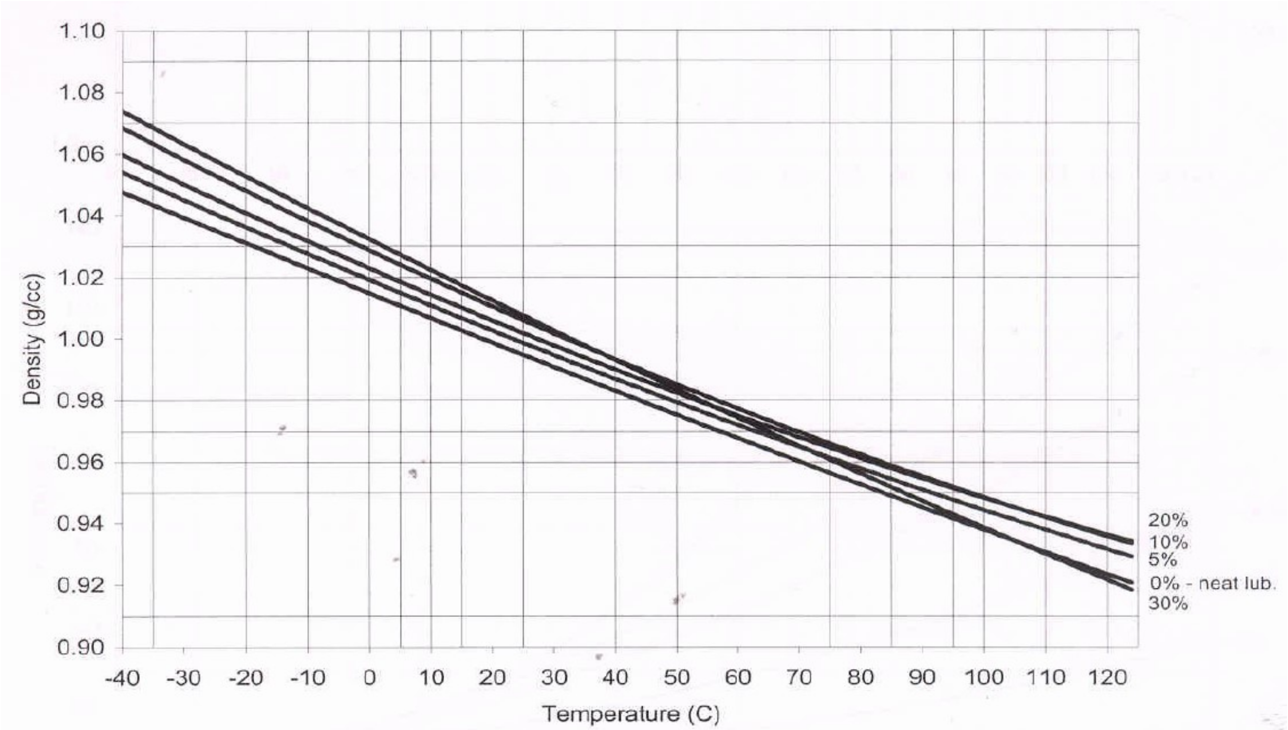
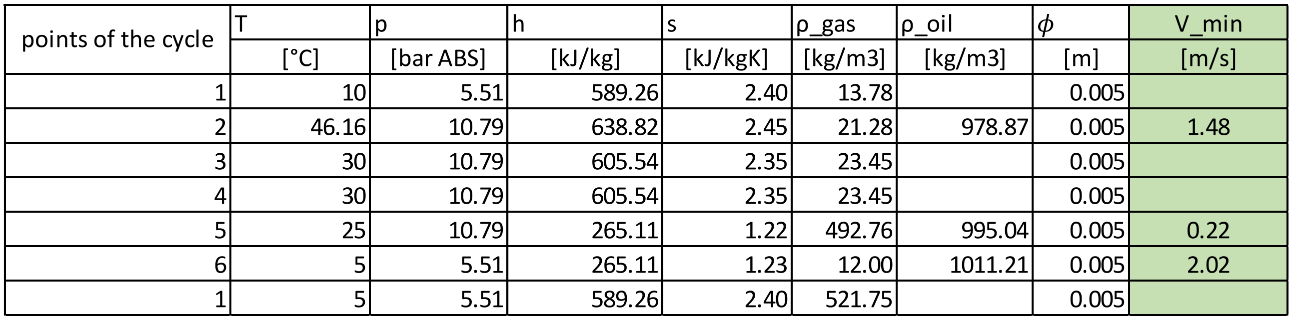


Figure 44: Temperature - oil density graph

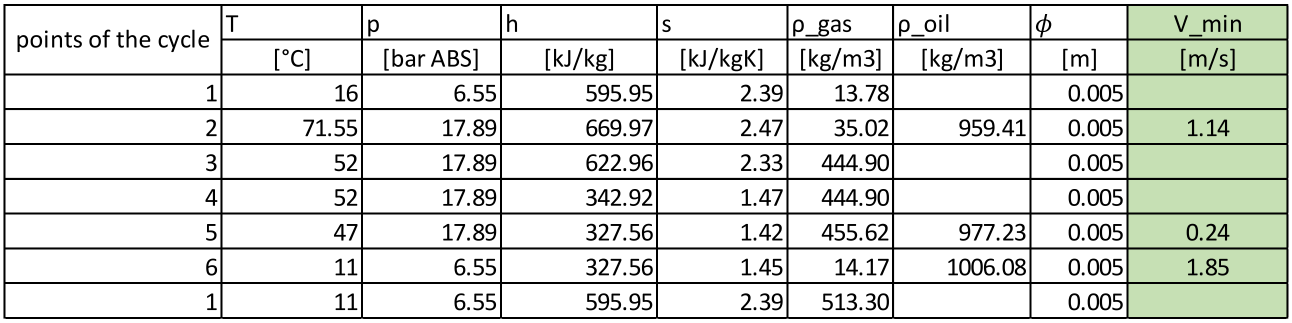
To determine the density of compressor oil, we use graphs provided by compressor manufacturers and we look for densities according to the temperatures of the point we want to consider. The study is carried out on the evaporators because the condenser will not have any difficulties related to the dragging of the oil being mostly liquid and the geometry is designed to make the refrigerant flow from top to bottom. In order to conduct this study and use the above formulas, we must decide how to take the density of the refrigerant. The density of the point 6 is complex to find because being inside the bell know to be a mixture composed of value of vapour and liquid with different densities between them. Being that the component that brings complications is the gaseous one, it is decided to use its density and to carry out a cautionary study assuming that all the refrigerant fluid has density equal to that of the gas phase.

Below the generated tables for operation in heating and cooling are reported thanks to which the results of refrigerant fluid speed will be compared to determine if and which geometries will be able to meet the minimum speed criterion for the return oil.



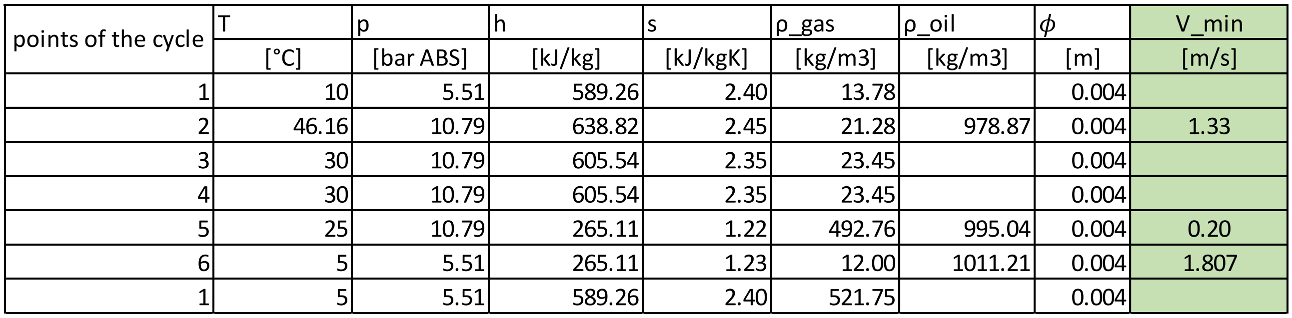
points of the cycle	T	p	h	s	ρ_gas	ρ_oil	ϕ	V_min
	[°C]	[bar ABS]	[kJ/kg]	[kJ/kgK]	[kg/m3]	[kg/m3]	[m]	[m/s]
1	10	5.51	589.26	2.40	13.78		0.005	
2	46.16	10.79	638.82	2.45	21.28	978.87	0.005	1.48
3	30	10.79	605.54	2.35	23.45		0.005	
4	30	10.79	605.54	2.35	23.45		0.005	
5	25	10.79	265.11	1.22	492.76	995.04	0.005	0.22
6	5	5.51	265.11	1.23	12.00	1011.21	0.005	2.02
1	5	5.51	589.26	2.40	521.75		0.005	

Figure 45: Min refrigerant velocity calculation 5mm source heat exchanger



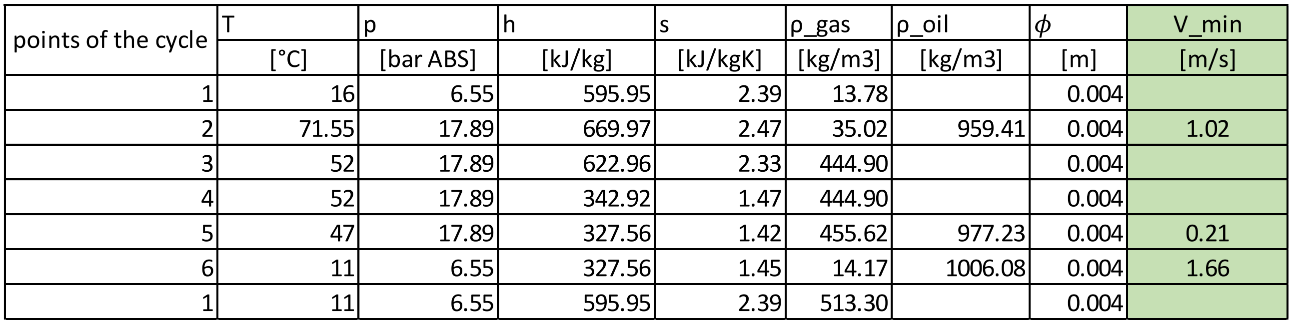
points of the cycle	T	p	h	s	ρ_gas	ρ_oil	ϕ	V_min
	[°C]	[bar ABS]	[kJ/kg]	[kJ/kgK]	[kg/m3]	[kg/m3]	[m]	[m/s]
1	16	6.55	595.95	2.39	13.78		0.005	
2	71.55	17.89	669.97	2.47	35.02	959.41	0.005	1.14
3	52	17.89	622.96	2.33	444.90		0.005	
4	52	17.89	342.92	1.47	444.90		0.005	
5	47	17.89	327.56	1.42	455.62	977.23	0.005	0.24
6	11	6.55	327.56	1.45	14.17	1006.08	0.005	1.85
1	11	6.55	595.95	2.39	513.30		0.005	

Figure 46: Min refrigerant velocity calculation 5mm treatment heat exchanger



points of the cycle	T	p	h	s	ρ_gas	ρ_oil	ϕ	V_min
	[°C]	[bar ABS]	[kJ/kg]	[kJ/kgK]	[kg/m3]	[kg/m3]	[m]	[m/s]
1	10	5.51	589.26	2.40	13.78		0.004	
2	46.16	10.79	638.82	2.45	21.28	978.87	0.004	1.33
3	30	10.79	605.54	2.35	23.45		0.004	
4	30	10.79	605.54	2.35	23.45		0.004	
5	25	10.79	265.11	1.22	492.76	995.04	0.004	0.20
6	5	5.51	265.11	1.23	12.00	1011.21	0.004	1.807
1	5	5.51	589.26	2.40	521.75		0.004	

Figure 47: Min refrigerant velocity calculation 4mm source heat exchanger

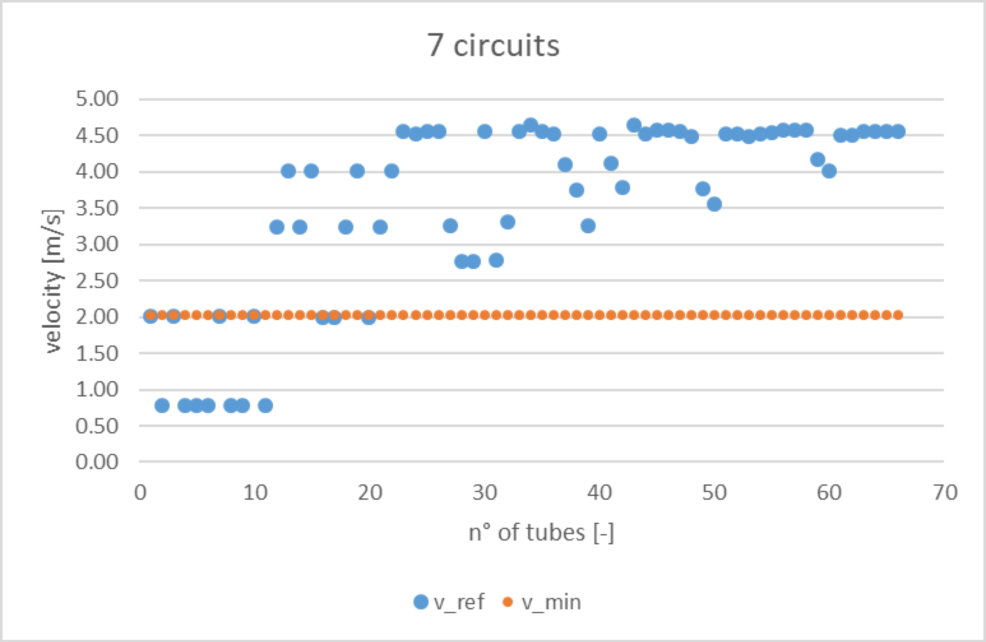


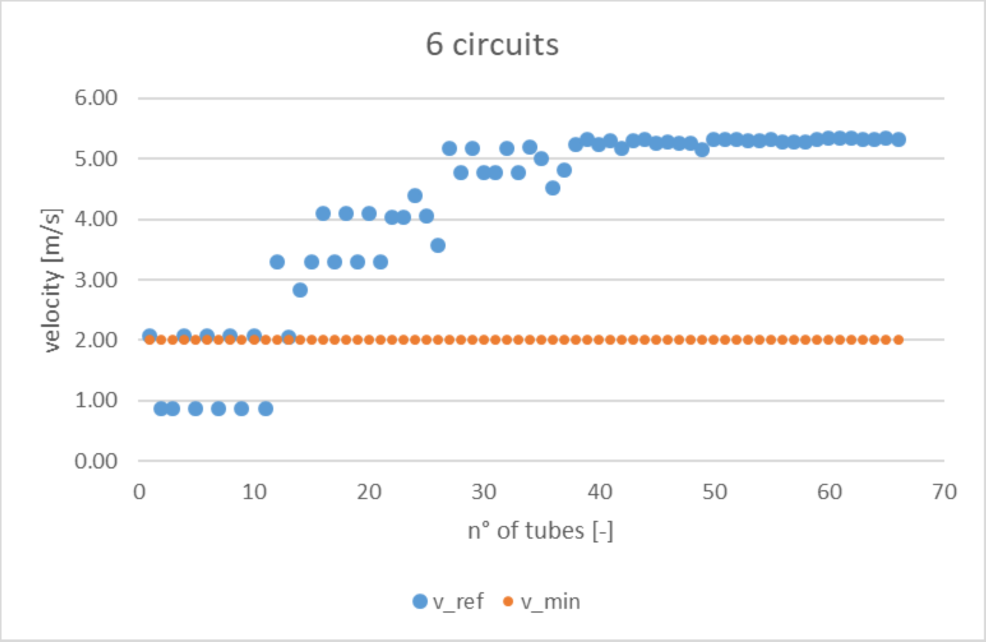
points of the cycle	T	p	h	s	ρ_gas	ρ_oil	ϕ	V_min
	[°C]	[bar ABS]	[kJ/kg]	[kJ/kgK]	[kg/m3]	[kg/m3]	[m]	[m/s]
1	16	6.55	595.95	2.39	13.78		0.004	
2	71.55	17.89	669.97	2.47	35.02	959.41	0.004	1.02
3	52	17.89	622.96	2.33	444.90		0.004	
4	52	17.89	342.92	1.47	444.90		0.004	
5	47	17.89	327.56	1.42	455.62	977.23	0.004	0.21
6	11	6.55	327.56	1.45	14.17	1006.08	0.004	1.66
1	11	6.55	595.95	2.39	513.30		0.004	

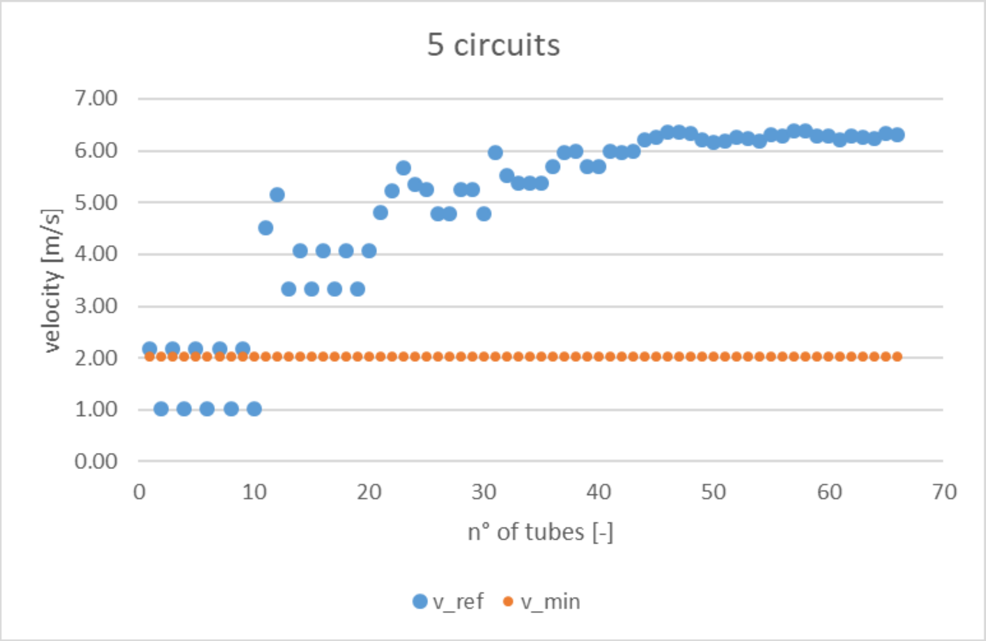
Figure 48: Min refrigerant velocity calculation 4mm treatment heat exchanger

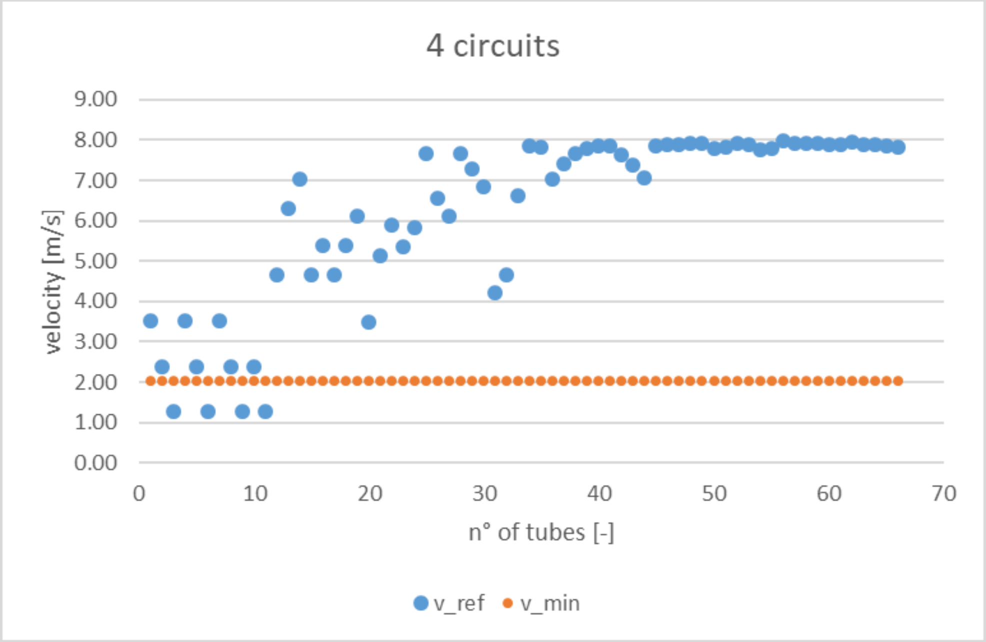
After that we repeated the above simulations, going to extract all the speeds that the software provides us if each tube of the circuit. Comparing the refrigerant speeds in the tubes with the minimum required speed for the calculated oil return we get the following graphs.

Source 5mm


7 circuits
velocity [m/s]
5.00
4.50
4.00
3.50
3.00
2.50
2.00
1.50
1.00
0.50
0.00
0
10
20
30
40
50
60
70
n° of tubes [-]
● v_ref
● v_min


6 circuits
velocity [m/s]
6.00
5.00
4.00
3.00
2.00
1.00
0.00
0
10
20
30
40
50
60
70
n° of tubes [-]
● v_ref
● v_min


5 circuits
7.00
6.00
5.00
4.00
3.00
2.00
1.00
0.00
velocity [m/s]
0
10
20
30
40
50
60
70
n° of tubes [-]
● v_ref ● v_min


4 circuits
9.00
8.00
7.00
6.00
5.00
4.00
3.00
2.00
1.00
0.00
velocity [m/s]
0
10
20
30
40
50
60
70
n° of tubes [-]
● v_ref ● v_min

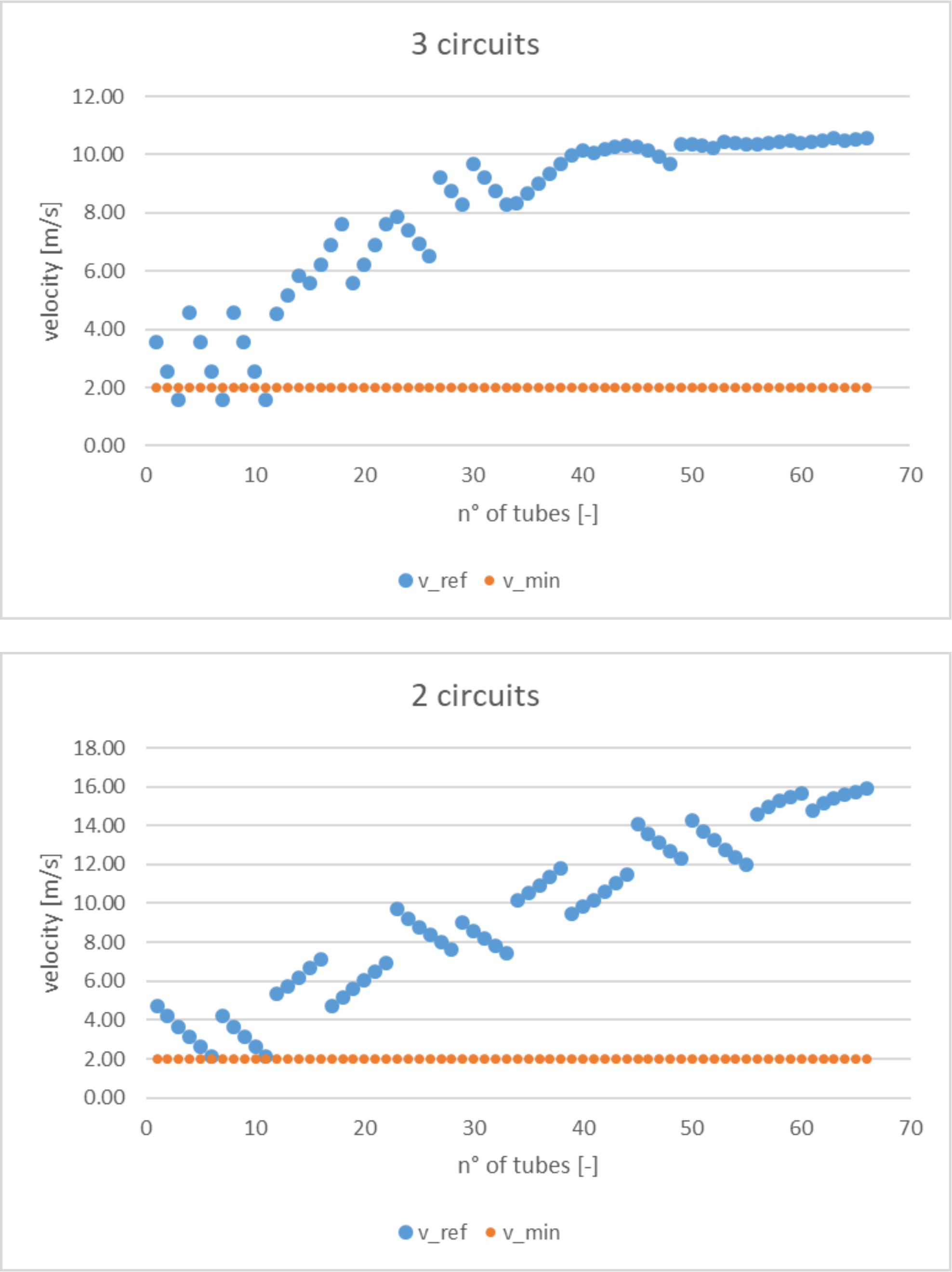
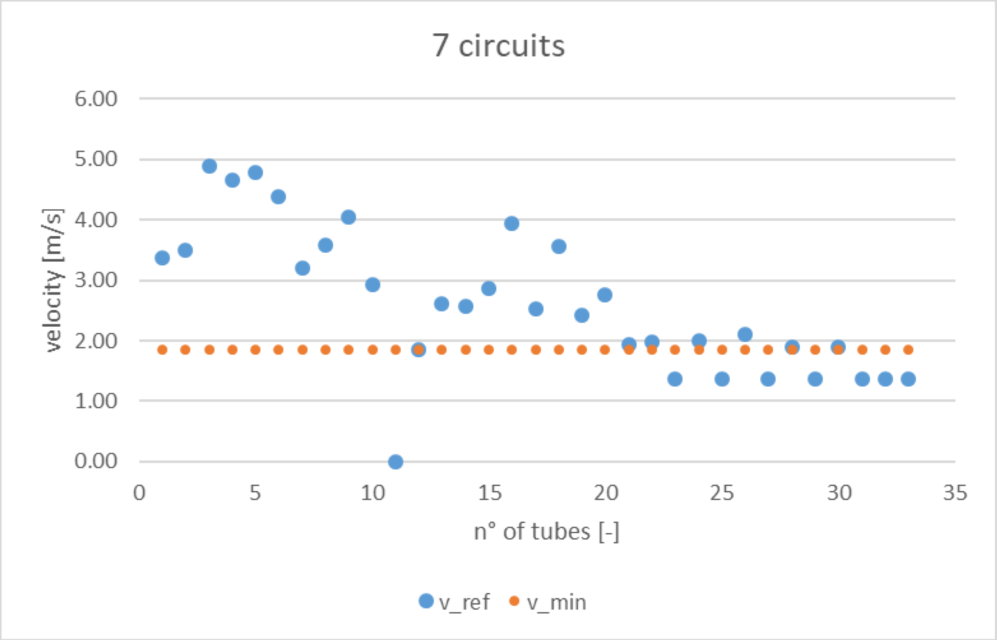
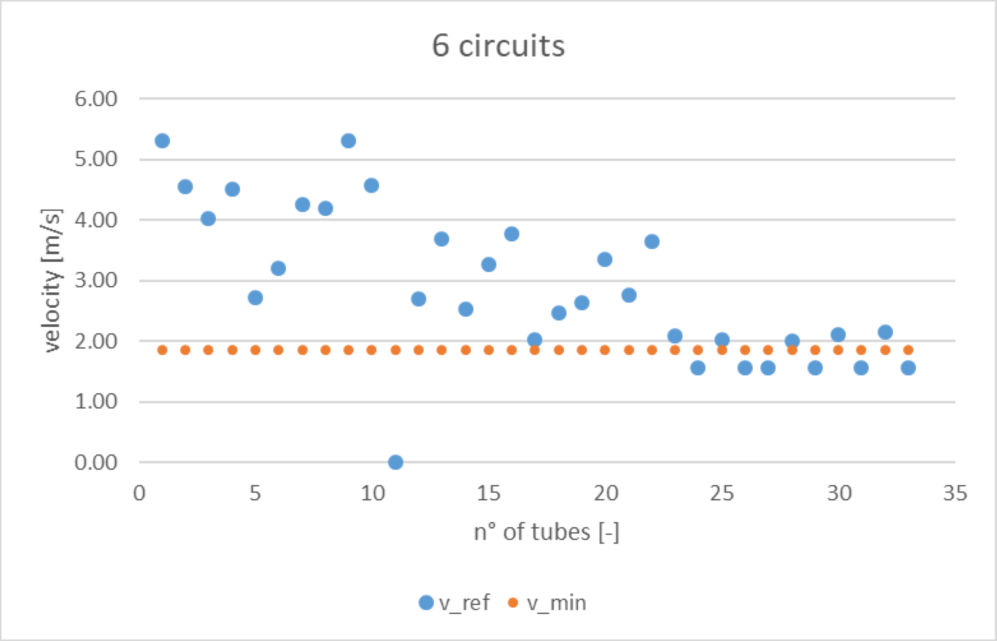
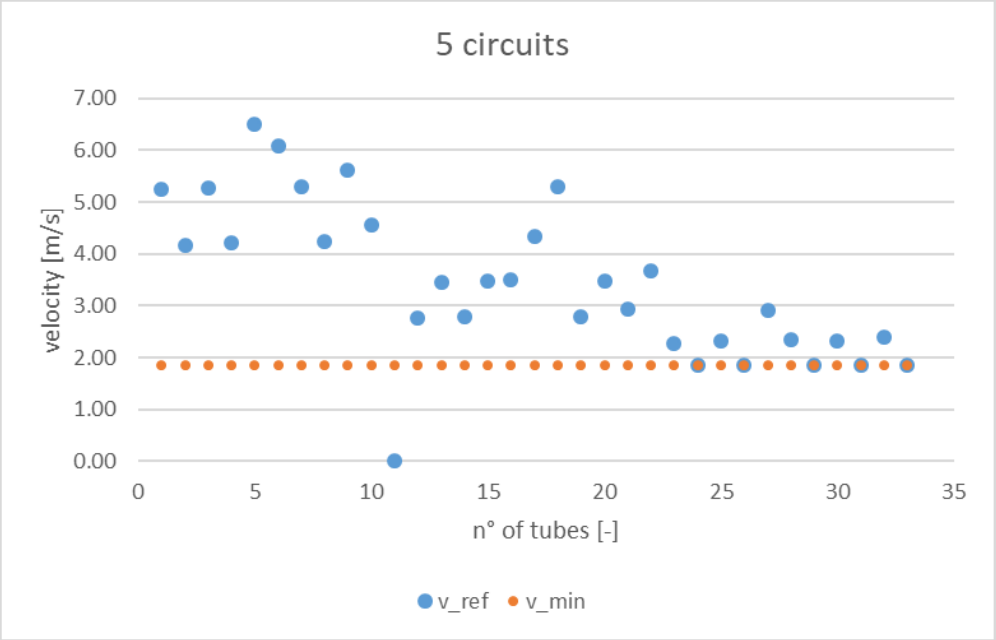


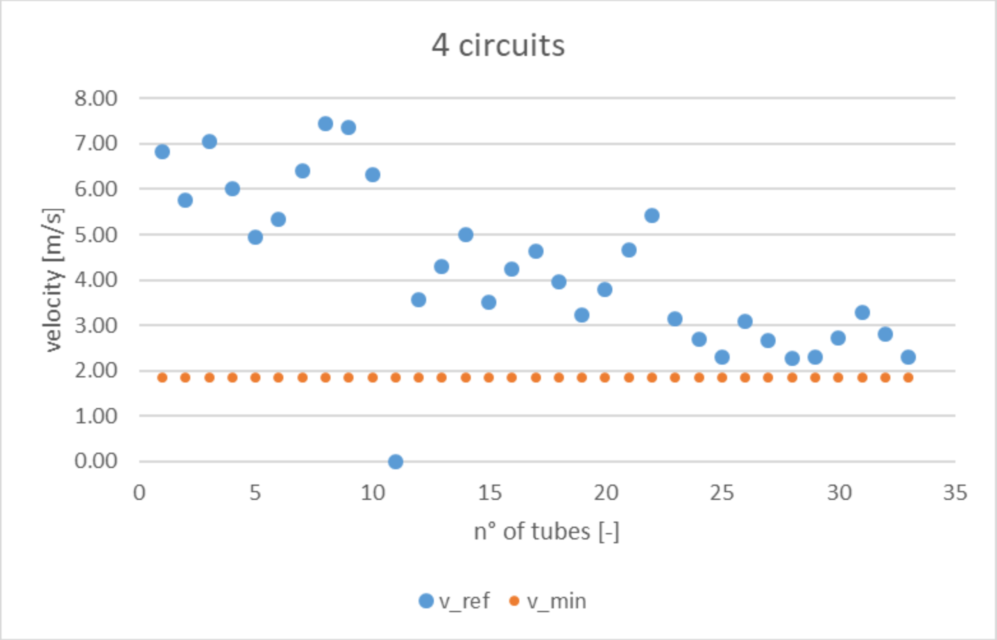
Figure 49: refrigerant and oil velocity for different n° of circuits for 5mm source heat exchanger

Treatment 5mm






5 circuits
velocity [m/s]
7.00
6.00
5.00
4.00
3.00
2.00
1.00
0.00
0
5
10
15
20
25
30
35
n° of tubes [-]
● v_ref
● v_min


4 circuits
velocity [m/s]
8.00
7.00
6.00
5.00
4.00
3.00
2.00
1.00
0.00
0
5
10
15
20
25
30
35
n° of tubes [-]
● v_ref
● v_min

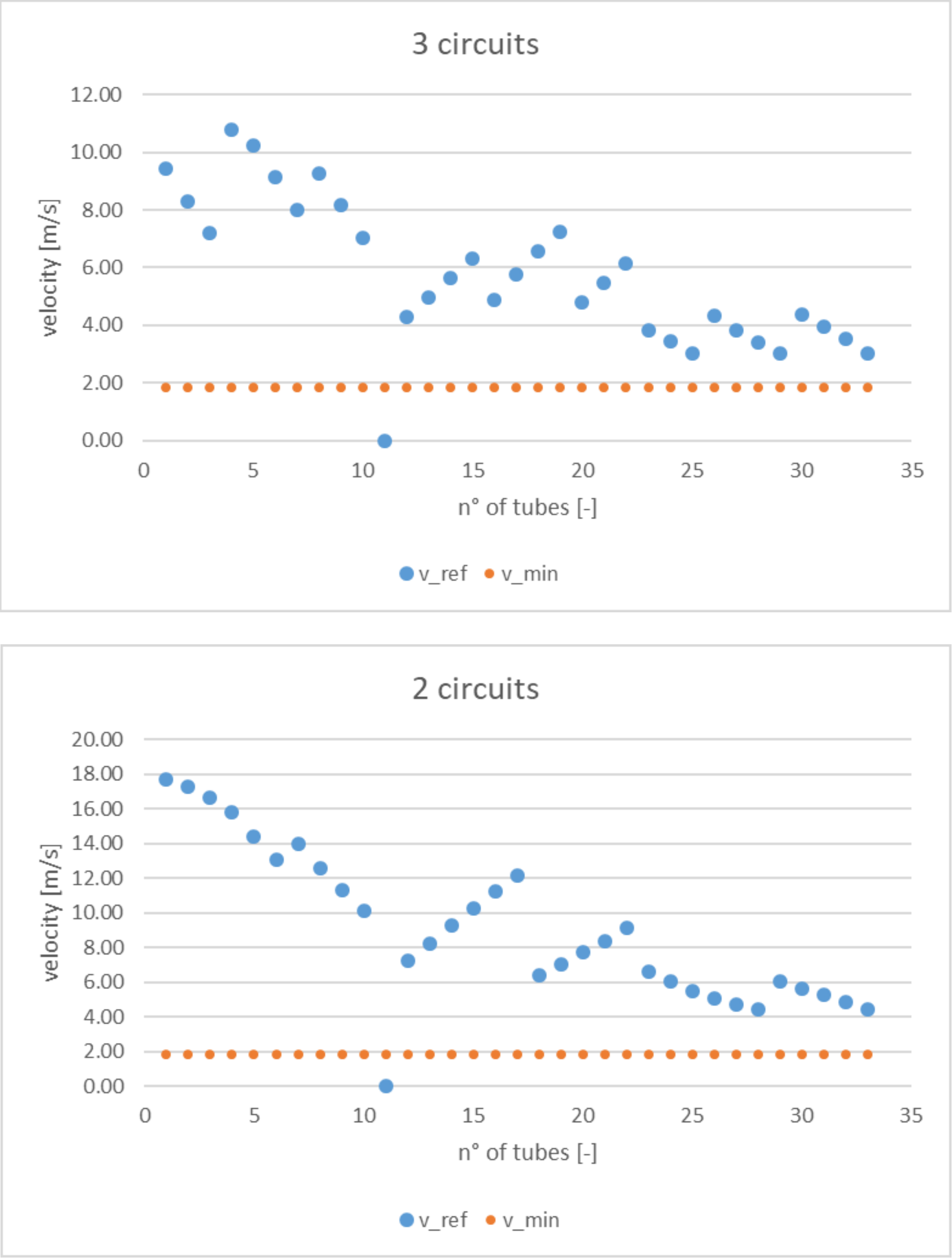
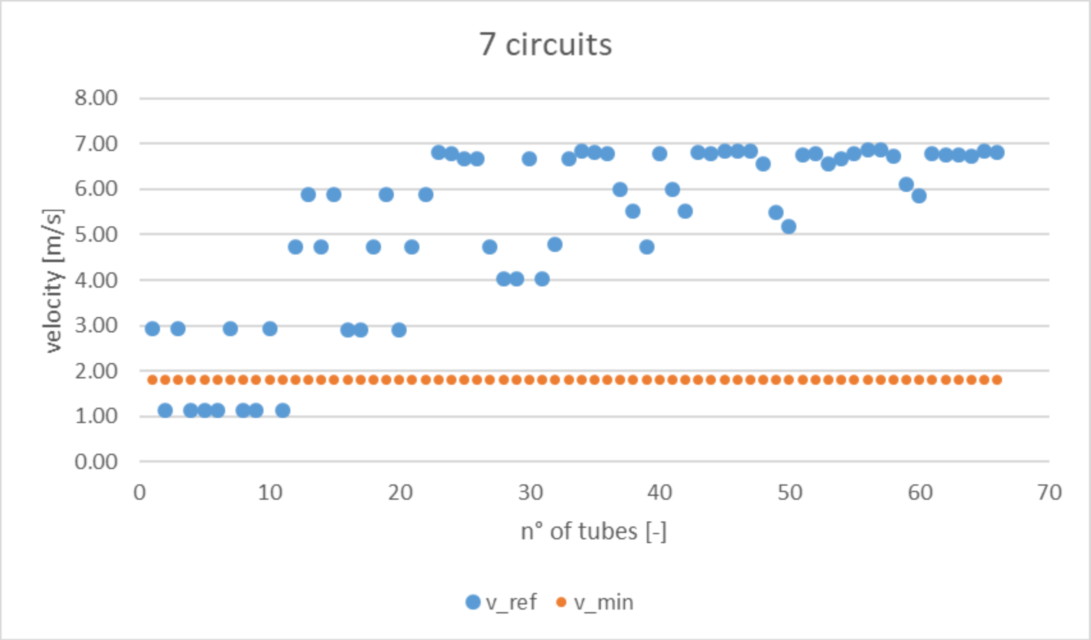
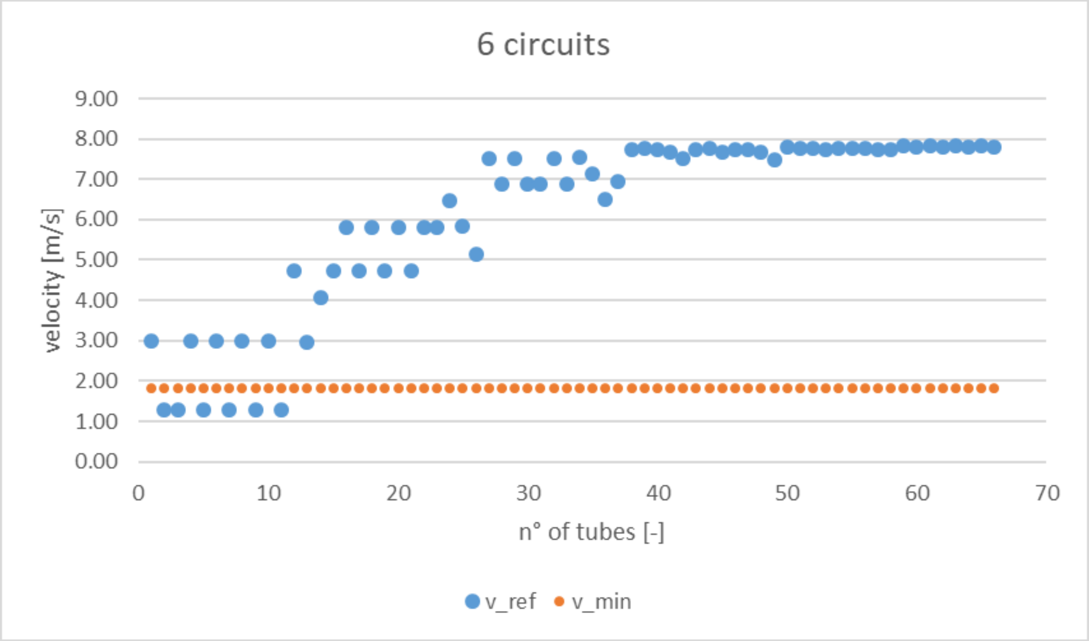
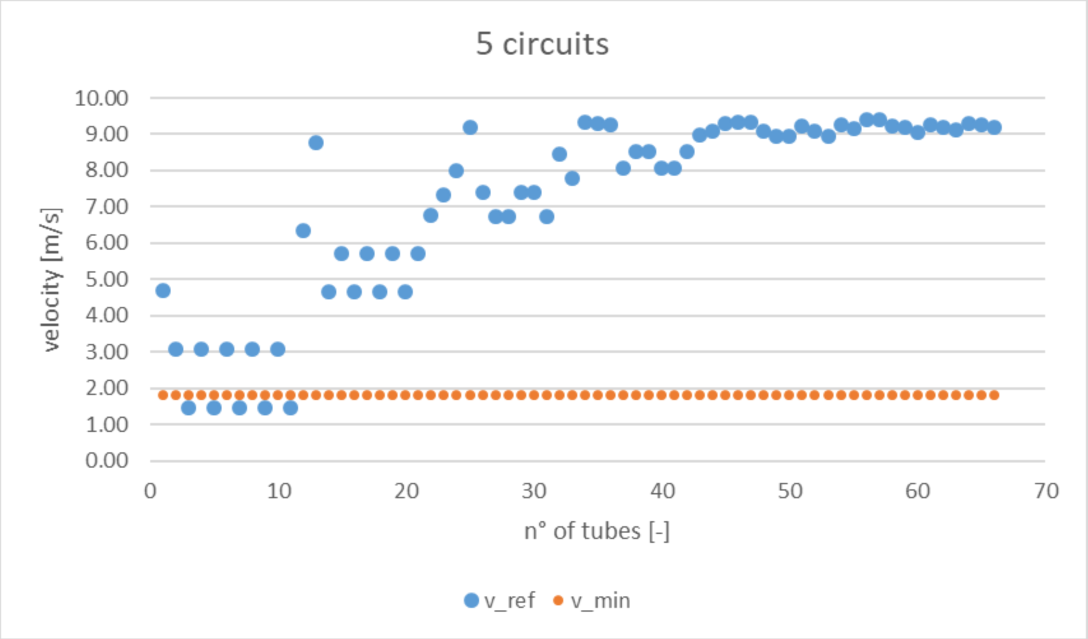


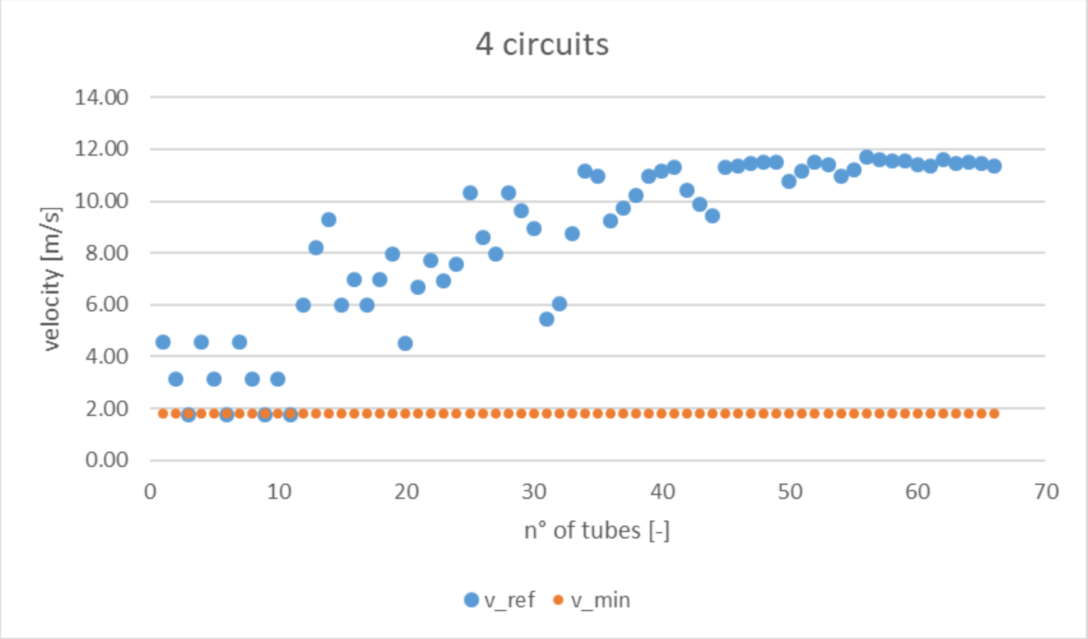
Figure 50: refrigerant and oil velocity for different n° of circuits for 5mm treatment heat exchanger

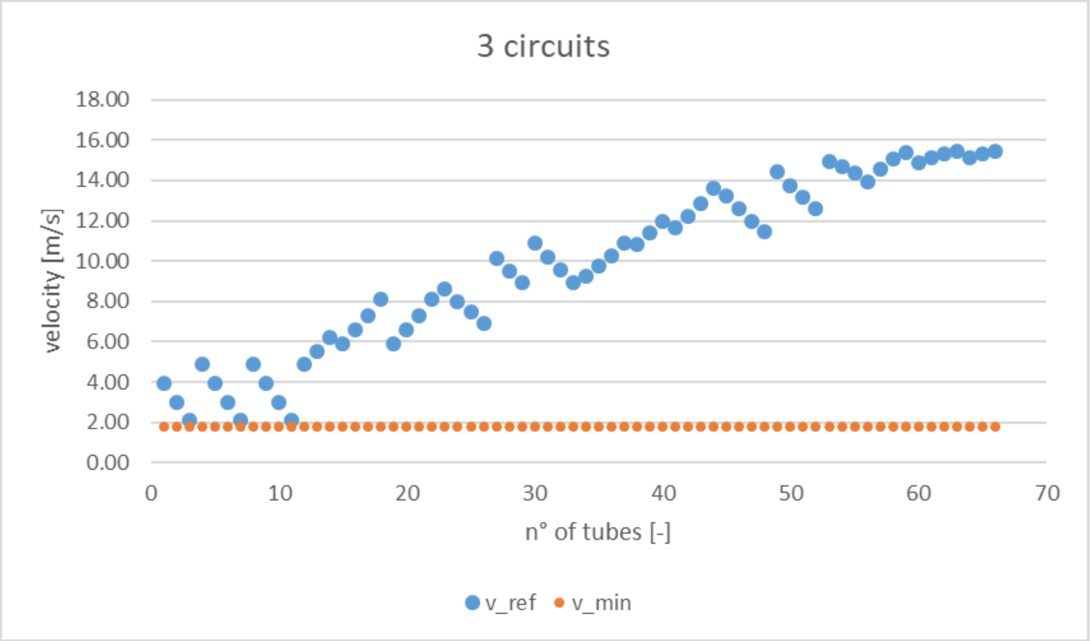
Source 4mm


7 circuits
velocity [m/s]
8.00
7.00
6.00
5.00
4.00
3.00
2.00
1.00
0.00
0
10
20
30
40
50
60
70
n° of tubes [-]
● v_ref
● v_min


6 circuits
velocity [m/s]
9.00
8.00
7.00
6.00
5.00
4.00
3.00
2.00
1.00
0.00
0
10
20
30
40
50
60
70
n° of tubes [-]
● v_ref
● v_min


5 circuits
velocity [m/s]
10.00
9.00
8.00
7.00
6.00
5.00
4.00
3.00
2.00
1.00
0.00
0
10
20
30
40
50
60
70
n° of tubes [-]
● v_ref ● v_min


4 circuits
velocity [m/s]
14.00
12.00
10.00
8.00
6.00
4.00
2.00
0.00
0
10
20
30
40
50
60
70
n° of tubes [-]
● v_ref ● v_min



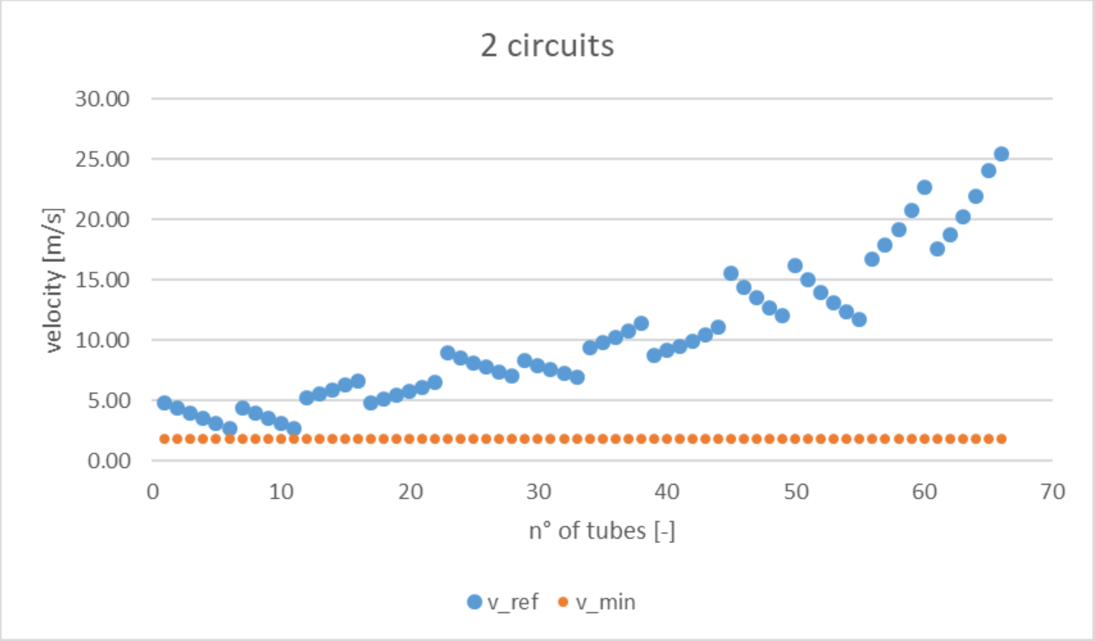
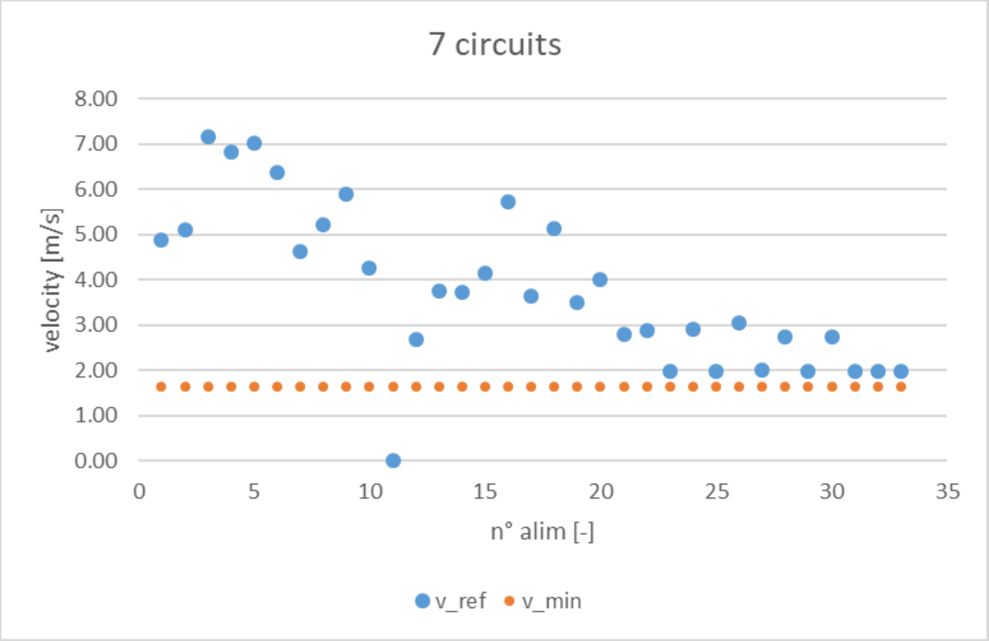
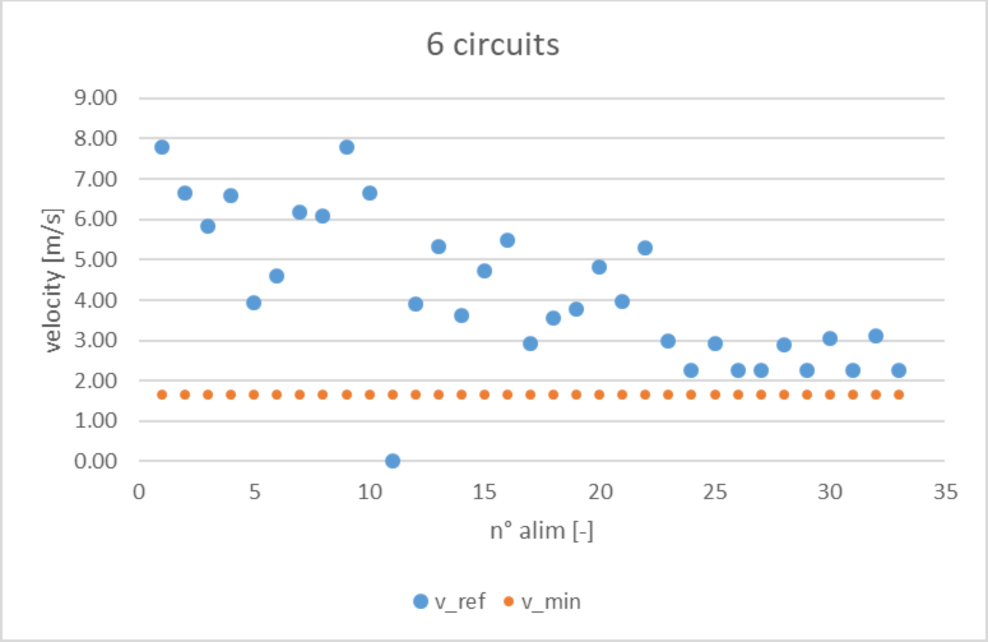
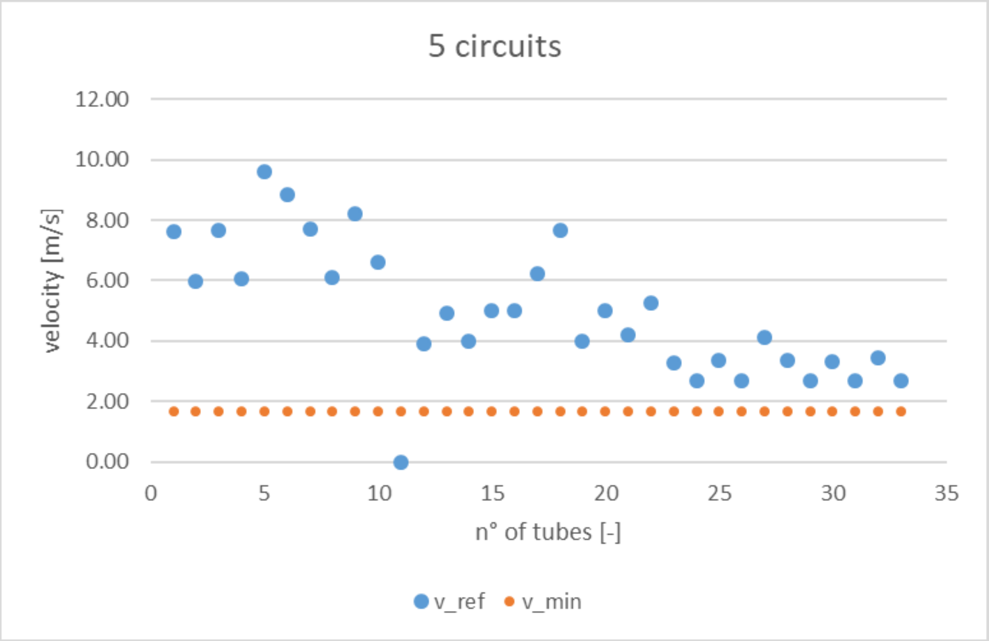


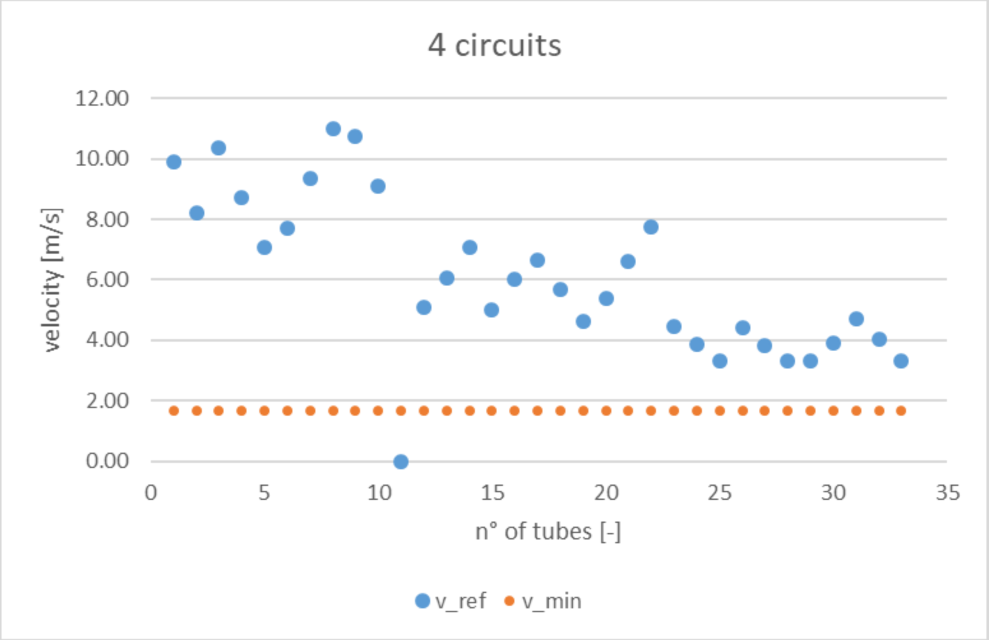
Figure 51: refrigerant and oil velocity for different n° of circuits for 4mm source heat exchanger

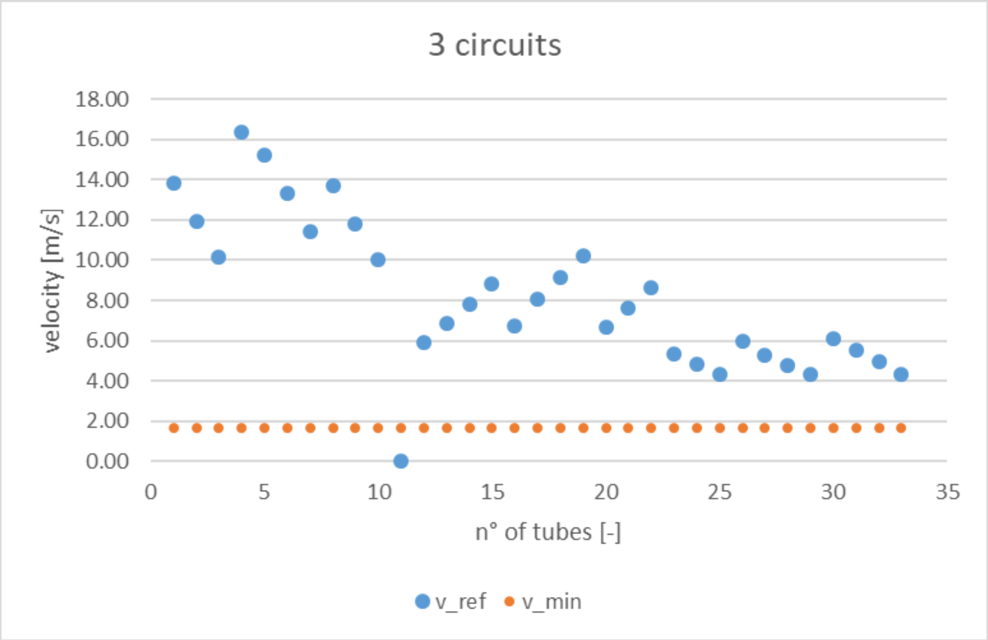
Treatment 4mm






5 circuits
velocity [m/s]
12.00
10.00
8.00
6.00
4.00
2.00
0.00
0
5
10
15
20
25
30
35
n° of tubes [-]
● v_ref ● v_min


4 circuits
velocity [m/s]
12.00
10.00
8.00
6.00
4.00
2.00
0.00
0
5
10
15
20
25
30
35
n° of tubes [-]
● v_ref ● v_min



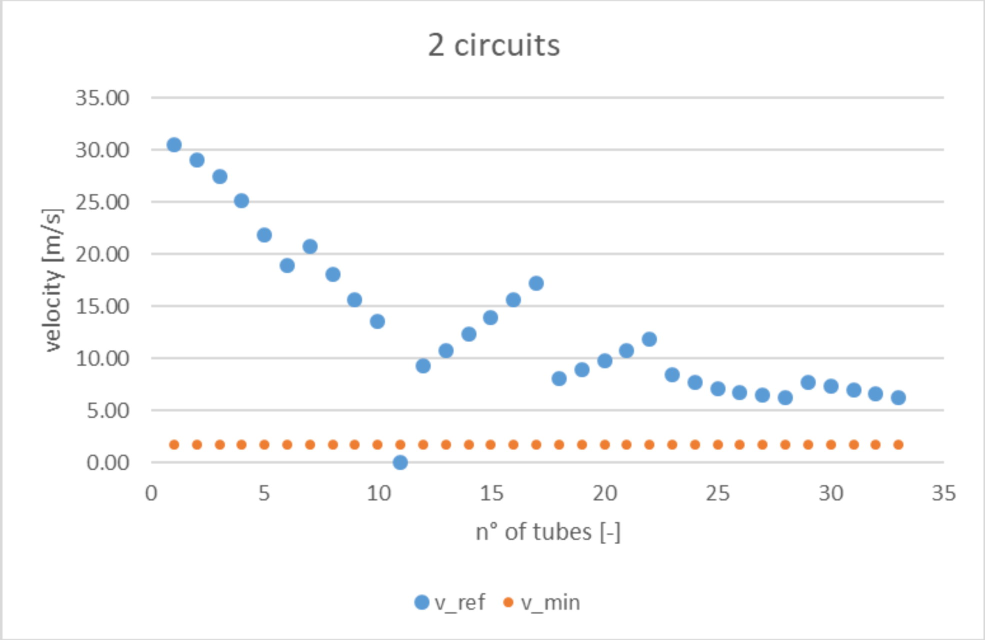


Figure 52: refrigerant and oil velocity for different n° of circuits for 4mm treatment heat exchanger

EVAPORATOR SOURCE (H)

- 5mm: from the graphs, we see how as the circuits decreases, the refrigerant speed increases thus exceeding the minimum speed required for the oil drag. In particular, we can see that initially on the 7 circuits we have 14 tubes where the v_min is not respected. These 14 tubes are exactly the first two tubes of each circuit (inlet). As we can see, this phenomenon occurs again up to 3 circuits, where in all these configurations the v_min is respected everywhere except on the first inlet tube of each circuit. Let's see how the geometry with 2 circuits has no value that falls below the minimum.

- 4mm: in this case, with smaller diameter pipes, we see that both the 2 circuits and the 3 circuits do not completely respect the vel_min limit.

EVAPORATOR TREATMENT (C)

- 5mm: in this case the 7 circuits have only the first tube (inlet) of each circuit that does not respect the v_min. Going down with the number of the circuits we see that also the 6 Alim has a behavior similar to the previous one. From 5alim to descend we see that all the tubes comply with the condition of vel_min, it is possible to note that the refrigerant speed is still low and close to the limit in the near future of the inputs of each circuit, but still respected.
- 4mm: as we can see when using 4 mm diameter for treatment piping, all configurations have in each tube a v_ref> v_min.

Analysis of the results

We know that, as circuits increase, the flow is divided into several ducts and this means that its speed in the single circuit is lower. The refrigerant velocity is high enough to ensure the oil drag, where there are more circuits we see that in the first duct of each circuit the speed is lower than the rest of the circuit and in some cases falls below the limit. This phenomenon can be justified because at the entrance of each circuit the refrigerant is composed of a biphasic mixture with a predominant amount of liquid and as we know the liquid has a higher density of steam and therefore a lower speed. We also consider that when the refrigerant is in the phase with a higher percentage of liquid we can assume that the oil does not have problems with dragging. As we can see by setting a comparison between 5mm and 4mm, the refrigerant speeds are higher overall and this means that, if for example we consider the configuration with 3 circuits for the source exchanger in 5mm and 4mm, it would be preferable to choose the one with a smaller diameter as the refrigerant speed always meets the minimum speed required for the oil drag. If we analyse the heat exchanger treatment we see how except for the 7 and 6 circuits all the configurations have higher speeds to the minimum, this implies that from this analysis can be excluded from the choice batteries in a precautionary way:

- Source 5mm: 7 circuits, 6 circuits because in addition to having many pipelines with v_ref< v_min, these speeds are much lower than the limit
- Treatment 5mm: 7 circuits, 6 circuits for the same reason.
- Source 4mm: only 7 circuits has 7 tubes with speeds much lower than the minimum
- Treatment 4mm: no exclusion, all meet the minimum required speed.

This exclusion is extremely precautionary because the speed not respected only on the first tube does not necessarily mean that all the oil is stuck in the latter because as already said the refrigerant is in a mixture with a liquid prevalence that is known we know we don't have oil drag problems. In addition, the minimum value of speed for oil return has been calculated at the inlet of the evaporative exchanger and therefore where a higher speed is required than that at the outlet of the exchanger.

5.3.3.2 *"In-Out analysis"*

Premise: no differences between 5 and 4 mm, maintained equal geometries.

For this unit, for reasons of space and units geometry, we want to preserve as much as possible the features of the original R32 unit. This also implies where the distributor and collector are located. There are two possible configurations:

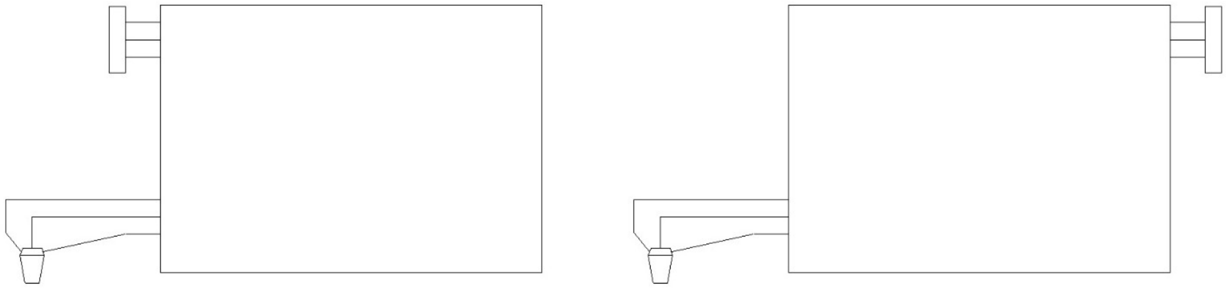


Figure 53; possible distributor - collector configuration

In the left side we see how they are arranged on the same side, in the right side they are arranged on opposite sides of the exchanger. You decide to use one configuration rather than the other based on technical considerations: if you have little space on one side you decide to place collector and distributor on the same side. Obviously we must also consider the size of the exchanger that is to be dimensioned: in the case of large (long) if we had distributor on one side, for example near the circuit fridge, and the collector on the other (therefore at a higher distance from the fridge circuit) we would need a duct leading from the bottom of the exchanger to the fridge circuit. This would imply a high length of the pipes and therefore high-distributed losses, more refrigerant charge to be introduced into the refrigerator circuit and also more heat losses along the connecting pipes.

In this unit, for reasons of lack of space in the outermost part of the unit, it was decided to have distributor and collector on the same side, towards the inside of the unit.

Analysing now the configurations designed for the source and treatment heat exchangers we can see that, after excluding some configurations from the previous analysis, the following configurations are excluded to obtain the desired result:

- Treatment 5mm: no excluded
- Treatment 4mm: 6 circuits excluded

- Source 5mm: no excluded
- Source 4mm: 6 circuits excluded

Obviously, it arises to wonder what is the reason for which it was not previously decided to have, in these excluded geometries, distributor and collector on the same side. Simply, as mentioned above, we wanted to use the same number of pipes for each configuration and going to search for the desired conditions, a pipe would be excluded from the configuration. If the comparison made in the previous chapters were then carried out, an error would be introduced since it would be made between exchangers with different quantities of pipes.

5.3.3.3 Return oil at partial load

We have already analyzed the method by which the oil return inside the circuit is evaluated analytically when we are at nominal load. Now the return of the oil will be evaluated even when we are under partial load. We expect to have lower refrigerant speeds than the nominal.

We use the same empirical formula for determining the minimum speed of the refrigerant required to ensure oil drag.

$$V_{min,gas} = \left(9.8\ \phi_D \left[\frac{\rho_{oil} - \rho_{gas}}{\rho_{gas}}\right]\right)^{0.5} \left[\frac{m}{s}\right]$$

Of course, we calculate the density of refrigerant and oil at the new operating points of the cycle.

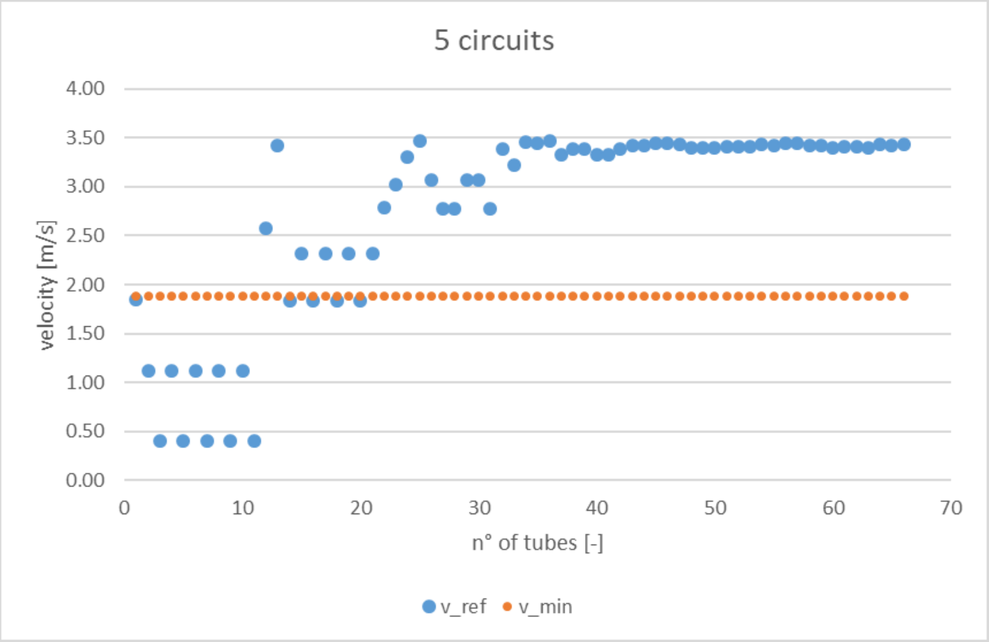
The procedure is repeated for all the remaining configurations both for the source heat exchanger and for the treatment heat exchanger in the evaporation operation because, as already mentioned, it is the most critical phase regarding the oil return.

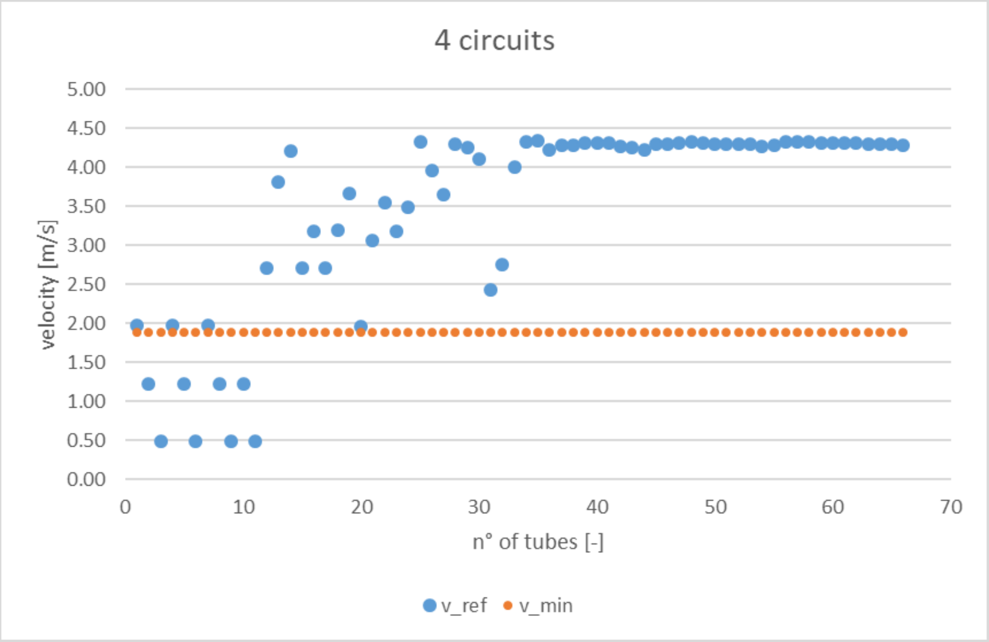
Also remember that after the various steps of introduction practical factors, will be simulated only the exchangers that have complied with the predetermined parameters, are listed below:

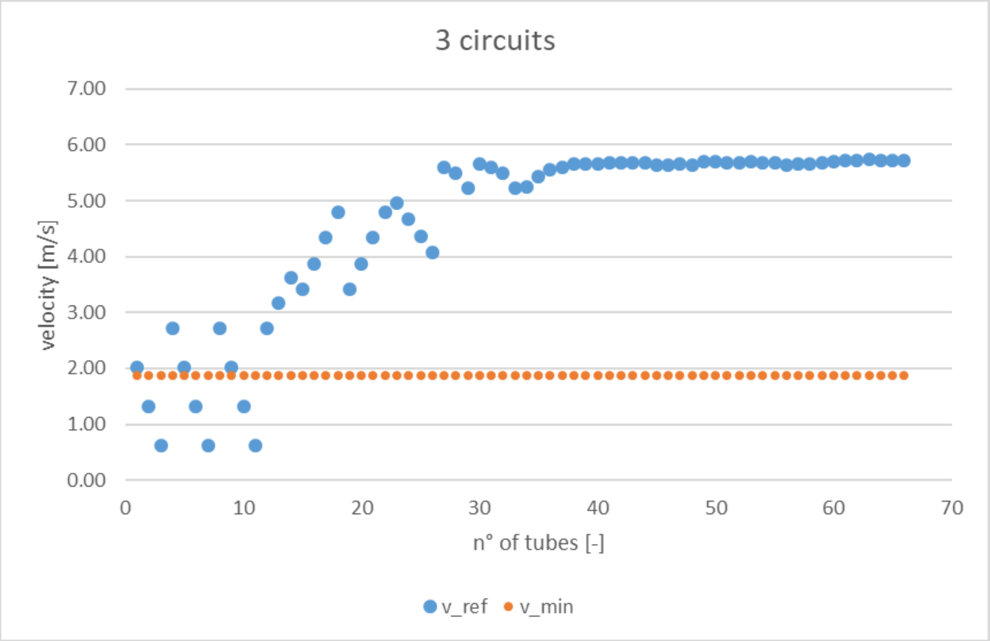
- Treatment 5mm: 5-4-3-2 circuits configurations
- Source 5mm: 5-4-3-2 circuits configurations
- Treatment 4mm: 5-4-3-2 circuits configurations
- Source 4mm: 5-4-3-2 circuits configurations

Below we see the revenue charts from the calculations:

- Source 5mm


5 circuits
velocity [m/s]
4.00
3.50
3.00
2.50
2.00
1.50
1.00
0.50
0.00
0
10
20
30
40
50
60
70
n° of tubes [-]
● v_ref ● v_min


4 circuits
velocity [m/s]
5.00
4.50
4.00
3.50
3.00
2.50
2.00
1.50
1.00
0.50
0.00
0
10
20
30
40
50
60
70
n° of tubes [-]
● v_ref ● v_min



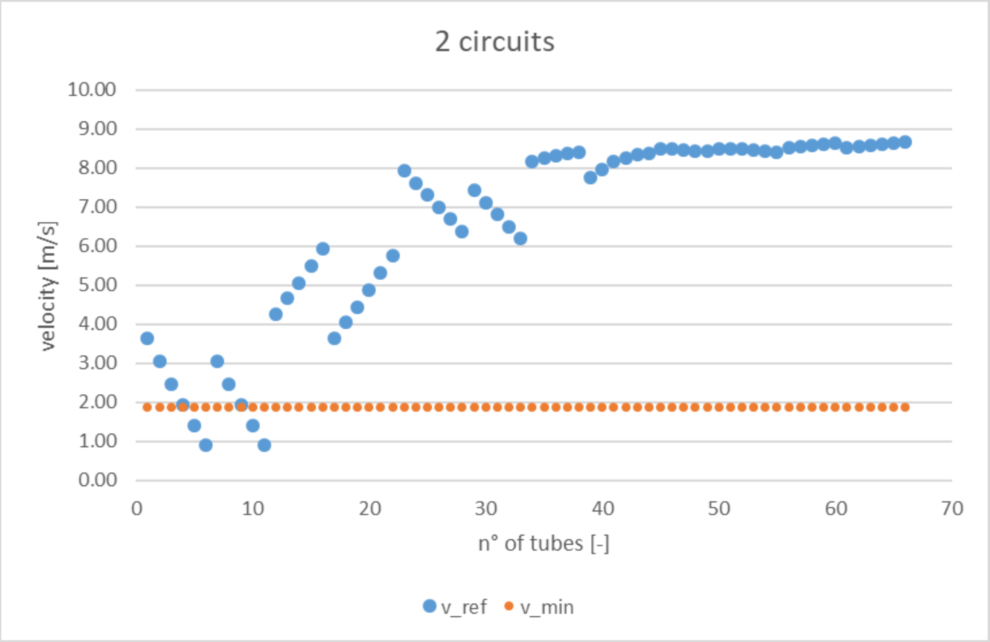
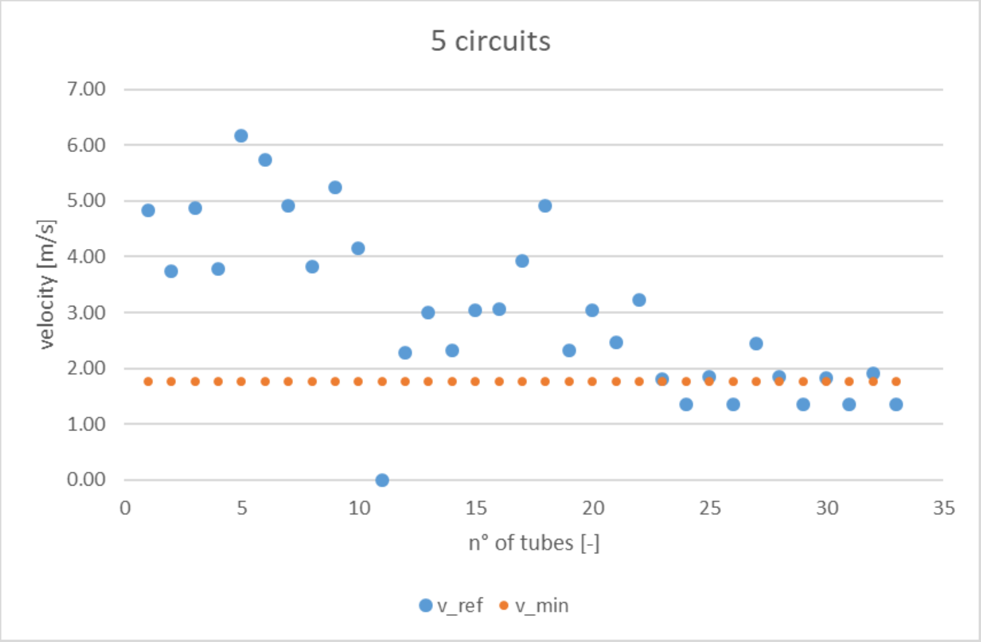
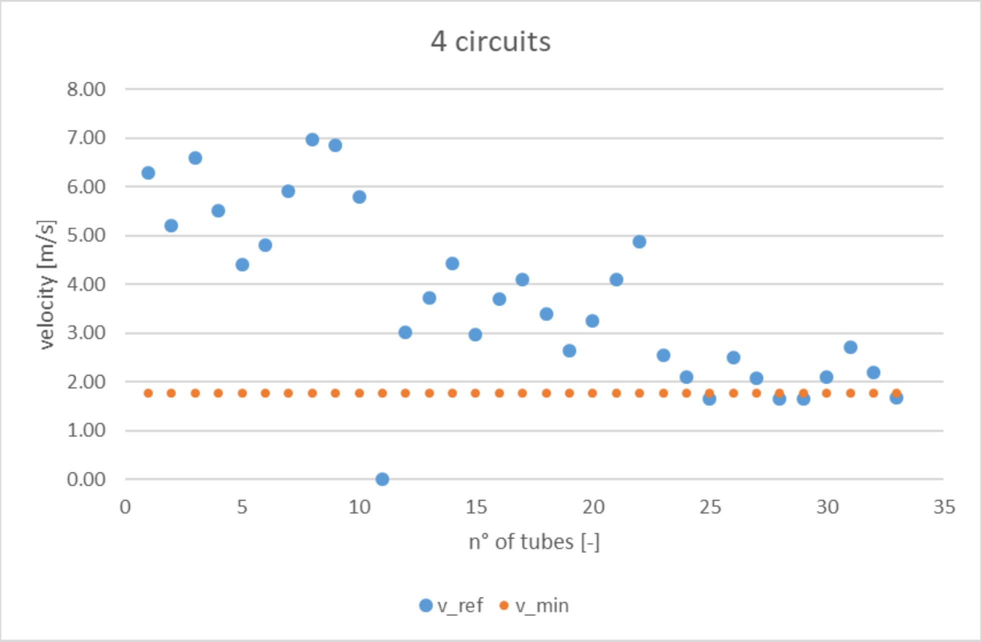


Figure 54: refrigerant and oil velocity for different n° of circuits for 5mm source heat exchanger at partial load

- Treatment 5mm





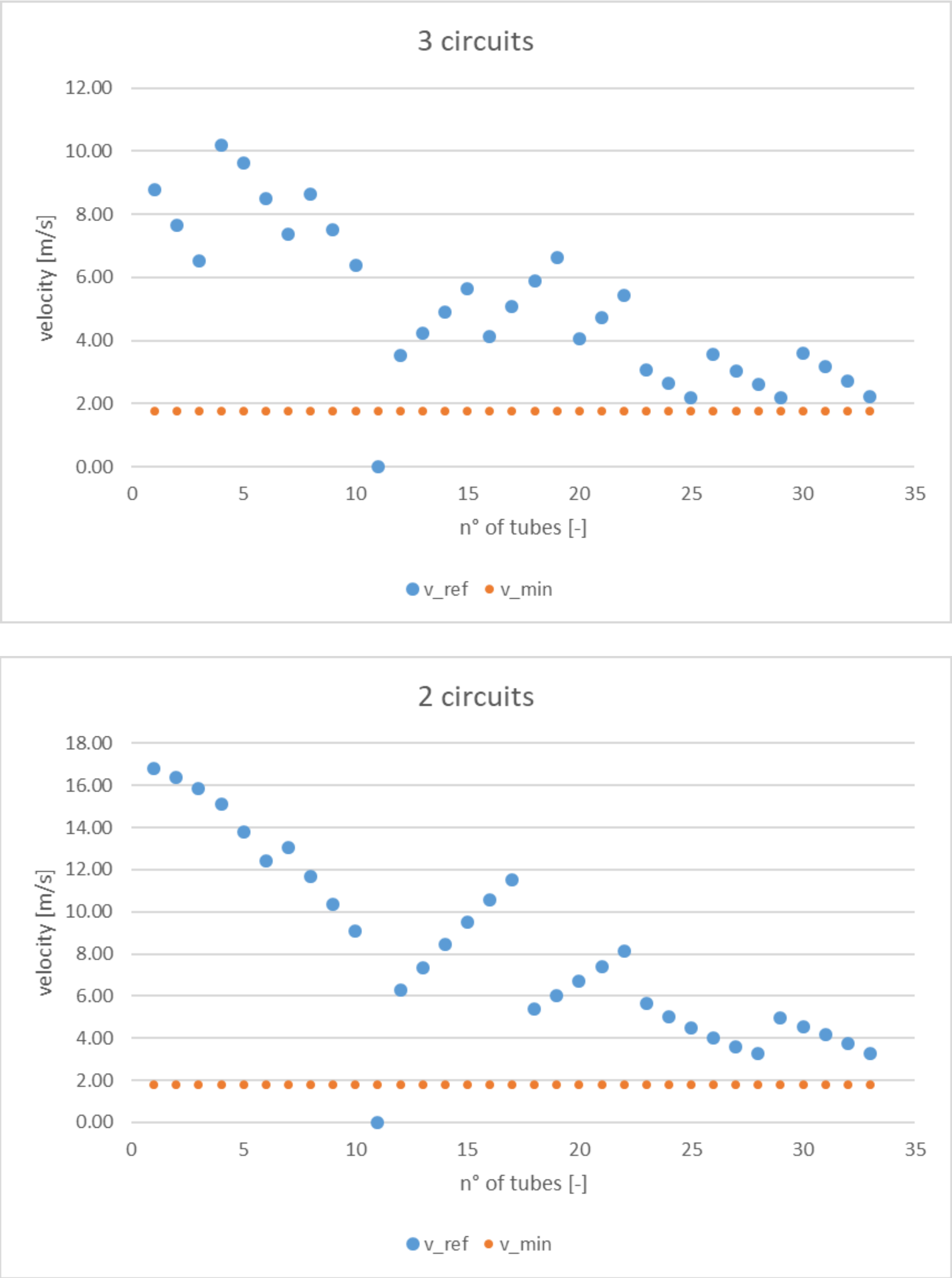
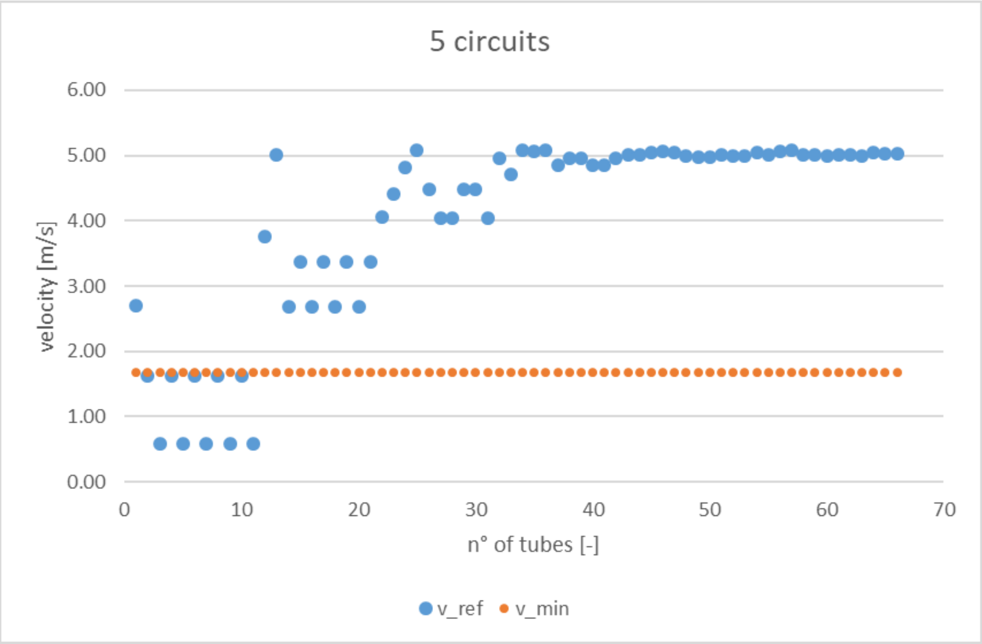
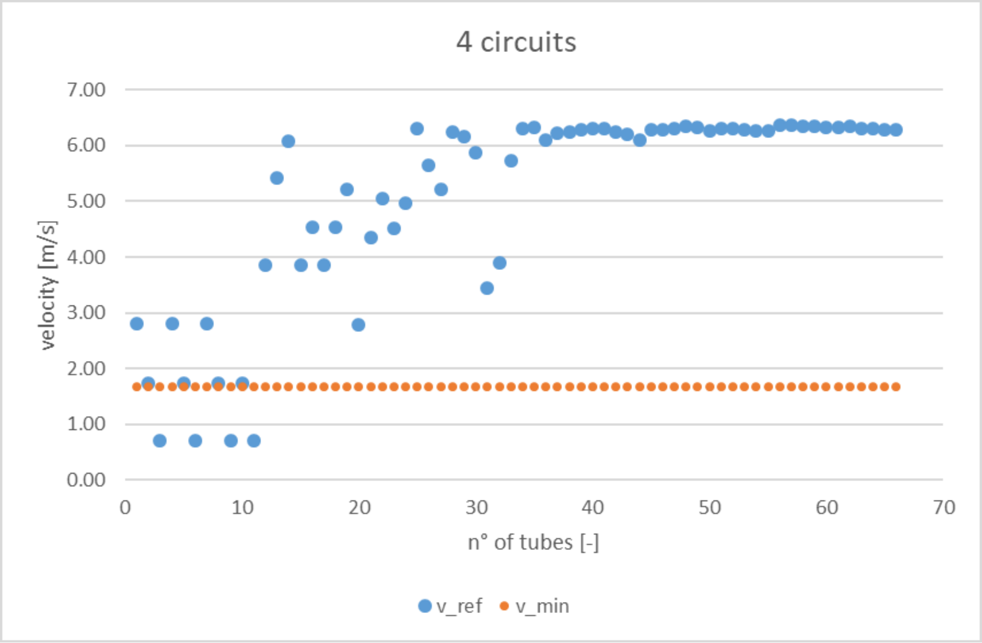
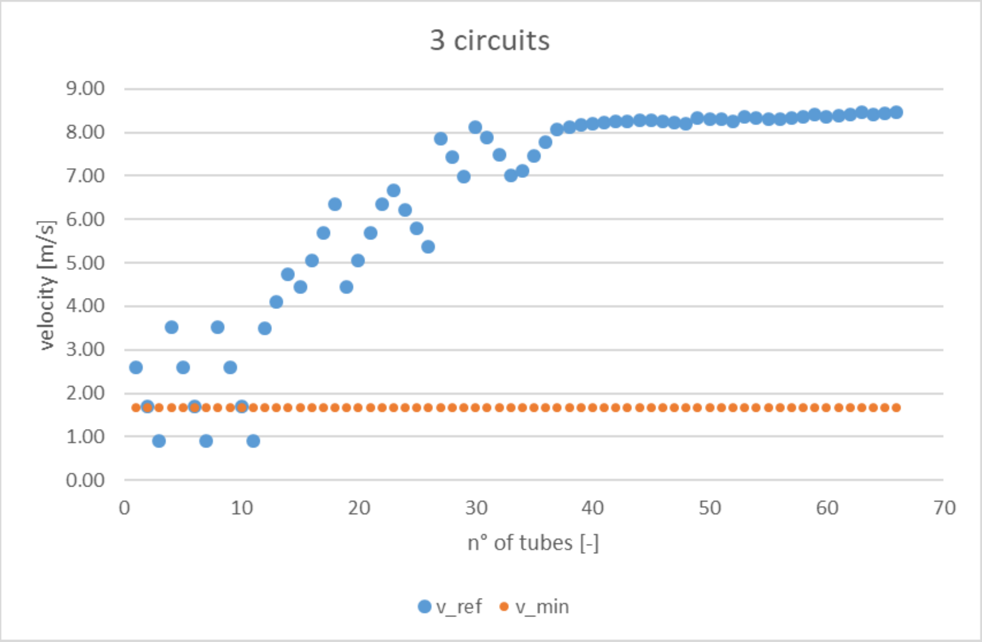


Figure 55: refrigerant and oil velocity for different n° of circuits for 5mm treatment heat exchanger at partial load

- Source 4mm


5 circuits
6.00
5.00
4.00
3.00
2.00
1.00
0.00
velocity [m/s]
0
10
20
30
40
50
60
70
n° of tubes [-]
● v_ref
● v_min


4 circuits
7.00
6.00
5.00
4.00
3.00
2.00
1.00
0.00
velocity [m/s]
0
10
20
30
40
50
60
70
n° of tubes [-]
● v_ref
● v_min



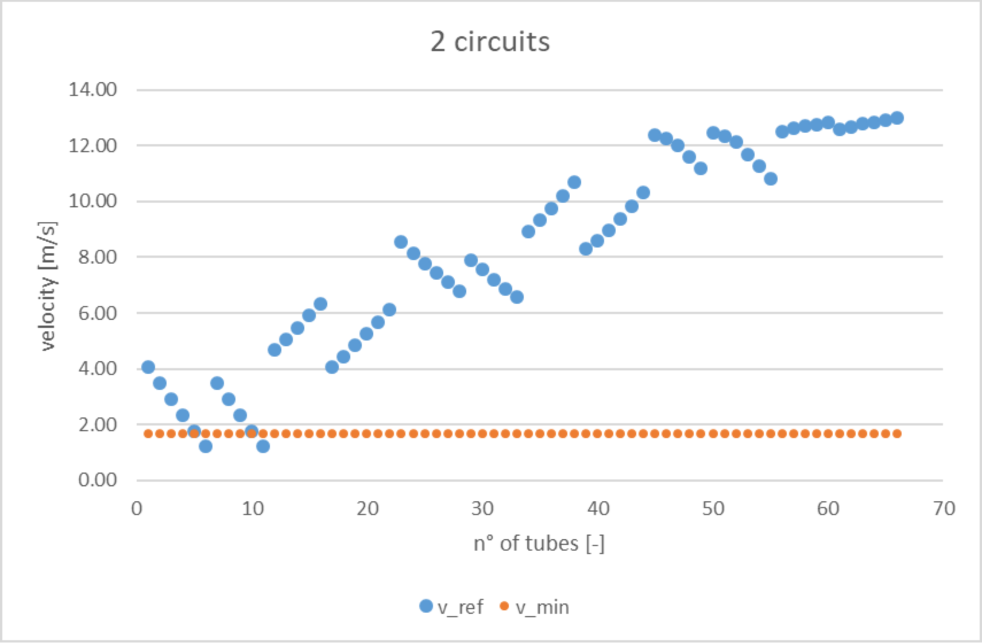
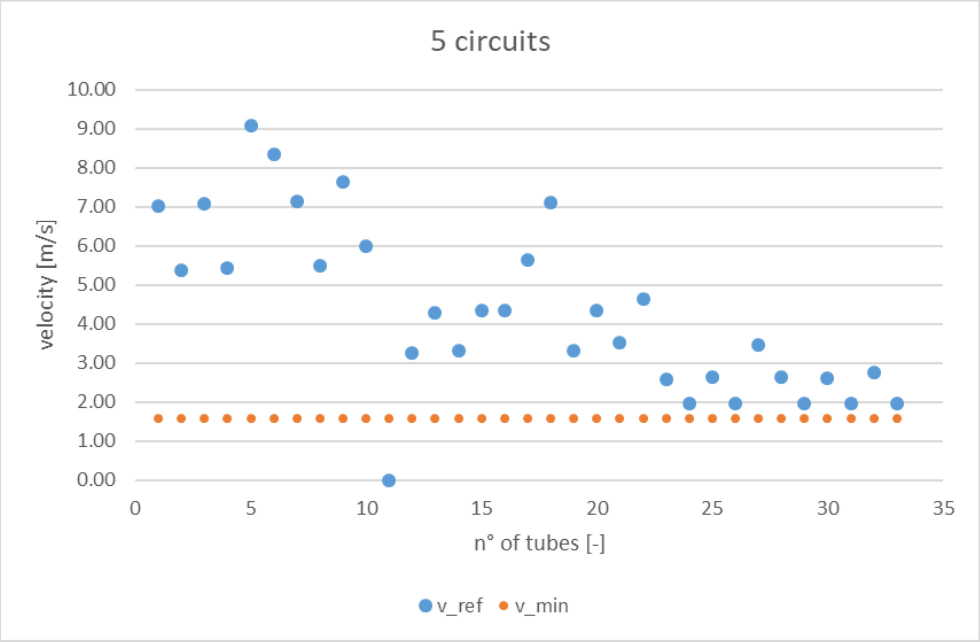
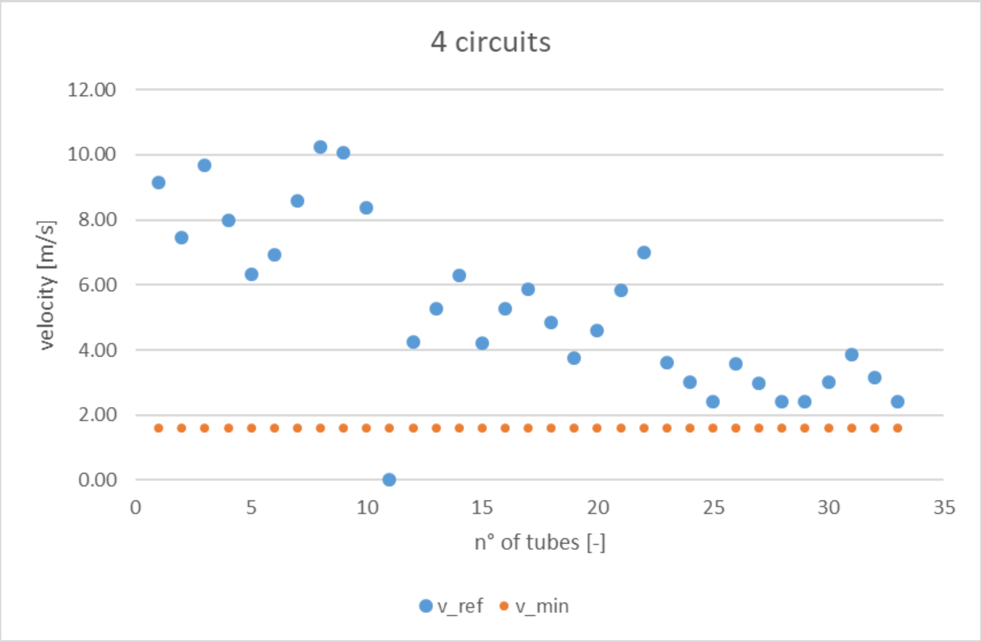
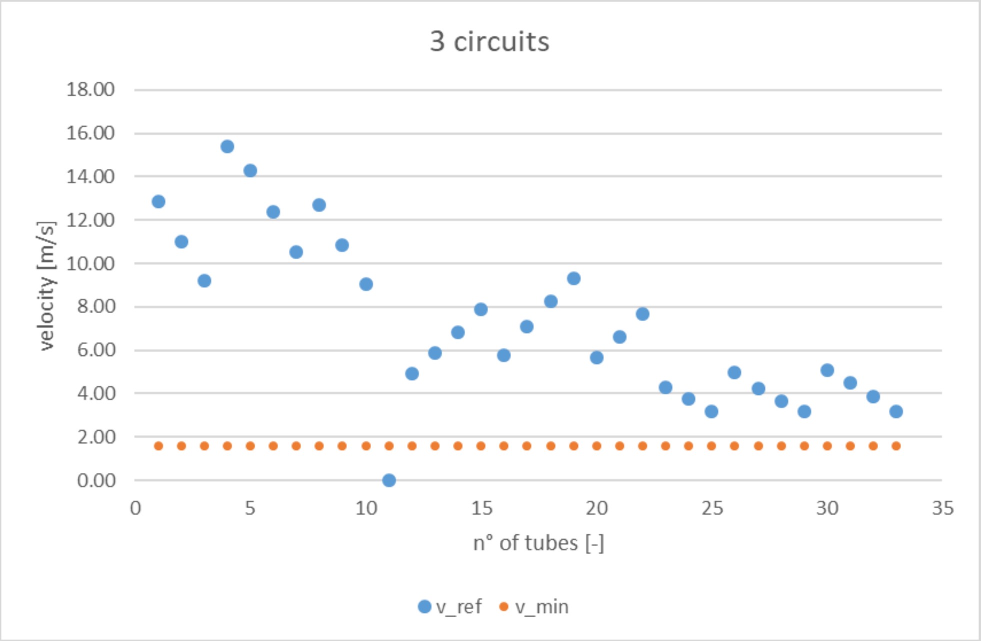


Figure 56: refrigerant and oil velocity for different n° of circuits for 4mm source heat exchanger at partial load

- Treatment 4mm







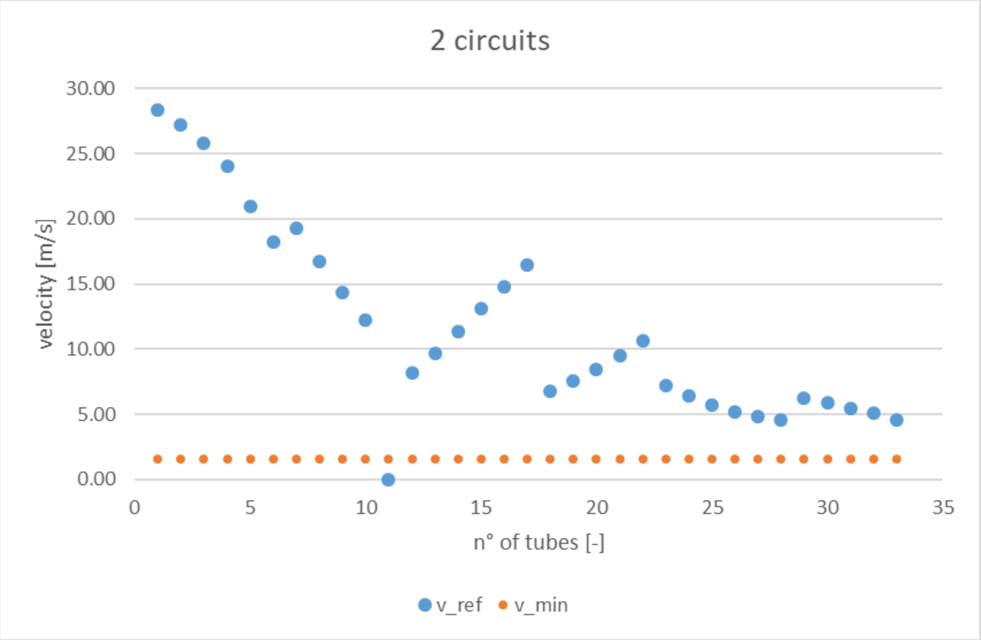


Figure 57: refrigerant and oil velocity for different n° of circuits for 4mm treatment heat exchanger at partial load

Source 5mm: As we can see, the heat exchanger with 5 circuits, while working correctly at nominal load, at partial load there are 15 tubes where at the minimum speed required for the oil drag is not respected. Going down of circuits we see how the speed generally rises. As we have already seen before, in the first two tubes of each circuit the minimum speed not respected, the effect fades as circuits decrease. In the 3 and 2 circuits we see that when the v_min is not respected, however it is quite close to the v_min oil.

Treatment 5mm: in the heat exchanger treatment we see that the speeds are completely respected both in the configuration with three circuits and in that with 2 circuits, while in the two remaining we find the same trend previously seen, where speed is not respected.

Source 4mm: also for the 4mm we see that the source heat exchanger, the configuration with five circuits has many pipes where the minimum speed is not respected. It has the same type of heat exchanger but with a diameter of the tubes at 5mm, but compared to this, it has slightly higher average speeds.

Treatment 4mm: as we can see in all the 4 mm treatment configurations we have that the speeds are respected, in this case we see that in the configurations with less circuits, we have high speeds that are close to 30m/s.

We can then perform a further exclusion from the possible choice of exchangers, below will be listed the remaining configurations that meet all the limitations seen above:

- Source 5mm: 5 and 4 circuits excluded, 3 and 2 circuits left
- Treatment 5mm: excluding 5 and 4 circuits, remaining 3 and 2 circuits
- Source 4mm: excluding 5 and 4 circuits, 3 and 2 circuits left
- Treatment 4mm: none excluded, 5, 4, 3 and 2 circuits left

5.3.3.4 Final choices

After conducting the theoretical analysis and introducing the practical limitations, we were able to perform a good skimming to be able to decide which exchangers to use on this unit.

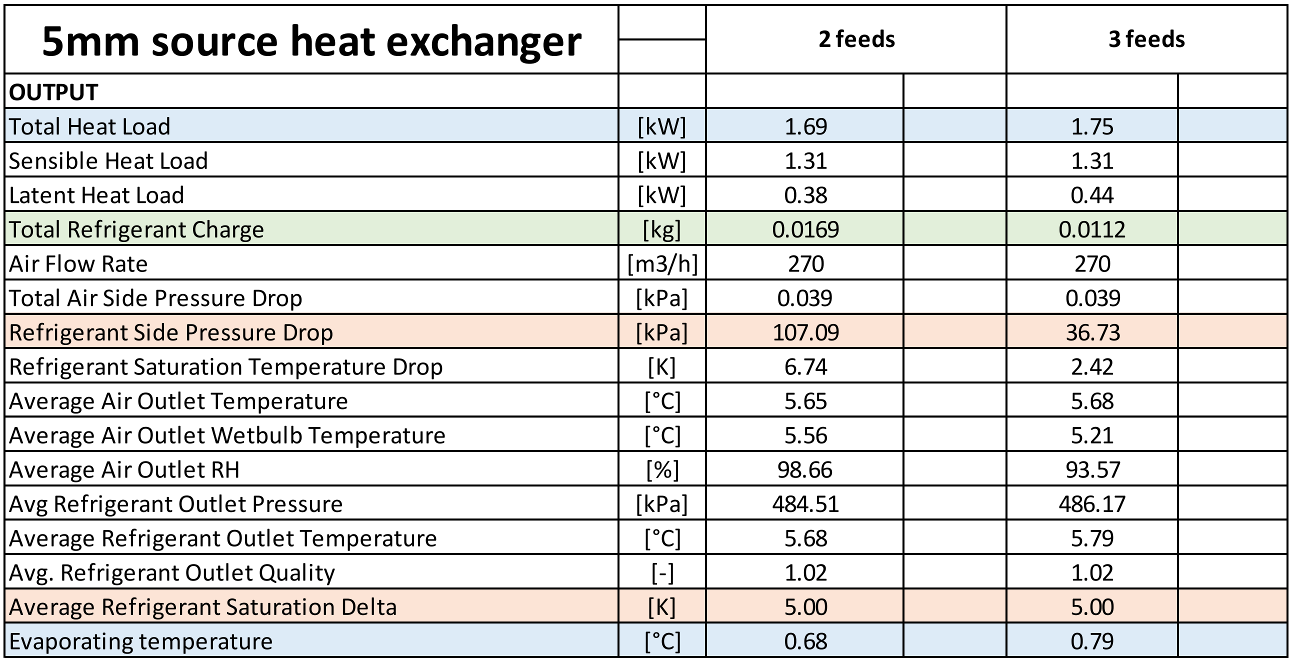
The realization of the prototype for the heat exchangers with 4mm pipes was entrusted to an external company, after a cost analysis, It was agreed that from an economic point of view it was not convenient to consider the 4mm line despite being very interesting from an engineering point of view.

The choice therefore falls on the 5mm configurations for both the source exchanger and the treatment exchanger.

Let’s see now the two possible choices for source exchanger and treatment exchanger:

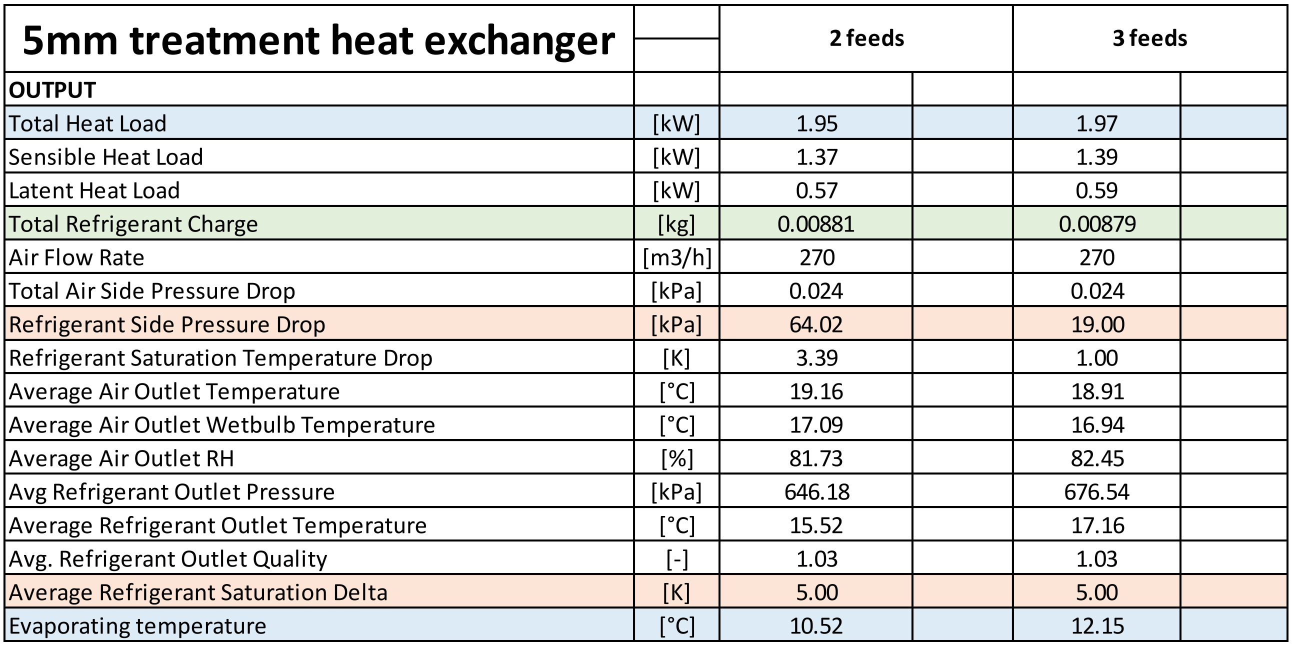
- 5mm source: 3 circuits, 2 circuits
- 5mm treatment: 3 circuits, 2 circuits

The following are the data related to the simulations conducted through Coildesigner of the latter in the evaporating conditions:



5mm source heat exchanger		2 feeds		3 feeds	
OUTPUT					
Total Heat Load	[kW]	1.69		1.75	
Sensible Heat Load	[kW]	1.31		1.31	
Latent Heat Load	[kW]	0.38		0.44	
Total Refrigerant Charge	[kg]	0.0169		0.0112	
Air Flow Rate	[m3/h]	270		270	
Total Air Side Pressure Drop	[kPa]	0.039		0.039	
Refrigerant Side Pressure Drop	[kPa]	107.09		36.73	
Refrigerant Saturation Temperature Drop	[K]	6.74		2.42	
Average Air Outlet Temperature	[°C]	5.65		5.68	
Average Air Outlet Wetbulb Temperature	[°C]	5.56		5.21	
Average Air Outlet RH	[%]	98.66		93.57	
Avg Refrigerant Outlet Pressure	[kPa]	484.51		486.17	
Average Refrigerant Outlet Temperature	[°C]	5.68		5.79	
Avg. Refrigerant Outlet Quality	[-]	1.02		1.02	
Average Refrigerant Saturation Delta	[K]	5.00		5.00	
Evaporating temperature	[°C]	0.68		0.79	

Figure 58: 5mm evaporator source heat exchanger values for the remaining two configuration



5mm treatment heat exchanger		2 feeds		3 feeds	
OUTPUT					
Total Heat Load	[kW]	1.95		1.97	
Sensible Heat Load	[kW]	1.37		1.39	
Latent Heat Load	[kW]	0.57		0.59	
Total Refrigerant Charge	[kg]	0.00881		0.00879	
Air Flow Rate	[m3/h]	270		270	
Total Air Side Pressure Drop	[kPa]	0.024		0.024	
Refrigerant Side Pressure Drop	[kPa]	64.02		19.00	
Refrigerant Saturation Temperature Drop	[K]	3.39		1.00	
Average Air Outlet Temperature	[°C]	19.16		18.91	
Average Air Outlet Wetbulb Temperature	[°C]	17.09		16.94	
Average Air Outlet RH	[%]	81.73		82.45	
Avg Refrigerant Outlet Pressure	[kPa]	646.18		676.54	
Average Refrigerant Outlet Temperature	[°C]	15.52		17.16	
Avg. Refrigerant Outlet Quality	[-]	1.03		1.03	
Average Refrigerant Saturation Delta	[K]	5.00		5.00	
Evaporating temperature	[°C]	10.52		12.15	

Figure 59: 5mm evaporator treatment heat exchanger values for the remaining two configuration

- Source heat exchanger: between these two configurations, the 2 circuits have very high refrigerant side pressure losses, over 100 kPa. With high losses we know that we would have a high refrigerant speed, which implies a good oil return, but speeding also involves possible vibrations inside the tubes. Moreover, from these simulations, where we fixed the power exchanged by these heat exchangers, it is not easily visible the potential loss of yield linked to high pressure losses, but in practice it is known that high pressure losses lead to a deterioration in the efficiency of the heat exchanger. In the other configuration, that is the one with 3 circuits, there are pressure drops of about 37 kPa, sufficient to obtain adequate speeds for oil return and we can consider them a good compromise to have a ratio between heat exchanger yield and adequate losses. The choice for the source heat exchanger then falls on the heat exchanger 5 mm 3 circuits.

- Treatment heat exchanger: Between the two remaining configurations, it is possible to note that unlike the source heat exchanger, we have the reverse situation; you can choose between the 2 circuits with load losses equal to 64kPa or 3 circuits with load losses equal to 19kPa. As we saw from the previous analysis on the minimum refrigerant speed to ensure oil return, both met the criteria. We can see that in partial load operation the 3 circuits configuration had many tubes where the minimum speed approached the minimum speed, so we wonder if in the event that the required power dropped further, the exchanger was able to guarantee the return of oil to the compressor. To ensure the return oil is preferred to adopt a precautionary measure and the choice for the heat exchanger treatment falls on the 5 mm configuration with 2 circuits at the expense of a lower evaporation temperature which results in lower efficiencies..

Below are the circuitry of the two heat exchangers selected.

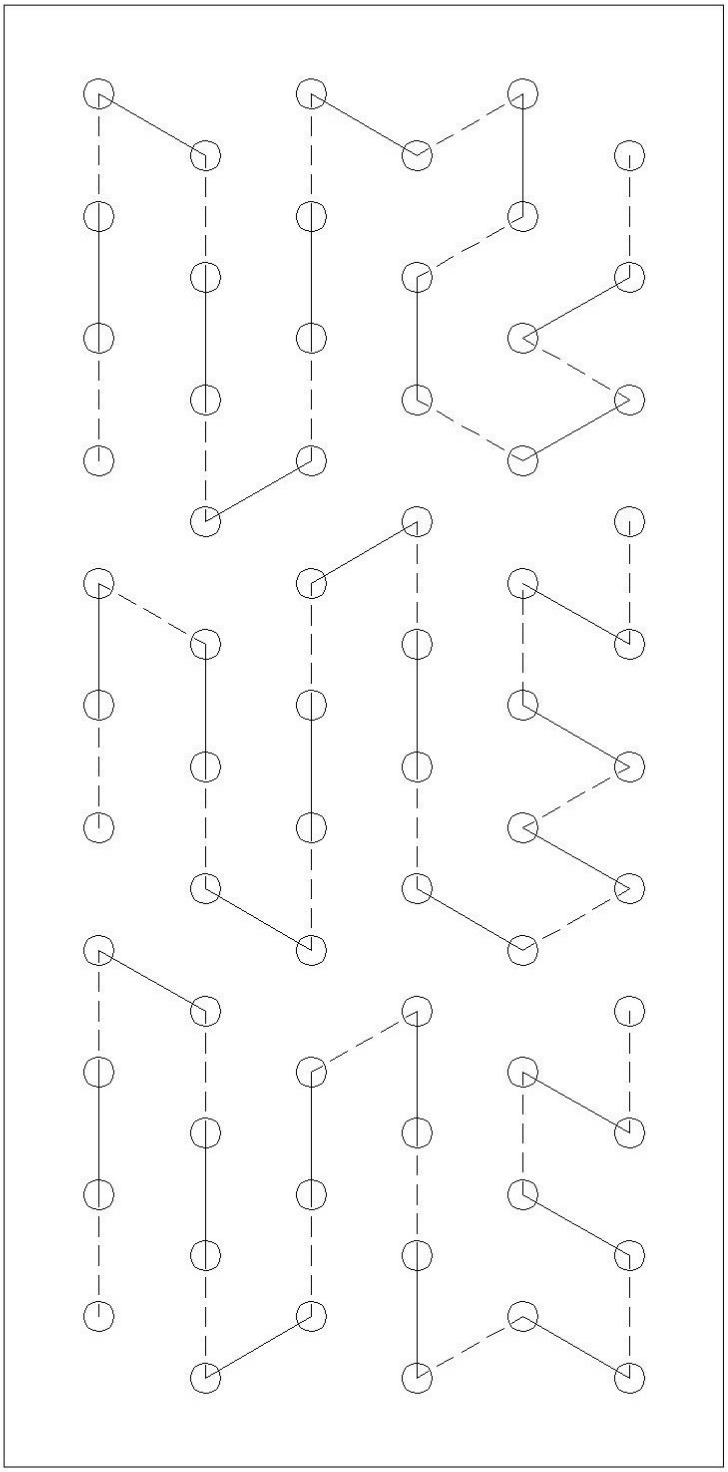


Figure 60: source heat exchanger circuitry

In the image above on the left we find the inlets of each circuit and on the right the outlets. The air passes through the exchanger from left to right, then defines the exchanger in concurrent.

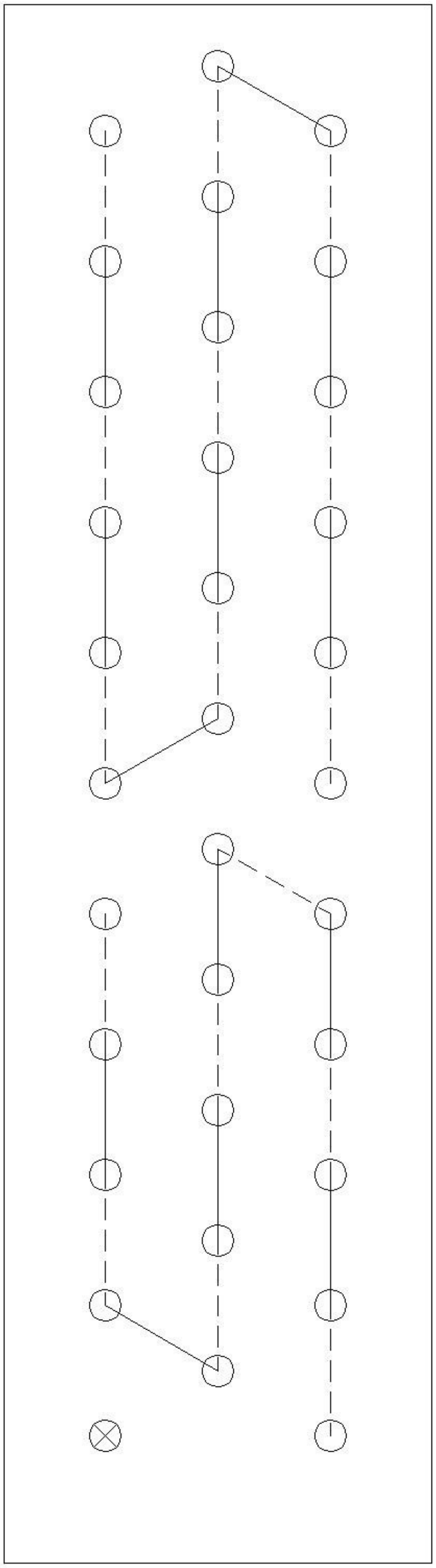


Figure 61: treatment heat exchanger circuitry

In the image above, on the right we find the inlets of each circuit and on the left the outlets. The air passes through the exchanger from left to right, then defines the exchanger in countercurrent. As we can see we have an unused tube in the lower left, this was kept plugged in all simulations of the same exchanger in order to compare the exchangers in the same conditions.

5.3.4 Analysis of the results

We initially defined the theoretical simulation of the heat exchangers. Simulations were performed at fixed initial conditions of condensation temperature, evaporation temperature, external temperature, SH, SC, RH and target exchanged power. 16 heat exchanger configurations were designed for both the treatment exchanger and the source exchanger. The configurations were different in the number of circuits (from 1 to 8 circuits) and in the external diameter of the tubes (5mm and 4mm). The behaviour of each exchanger was evaluated using the software Coildesigner and export data into an Excel sheet, graphing them in function of: pressure drop refrigerant side, refrigerant charge, evaporation and condensation temperature.

The heat exchangers were then compared during evaporator and condensation operation, and the geometries with an external diameter of 5mm and 4mm for both the treatment exchanger and the source exchanger.

After that, some practical limitations were included to understand if the sized exchangers could be used in real life and not only theoretical application. The limitations analysed will be briefly listed and summarized below.

- Minimum refrigerant speed to ensure oil return: to evaluate the minimum refrigerant speed that ensures oil return we used an empirical formula where the refrigerant and oil densities are required at the given operating point. From the software, we extrapolated the refrigerant speeds for each configuration in the evaporator operation only because this is the most critical phase for the return of the oil. We analysed tube by tube if the simulated speeds met this minimum calculated speed. Configurations that did not meet these minimum speeds were therefore excluded.
- In-Out analysis: from this analysis, we saw that in the case of the VMC in question, it was necessary to have the distributor and the collector on the same side of the exchanger, this limitation excluded some configurations that did not meet this criterion.
- Partial load oil return: we have re-simulated all the remaining configurations at partial load condition therefore with lower heat exchange powers. Calculate again the refrigerant and oil densities at the new operating points and evaluate the speeds under partial load conditions. This limitation has led to discard different configurations, but remember that often, during the entire operating period of the unit, the thermal load required is not constant and varies with external conditions and that therefore this analysis is fundamental.

- Economic cost: Another practical factor to consider is the economic factor. As we have seen, the unit must be optimized from an economic point of view to make profitable the development of the units.

Taking into account these limitations, we have discarded the configurations that did not meet the parameters introduced, until we obtained two possible choices both for the treatment exchanger and for the source exchanger. So we started 32 simulations total and then from 16 simulations for each exchanger up to two possible configurations to choose from.

Process of elimination				
	Treatment heat exchanger		Source heat exchanger	
	5mm	4mm	5mm	4mm
$V_{min\ oil\ return}$	7-6 circuits	/	7-6 circuits	7 circuits
IN_OUT analysis	/	6 circuits	/	6 circuits
$V_{min\ return\ oil}$ partial load	5-4 circuits	/	5-4 circuits	5-4 circuits
already industrially mature	/	5-4-3-2 circuits	/	3-2 circuits
Remaining configurations	3 circuits 2 circuits	/	3 circuits 2 circuits	/

Table 18: the circuits mentioned in the table are the ones discarded. "/" means no elimination

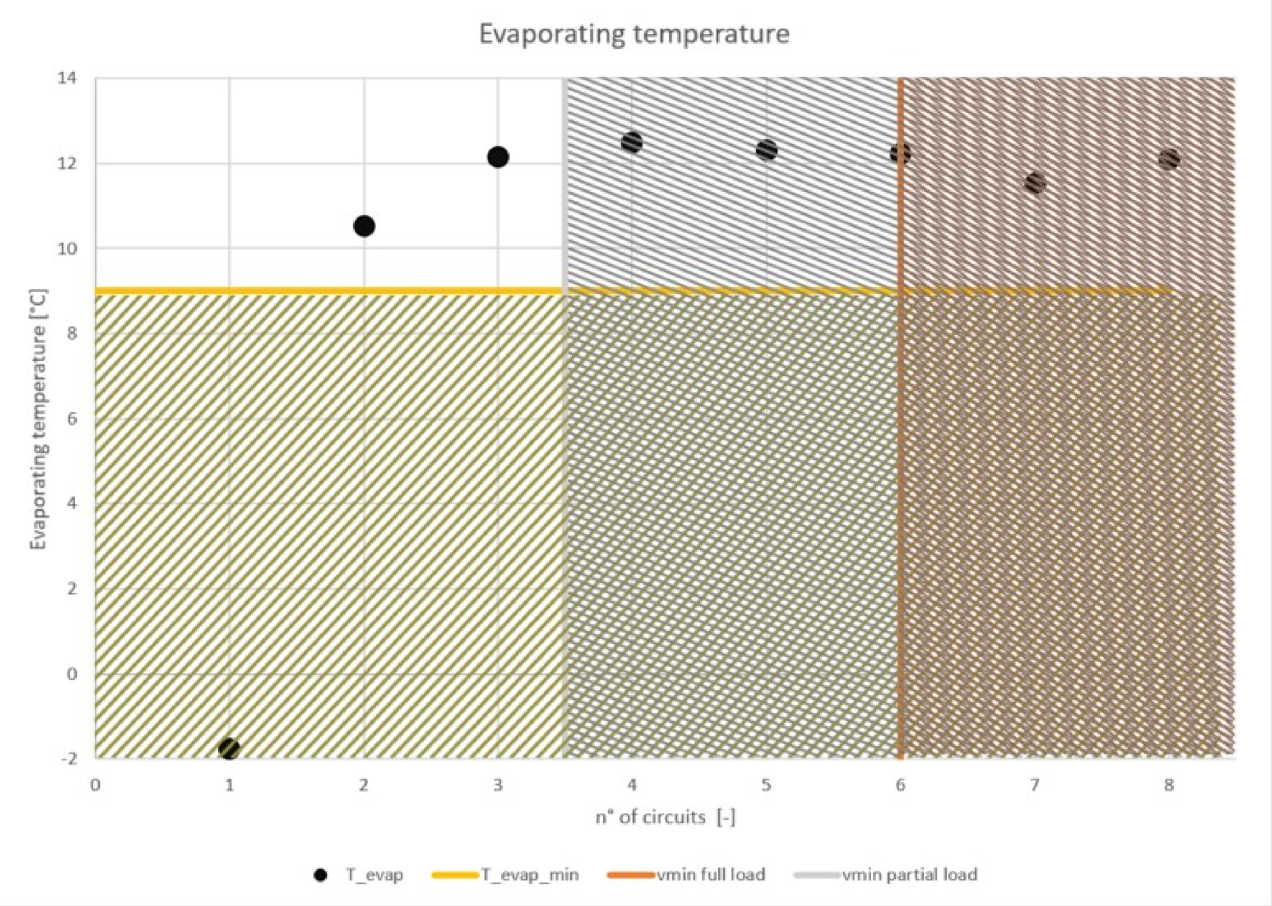


Figure 62: Elimination graph for 5mm treatment heat exchanger

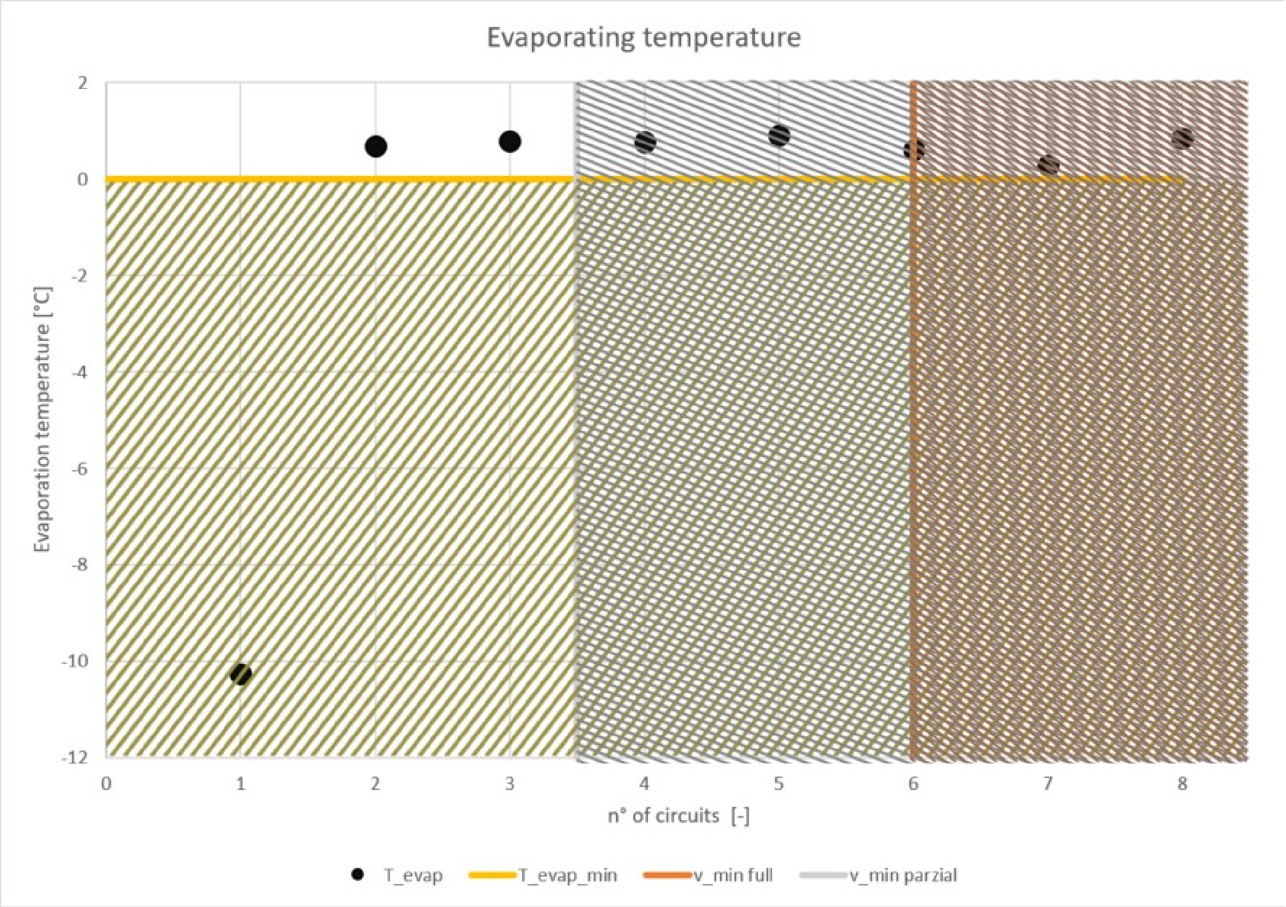


Figure 63: Elimination graph for 5mm source heat exchanger

The remaining two configurations were compared for both exchangers and the analysis revealed that:

- Treatment heat exchanger: the 5mm two-circuit configuration is chosen because it is the configuration that has a good compromise between pressure drop and heat exchanger yield. The configuration with three circuits 5mm has too low pressure losses and at partial load operation some pipes did not reach the minimum speed required for oil return.
- Source: Between the two remaining configurations at 5mm, the one with two circuits has been discarded because it has high pressure drops that ensure the return of the oil, but on the other hand, we know we have high internal tube refrigerant fluid velocities which can lead to possible vibrations and inefficiencies. It is then chosen the configuration with 3 circuits 5mm diameter that is a good exchanger and that meets the conditions established during the process of choice.

5.4 Scroll air to water heat pump (A-W)

As already seen for the VMC (A-A) it was decided to keep as much as possible the shapes and dimensions of the initial unit and then to size a heat exchanger with fixed geometry. The sizing of the heat exchangers this time will be related to the air side exchanger only. The heat exchanger consists of two heat exchangers arranged in "V". The procedure for the choice of heat exchangers was the same as for the VMC (A-A).

The power to be exchanged at the air side heat exchanger is determined by the power analysis (useful effect) of the plate heat exchanger. The power for the sizing is then obtained as the difference or the sum of the powers of the useful effect with that absorbed by the compressor.

In order not to burden the text and having already explained in detail the procedure, it was decided to report directly the results of the heat exchangers chosen.

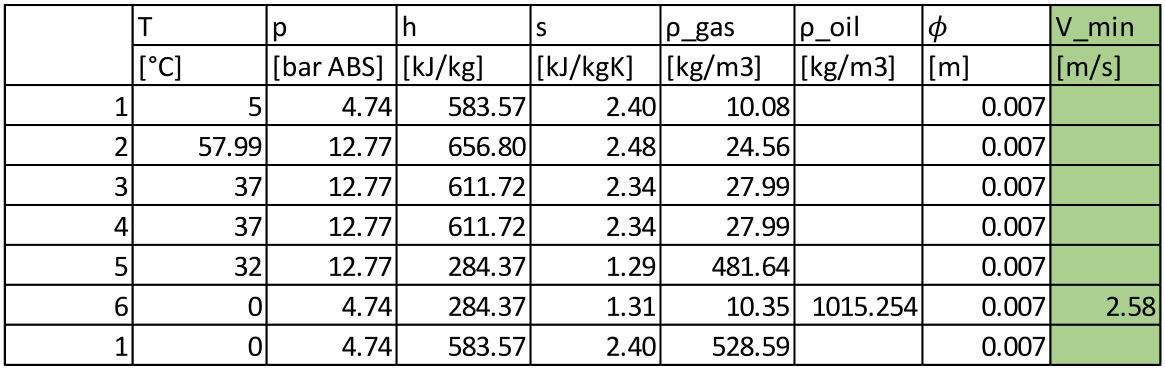
The simulation conditions at nominal load (per 80kW) are:

	T_{cond} [°C]	T_{evap} [°C]	T_{ext} [°C]	RH [%]	P_{exch} [kW]
Cooling condition	47	5	35	47	50
Heating condition	37	0	7	87	30

Table 19: simulation condition scroll at nominal load

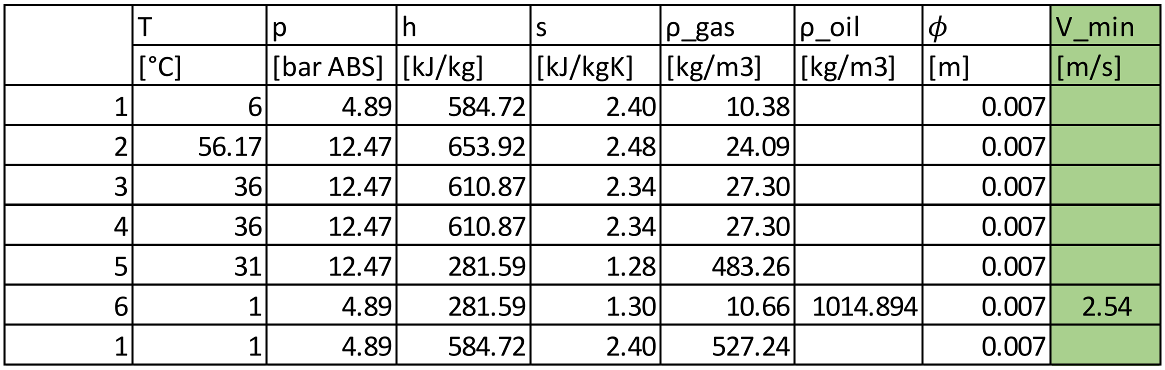
the same procedure as seen before is used. We use the same formula for calculating the minimum speed to ensure oil return. Obviously, the operating points and the density of the oil will be different from the previous case.

$$V_{min,gas} = \left(9.8\ \phi_D\ \left[\frac{\rho_{oil} - \rho_{gas}}{\rho_{gas}}\right]\right)^{0.5}\ \left[\frac{m}{s}\right]$$



	T	p	h	s	ρ_gas	ρ_oil	ϕ	V_min
	[°C]	[bar ABS]	[kJ/kg]	[kJ/kgK]	[kg/m3]	[kg/m3]	[m]	[m/s]
1	5	4.74	583.57	2.40	10.08		0.007	
2	57.99	12.77	656.80	2.48	24.56		0.007	
3	37	12.77	611.72	2.34	27.99		0.007	
4	37	12.77	611.72	2.34	27.99		0.007	
5	32	12.77	284.37	1.29	481.64		0.007	
6	0	4.74	284.37	1.31	10.35	1015.254	0.007	2.58
1	0	4.74	583.57	2.40	528.59		0.007	

Figure 64: Resume of calculation minimum velocity of refrigerant for scroll heat pump at full load



	T	p	h	s	ρ_gas	ρ_oil	ϕ	V_min
	[°C]	[bar ABS]	[kJ/kg]	[kJ/kgK]	[kg/m3]	[kg/m3]	[m]	[m/s]
1	6	4.89	584.72	2.40	10.38		0.007	
2	56.17	12.47	653.92	2.48	24.09		0.007	
3	36	12.47	610.87	2.34	27.30		0.007	
4	36	12.47	610.87	2.34	27.30		0.007	
5	31	12.47	281.59	1.28	483.26		0.007	
6	1	4.89	281.59	1.30	10.66	1014.894	0.007	2.54
1	1	4.89	584.72	2.40	527.24		0.007	

Figure 65: Resume of calculation minimum velocity of refrigerant for scroll heat pump at partial load

After these calculation, we are going to extract all the speeds that the software provides us if each tube of the circuit. Comparing the refrigerant speeds in the tubes with the minimum required speed for the calculated oil return we get the following graphs.

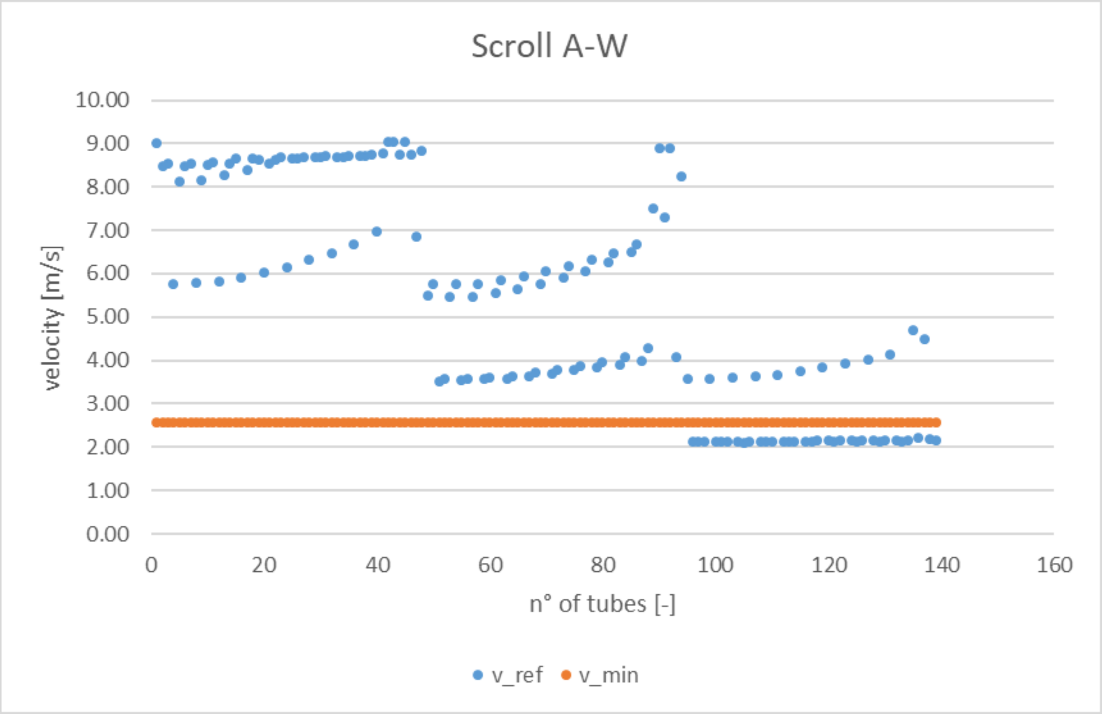


Figure 66: Comparison minimum refrigerant velocity – simulated refrigerant velocity of scroll heat pump at full load

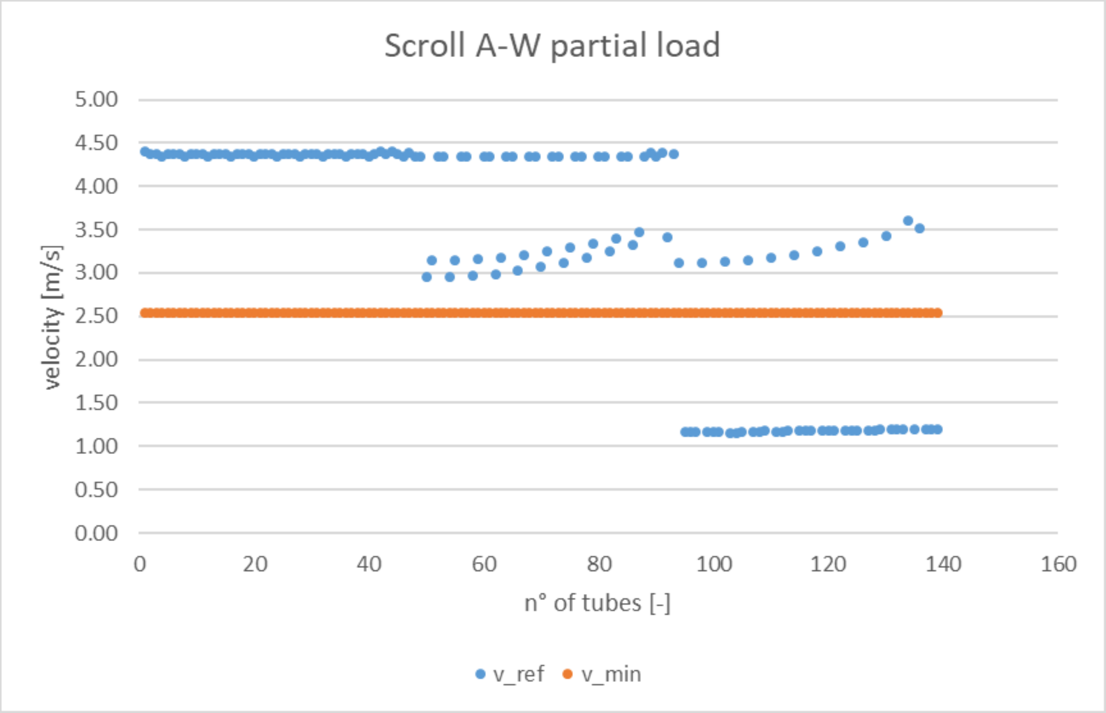
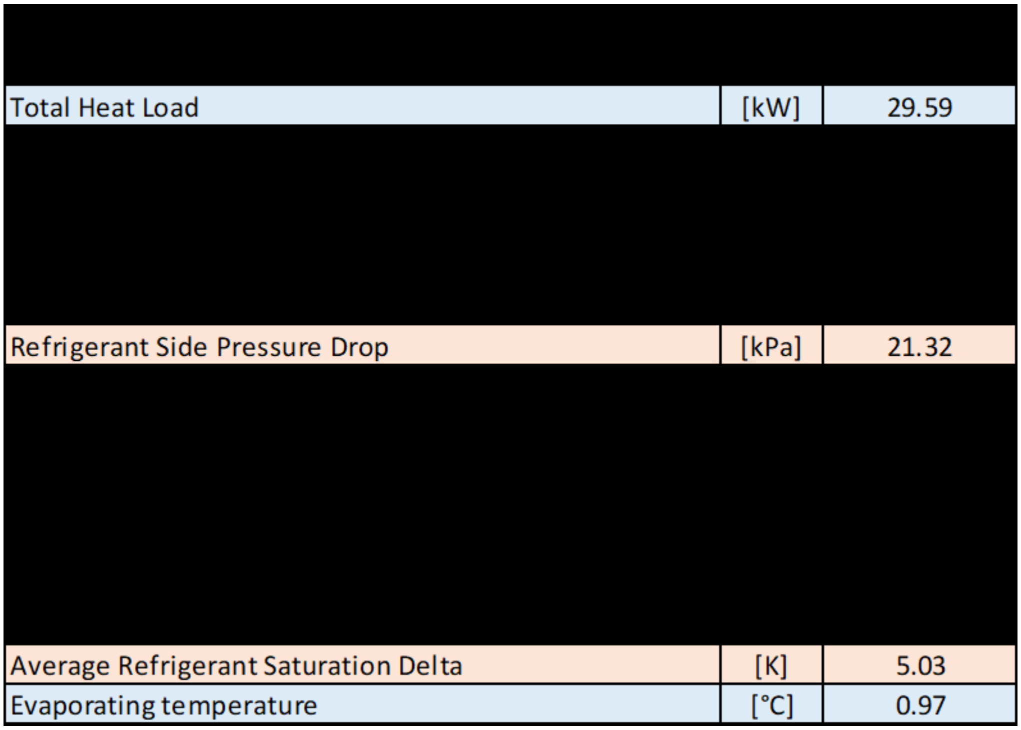


Figure 67: Comparison minimum refrigerant velocity – simulated refrigerant velocity of scroll heat pump at partial load

As in the previous case, we have set the density conditions of the refrigerant in the worst condition and that is when the density considered is equal to that of the steam phase only. Remember that this reasoning was performed because we know that as the density increases the minimum speed required for the return of oil decreases, so to be cautionary we decided to consider only steam phase throughout the evaporator.

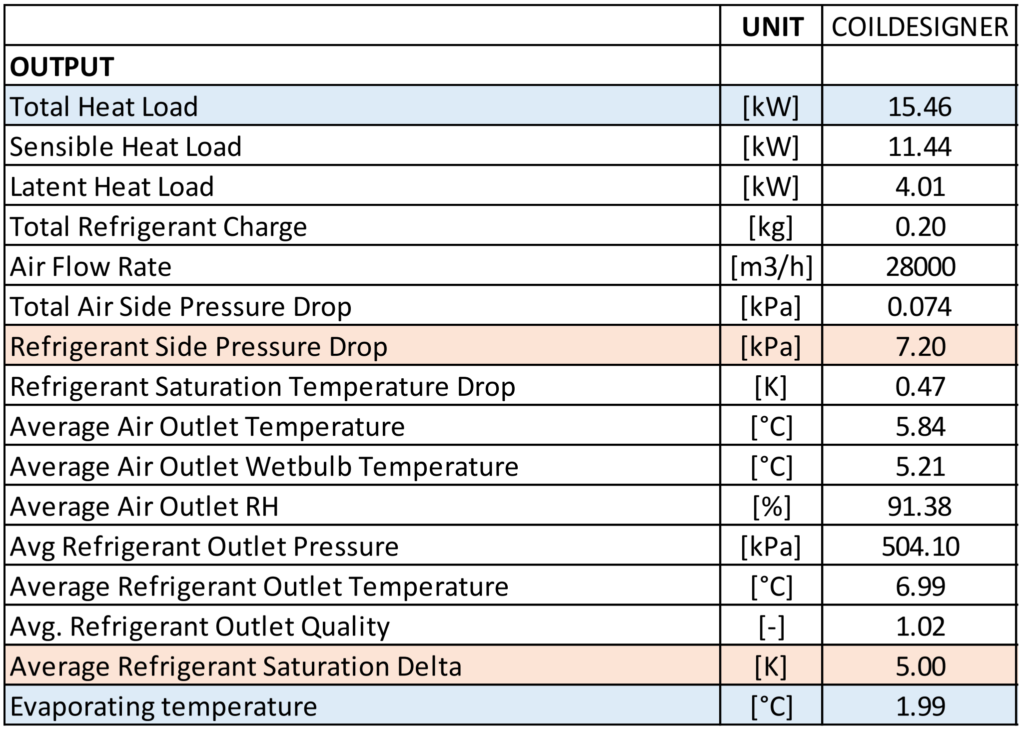
We see that some pipes from the 100th onwards do not comply with the imposed conditions, these are distributed in groups of 3 and are exactly the input pipes of each circuit. These results do not surprise us because we know that in those tubes we mainly have liquid R290.

In both graphs, we see the same speed trend in the first two tubes of each circuit. At nominal load we see that we reach a maximum of 9.5 m/s, while at partial load we do not exceed 4.5m/s and the last tubes of the various circuits all behave equally, in fact we see a line almost straight and constant composed of the last two tubes of each circuit.



[redacted]	[redacted]	[redacted]
Total Heat Load	[kW]	29.59
[redacted]	[redacted]	[redacted]
Refrigerant Side Pressure Drop	[kPa]	21.32
[redacted]	[redacted]	[redacted]
Average Refrigerant Saturation Delta	[K]	5.03
Evaporating temperature	[°C]	0.97

Figure 68: Data from CoilDesigner simulation at full load



	UNIT	COILDESIGNER
OUTPUT		
Total Heat Load	[kW]	15.46
Sensible Heat Load	[kW]	11.44
Latent Heat Load	[kW]	4.01
Total Refrigerant Charge	[kg]	0.20
Air Flow Rate	[m3/h]	28000
Total Air Side Pressure Drop	[kPa]	0.074
Refrigerant Side Pressure Drop	[kPa]	7.20
Refrigerant Saturation Temperature Drop	[K]	0.47
Average Air Outlet Temperature	[°C]	5.84
Average Air Outlet Wetbulb Temperature	[°C]	5.21
Average Air Outlet RH	[%]	91.38
Avg Refrigerant Outlet Pressure	[kPa]	504.10
Average Refrigerant Outlet Temperature	[°C]	6.99
Avg. Refrigerant Outlet Quality	[-]	1.02
Average Refrigerant Saturation Delta	[K]	5.00
Evaporating temperature	[°C]	1.99

Figure 69: Data from CoilDesigner simulation at partial load

Analysis of the results

As mentioned above, we have only reported the heat exchanger chosen for the prototype and not all the configurations studied. Let’s analyse this exchanger now.

The exchanger consists of 3 ranks with 46 tubes for each rank with 33 circuits. The outer diameter of the tubes is 7mm with a length of tube 2950 mm. We also analysed the configurations at 5mm diameter, but the results obtained with these geometries did not produce acceptable values with our selection process. In particular, the geometries with a smaller diameter had the same problems as the 4mm diameter in the unit previously analysed: a pressure drop refrigerant side too high and so the refrigerant speed also too much high.

In addition, in this case we preferred to analyse the exchanger during the evaporator operation because we know to be the most critical phase regarding the drag oil problem. It has been designed in such a way as to favour evaporation and therefore the circuits are arranged with a trend from the bottom (inlet) to the top (outlet). Remember that this configuration is also optimal for condenser operation as it allows the steam phase to condense and slide down.

The air flow and the refrigerant flow are counter current to promote maximum heat exchange. We tried to have a circuitry as symmetrical as possible between the various circuits, but in the first circuit (starting from the top) and in the last two, we have different circuitry than the others, with more tubes per circuit. This is not necessarily negative because, in the last two circuits, during the evaporation phase, we know that the refrigerant in the liquid phase is greater than the steam phase, In addition, these two are the most distant from the fans and therefore the air flow through them will be less than the next. Having a slightly larger amount of tubes means having longer tubes to carry out the exchange and thus ensure complete evaporation, in many cases this choice is made to counteract possible inefficiencies and to ensure evaporation even in the most disadvantaged circuits and with less air flow. Also in this case we wanted to have distributor and collector on the same side of the exchanger and to do this you need to have an even number of tubes per circuit(e.g.: 2-4-6-8-etc).

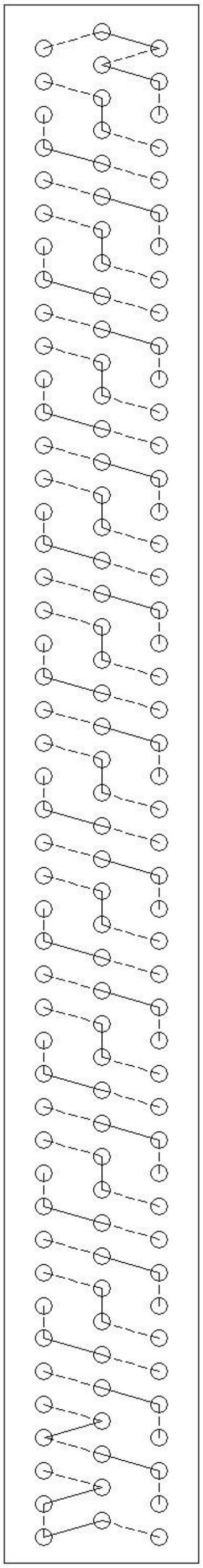


Figure 70: Scroll heat exchanger circuitry

As we can see from the image above, on the right we find the inlet tube for each circuits and on the left the outlet. Air passes through the exchanger from left to right. The connections between continuous line tubes indicate that the connection between tubes is in the front, where the observer is, while the dotted lines indicate that the connection is in the rear of the exchanger.

From the graphs of the minimum refrigerant speeds for oil return we can see that the heat exchanger does not present problems for the return of the oil except in the first tube of each circuit. This is acceptable if we consider that in that tube we have mainly liquid phase and therefore the problem of oil drag is ensured by the liquid refrigerant.

In general, we see that the refrigerant charge required is quite high, but also acceptable as we are dealing with a unit of several tens of kilowatts. Remember that this unit is placed outside so it does not have to be subject to maximum limits of refrigerant charge. However, having a unit with a low amount of refrigerant charge and with high yields and COP is convenient as an economic factor takes over.

Regarding the pressure drop refrigerant side, we see that they are in a range that we can feel good as they are high enough to have a good cooling speed and therefore good return of the oil and are low enough not to create inefficiencies and therefore go to affect the surrender.

We want to remember the unit will have two heat exchangers on the air side, but the study is conducted on one of the two heat exchangers because the other will be arranged symmetrically with respect to the latter and therefore will have the same behaviour.

5.5 Screw air to water heat pump (A-W)

Also in this unit, we size the fin and tube exchangers that exchange heat with air. In this case, the heat exchangers are 4 and are positioned at "M inverted", this configuration is used to optimize the heat exchange. The procedure for choosing the heat exchangers is the same as the previous ones; we size a heat exchanger, so the power to be disposed of by the heat exchangers is divided by 4. We report only the final heat exchanger chosen for the unit prototype.

The simulation conditions are as follows:

	T_{cond} [°C]	T_{evap} [°C]	T_{ext} [°C]	RH [%]	P_{exch} [kW]
Cooling condition	50	3	35	47	182
Heating condition full load	48	-3	7	87	97
Partial load	45	5	7	87	45

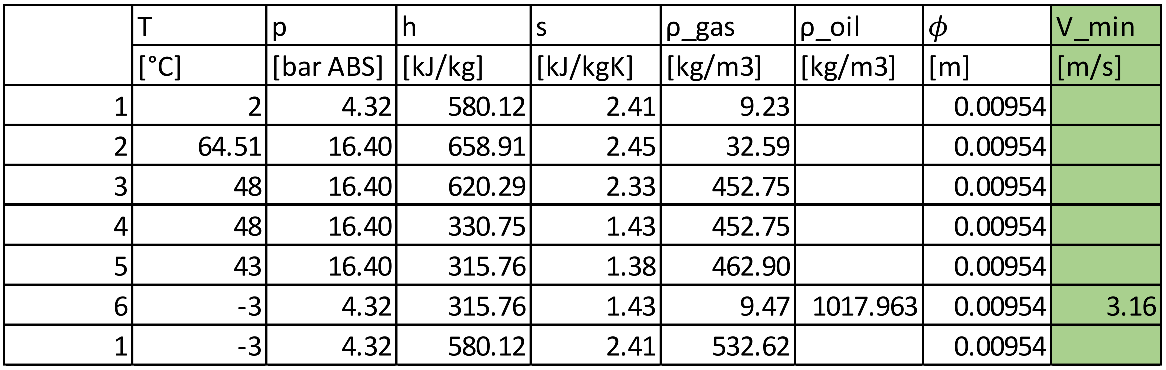
Table 20: Simulation condition for screw heat pump

Unlike the VMC heat pump air to air, it is preferred to have distributors and collectors on the opposite sides of the exchangers, with the distributors closer to the mechanical part of the circuit and therefore closer to the compressors. The length of the exchangers is significantly high (5.85m).

As seen above, we must assess whether the coolant speed is sufficient to ensure the return of the oil during operation by evaporator. We use the same empirical formula:

$$V_{min,gas} = \left(9.8\ \phi_D\ \left[\frac{\rho_{oil} - \rho_{gas}}{\rho_{gas}}\right]\right)^{0.5} \left[\frac{m}{s}\right]$$

In this case, the diameter of the tubes is significantly greater, in fact we consider the diameter of 3/8" which corresponds to 9.525 mm. The gas density is calculated at the inlet to the evaporator immediately after the expansion process. The cycle is evaluated with the REFPROP program under the above conditions. The following results are obtained:

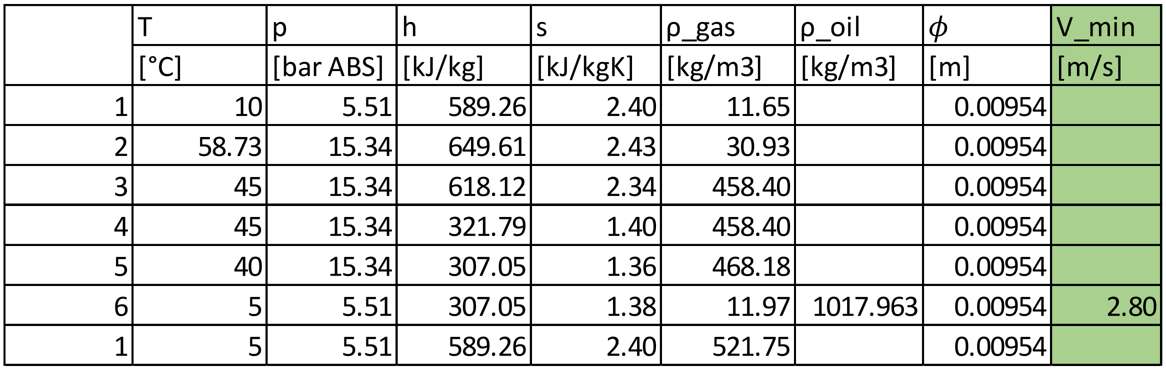


	T	p	h	s	ρ_gas	ρ_oil	ϕ	V_min
	[°C]	[bar ABS]	[kJ/kg]	[kJ/kgK]	[kg/m3]	[kg/m3]	[m]	[m/s]
1	2	4.32	580.12	2.41	9.23		0.00954	
2	64.51	16.40	658.91	2.45	32.59		0.00954	
3	48	16.40	620.29	2.33	452.75		0.00954	
4	48	16.40	330.75	1.43	452.75		0.00954	
5	43	16.40	315.76	1.38	462.90		0.00954	
6	-3	4.32	315.76	1.43	9.47	1017.963	0.00954	3.16
1	-3	4.32	580.12	2.41	532.62		0.00954	

Figure 71: Calculation minimum refrigerant velocity screw heat pump at full load

We see that the minimum speed required is significantly higher than the other units in fact, as seen in the formula, the V_min is proportional to the diameter of the tube.

The table on partial load operation is shown below.



	T	p	h	s	ρ_gas	ρ_oil	ϕ	V_min
	[°C]	[bar ABS]	[kJ/kg]	[kJ/kgK]	[kg/m3]	[kg/m3]	[m]	[m/s]
1	10	5.51	589.26	2.40	11.65		0.00954	
2	58.73	15.34	649.61	2.43	30.93		0.00954	
3	45	15.34	618.12	2.34	458.40		0.00954	
4	45	15.34	321.79	1.40	458.40		0.00954	
5	40	15.34	307.05	1.36	468.18		0.00954	
6	5	5.51	307.05	1.38	11.97	1017.963	0.00954	2.80
1	5	5.51	589.26	2.40	521.75		0.00954	

Figure 72: Calculation minimum refrigerant velocity screw heat pump at partial load

From the simulations, carried out through Coildesigner software, the speed of the tube coolant is evaluated and graphited according to the number of tubes.

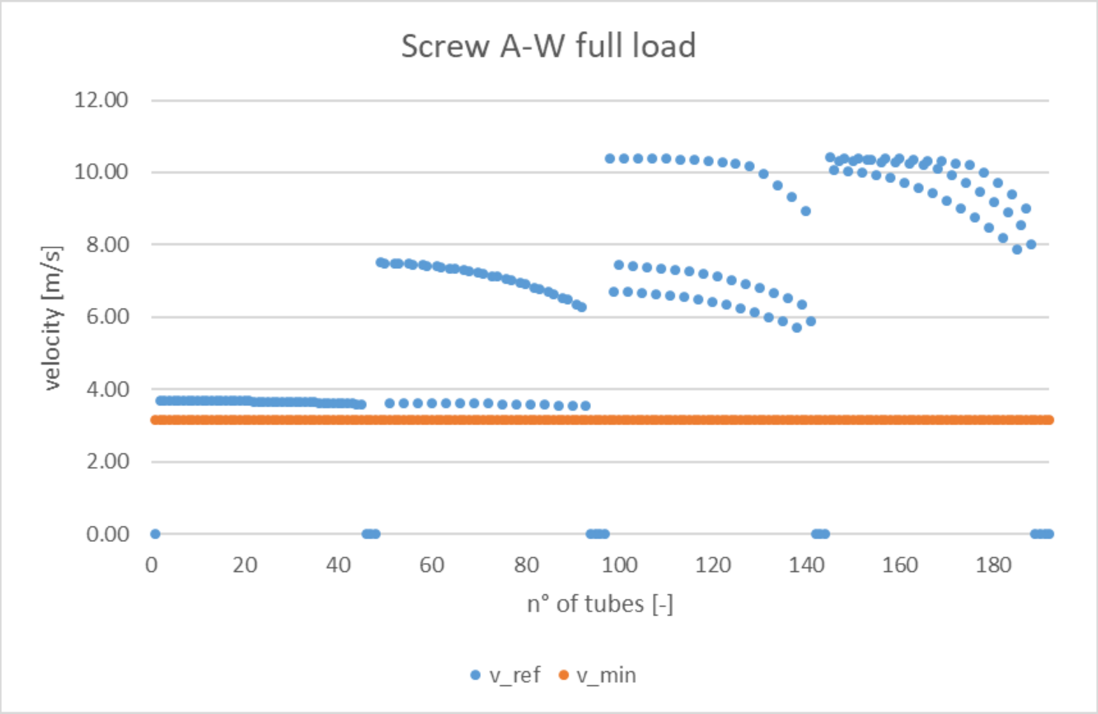


Figure 73: Comparison minimum refrigerant velocity - simulated refrigerant velocity of screw heat pump at full load

At nominal load we see that all the tubes meet the minimum speed required, the tubes that have a speed of 0 m/s, are the tubes excluded that, for geometry reasons, are not crossed by the fluid.

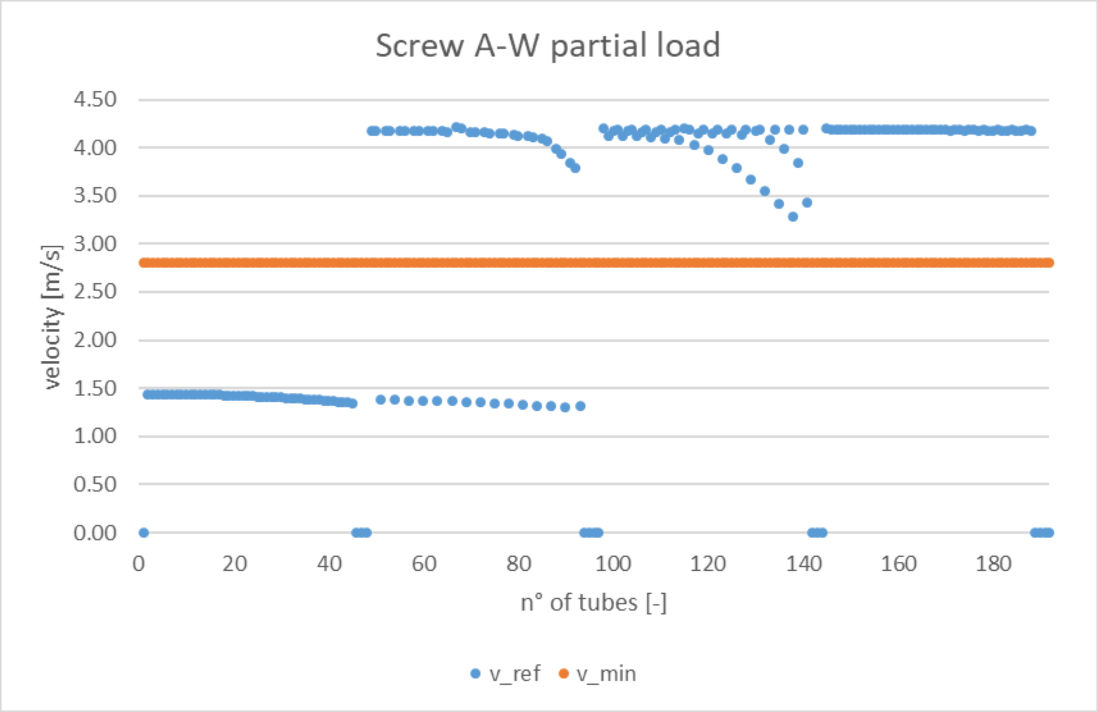
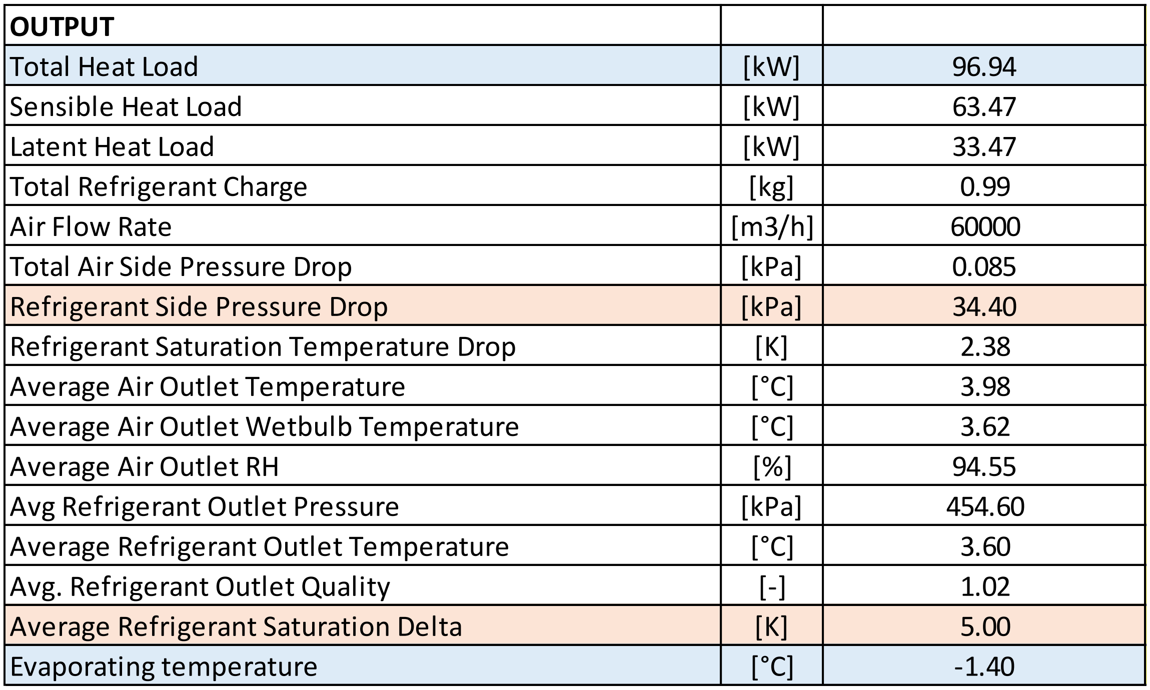


Figure 74: Comparison minimum refrigerant velocity - simulated refrigerant velocity of screw heat pump at partial load

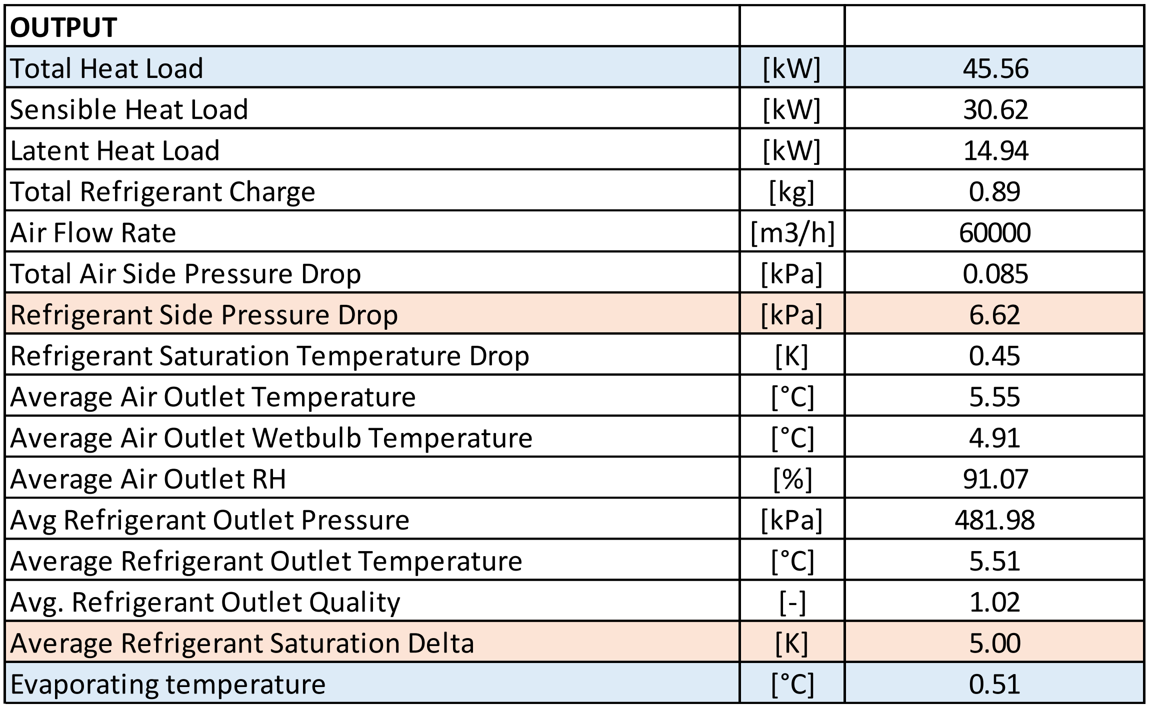
At partial load, we can see that some tubes do not meet the minimum speed required. In particular, the speeds below 2.8 m/s are the input speeds of each circuit. This result is acceptable as we remember that the minimum speed and the density are evaluated in the worst conditions and that is considering at the entrance of each circuit only steam phase, while in reality we know have more liquid phase than steam.

Below are the output values from Coildesigner for nominal and partial load operation:



OUTPUT		
Total Heat Load	[kW]	96.94
Sensible Heat Load	[kW]	63.47
Latent Heat Load	[kW]	33.47
Total Refrigerant Charge	[kg]	0.99
Air Flow Rate	[m3/h]	60000
Total Air Side Pressure Drop	[kPa]	0.085
Refrigerant Side Pressure Drop	[kPa]	34.40
Refrigerant Saturation Temperature Drop	[K]	2.38
Average Air Outlet Temperature	[°C]	3.98
Average Air Outlet Wetbulb Temperature	[°C]	3.62
Average Air Outlet RH	[%]	94.55
Avg Refrigerant Outlet Pressure	[kPa]	454.60
Average Refrigerant Outlet Temperature	[°C]	3.60
Avg. Refrigerant Outlet Quality	[-]	1.02
Average Refrigerant Saturation Delta	[K]	5.00
Evaporating temperature	[°C]	-1.40

Figure 75: Value from CoilDesigner for screw heat pump at nominal load



OUTPUT		
Total Heat Load	[kW]	45.56
Sensible Heat Load	[kW]	30.62
Latent Heat Load	[kW]	14.94
Total Refrigerant Charge	[kg]	0.89
Air Flow Rate	[m3/h]	60000
Total Air Side Pressure Drop	[kPa]	0.085
Refrigerant Side Pressure Drop	[kPa]	6.62
Refrigerant Saturation Temperature Drop	[K]	0.45
Average Air Outlet Temperature	[°C]	5.55
Average Air Outlet Wetbulb Temperature	[°C]	4.91
Average Air Outlet RH	[%]	91.07
Avg Refrigerant Outlet Pressure	[kPa]	481.98
Average Refrigerant Outlet Temperature	[°C]	5.51
Avg. Refrigerant Outlet Quality	[-]	1.02
Average Refrigerant Saturation Delta	[K]	5.00
Evaporating temperature	[°C]	0.51

Figure 76: value of CoilDesigner for screw heat pump at partial load

Analysis of the results

The chosen heat exchanger consists of 4 ranks with 40 tubes per rank, the circuits are 49. The length of each tube is 5850 mm the total height of each exchanger is 1200 mm. The study of the evolution of the air velocities passing through the exchangers has been evaluated with an air velocity profile polynomial equation, of the following type:

$$v = a + bx + cx^2 + dx^3$$

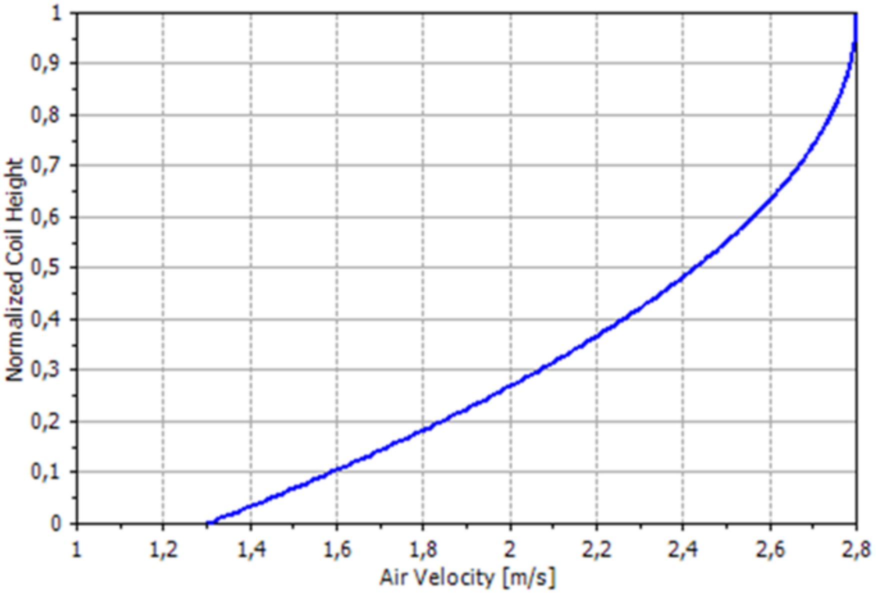


Figure 77: Graph of polynomial equation for the calculation of the air velocity pass through the heat exchanger

This process is fundamental: when the heat exchangers are very high, the tubes will be run over by different air flows depending on the tube-fan distance. In particular, the farther the tube will be, the lower the air velocity at that point and therefore the lower the heat exchange promoted by the air flow.

Below is reported the circuitry of the heat exchanger.

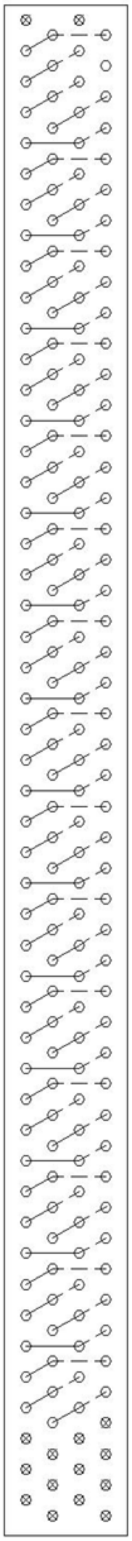


Figure 78: Screw heat exchanger circuitry

The air passes through the exchanger from left to right, the inlet of each circuit is located in the left part of the exchanger and then the exchange takes place in concurrent flow. It is possible to notice that the last tubes in the lower part are plugged because, as we have said several times, we have sized the exchanger starting from a fixed geometry in the initial conditions. It was decided not to use the tubes in the lower part of the exchanger because they are not necessary, also being the most distant from the fans, the heat exchange of these is the most disadvantaged. As we can see, the circuitry is symmetrical in groups of 4 circuits.

The values returned by the software are promising, the pressure losses on the refrigerant side are not too high and this allows to have a good performance of the heat exchangers in both evaporator and condenser operation, ensuring in any case also the minimum speed required for the return oil.

Chapter 6: Heat exchanger laboratory test

6.1 Introduction

In this chapter, the tests performed on two units in order to obtain all the necessary practical data to be compared with the theoretical data will be reported. The standard data collection procedure will also be described.

To perform data collection, you must have:

- Equipment
- Software data collection (synoptic)
- Manual data collection (e.g.,÷ air heat exchanger tracking, heat exchanger temperature tracking)

6.2.1 Instruments/probes

You can divide the instruments and probes based on their position.

Mounted directly on the unit's components:

- Thermocouples on the circuit in contact with the tubes
- Bulb probes for both air and water measurement
- Pressure probes, high and low pressure transducers

Instruments for measuring purposes:

- Contact thermometer
- Thermal camera
- Anemometer

Sensors in the room:

- Temperature and relative humidity meter
- Δp and water flow meters

Various types of sensors are placed on the units near the points of interest. In particular, temperature and pressure probes are used. These probes return the pressure and temperature values, but in some cases, for a more in-depth analysis, industrial instrumentation has also been used for a more precise evaluation of the individual circuits.

6.2.2 Software data collection - Room performance test

The most important test to set comparisons between theoretical data calculated by software described in Chapter 5 and real data is the performance test of the room.

Objective: To verify that the unit complies with the performance and operation values of the project leader, in compliance with the directives.

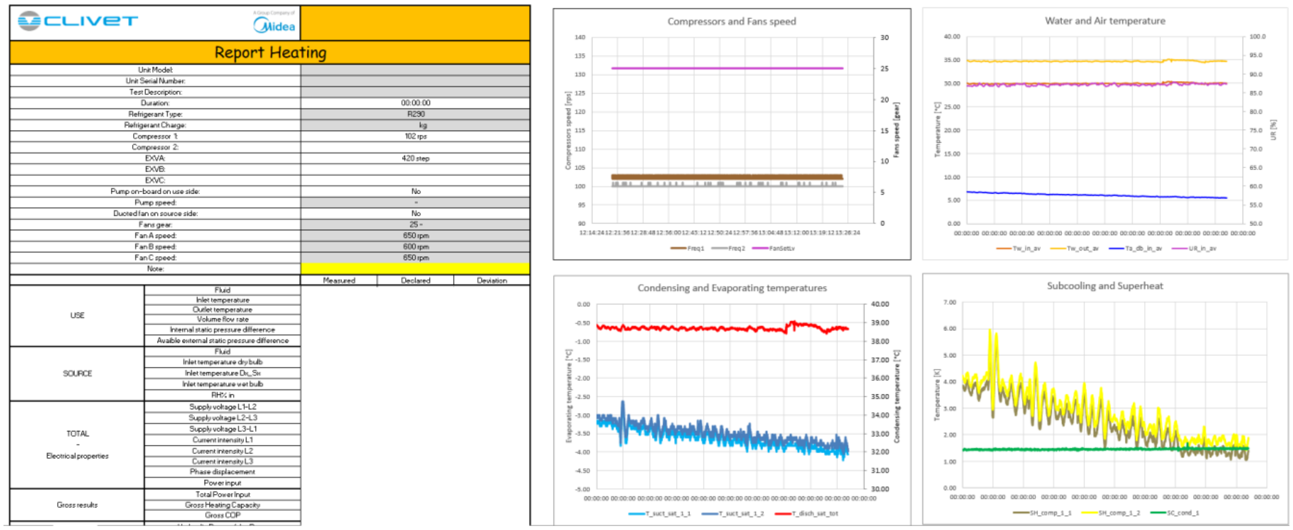
The sequences of the operational phases for the tests starting from the prototype area are defined below. The Test Room operator performs the following activities. For all units produced in the prototype area, the testing activities are carried out directly in the test room. The general procedure is divided into several steps:

- Connecting the unit in the test room: In this phase, the pressure transducers, temperature and humidity probes are connected to the test point circuit.
- Connect the unit to power supplies.
- Instrument calibration is checked: Each temperature and pressure sensor on board the unit is compared with the temperature and pressure sensors of the test stations. This control is carried out by carrying out two specific phases, the first of "calibration probes" verification with the unit in OFF state and with sensors at rest and the second of verification with the unit in operation. During the verification phases, if there are inconsistent temperatures between the sensors placed in the unit and the sensors, checks are carried out to determine which sensors are stuck or misplaced.
- Activation test: the software is started in the test room where the values of refrigerant type, project name, are set and then activated the "Synoptic". Then the software automatically connects to the machine.
- Set the test conditions provided by the project leader with the qualification plan.
- Activate the rehearsal room.
- Activate test adjustment.
- Turn on the power supply of the unit.
- Select the water and air temperature graph display.
- Using the graph check the stability of air and water temperatures.

You get a window where schematically you can see the circuit as we drew it, with the relative pressure and temperature measurements of each component inside the unit.

Obviously thanks to the synoptic you can, at a glance, evaluate on the unit is in line with the data we expect to obtain during the test.

Once the temperature conditions are stable, the test is recorded and then the data is reported in a template "Report" of easier reading. Below is an example of a test run.



Report Heating

Unit Model:	
Unit Serial Number:	
Test Description:	
Duration:	00:00:00
Refrigerant Type:	R290
Refrigerant Charge:	kg
Compressor 1:	102 rps
Compressor 2:	
EXVA:	420 step
EXVB:	
EXVC:	
Pump on-board on use side:	No
Pump speed:	-
Ducted fan on source side:	No
Fans gear:	25 -
Fan A speed:	650 rpm
Fan B speed:	600 rpm
Fan C speed:	650 rpm
Note:	

		Measured	Declared	Deviation
USE	Fluid			
	Inlet temperature			
	Outlet temperature			
	Volume flow rate			
	Internal static pressure difference			
	Avaible external static pressure difference			
SOURCE	Fluid			
	Inlet temperature dry bulb			
	Inlet temperature Dx_Sx			
	Inlet temperature wet bulb			
	RH% in			
TOTAL - Electrical properties	Supply voltage L1-L2			
	Supply voltage L2-L3			
	Supply voltage L3-L1			
	Current intensity L1			
	Current intensity L2			
	Current intensity L3			
	Phase displacement			
	Power input			
Gross results	Total Power Input			
	Gross Heating Capacity			
	Gross COP			

Figure 79: Example of template for test report

6.2.3 Manual data collection

Many types of test are performed on the units to be tested, in particular, regarding this thesis, only those that concern the parts related to the exchanger and the factor connected with them will be explained.

Heat exchanger air flow test/tracking

This test is carried out on heat exchangers using an anemometer. The surface of the exchanger is virtually divided into areas of about 30cmx30cm and the air velocity of each section is measured. The air speed mapping should be repeated for each rotational speed of the fans and then for the gear number of the fans of interest (for example: 20-26-28-32 gear). The values are then reported in an Excel file where the average speed and the measured flow is calculated:

$$v_m = \sum_{j=1}^{n} vj \left[\frac{m}{s}\right]$$

$$\dot{m}_{air_{measured}} = S_{HE} * v_m * 3600 \left[\frac{m^3}{h}\right]$$

To get the real flow rate consider the measured minus a 10% precaution:

$$\dot{m}_{air_{real}} = \dot{m}_{air_{measured}} - 10\%\ \dot{m}_{air_{measured}} \left[\frac{m^3}{h}\right]$$

The objective is to obtain an average air flow value that passes through the exchanger according to the speed of the fans. An example of tracking is shown in the image below. As we can see, the speeds and therefore the flow rates for each of the exchangers that make up the exchanger pack are plotted, in this case divided into right and left exchangers. With the data of this test it is then possible to simulate on CoilDesigner the real conditions of the air flow through the unit's heat exchangers.

Heat exchanger temperature test/tracking

The heat exchanger temperature tracking procedure is performed with a contact thermometer and is as follows:

- Set the unit under defined conditions (example: nominal load condition or partial load condition)
- Wait until the unit to reach stable conditions
- Using a contact thermometer, the temperatures of each circuits are taken in this order: power supply- any curves of the circuit-output
- Insert these temperatures in a table and calculate the superheating and subcooling, as the difference from the temperature read between the synoptic and the outer temperature from each circuit

6.3 Organization of practical data

The data required for the comparisons to be made in Chapter 7 are given below for both units. The screw heat pump has not been analysed from the practical point of view because due to several delays in the tests, the data that would be reported would be partial and could not be used for a theoretical-practical comparison.

6.3.1 VMC heat pump (A-A)

The tests performed for the VMC heat pump (A-A) are divided in two categories:

- Heating condition
- Cooling condition

6.3.1.1 First trial

During the first laboratory tests, it was noticed that in the treatment heat exchanger (in evaporator operation) the temperatures detected by the probes in the two circuits were very different with a temperature difference between circuits of about 15 °C. This led to abnormal overheating and the heat exchanger output and in general, the unit did not reach the standard set. After several analyses, the cause of the imbalance was identified: during the design, two circuits were drawn with a difference of 4 tubes.

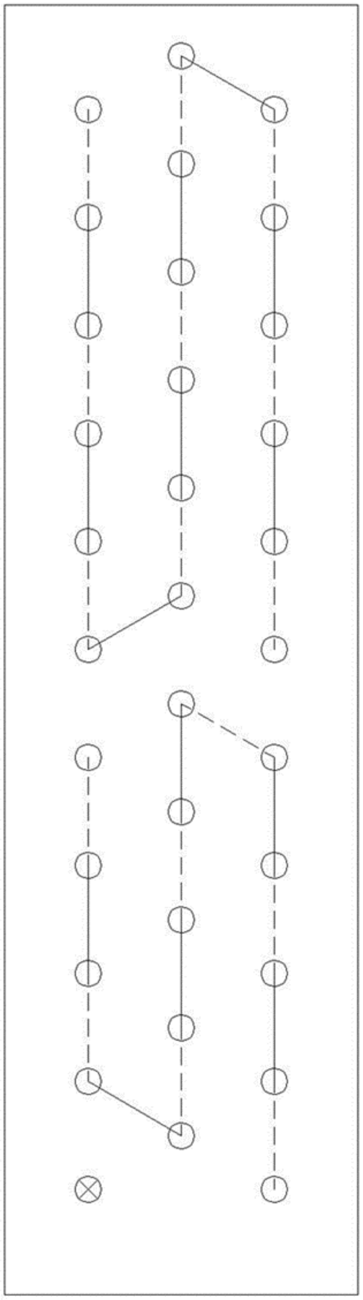
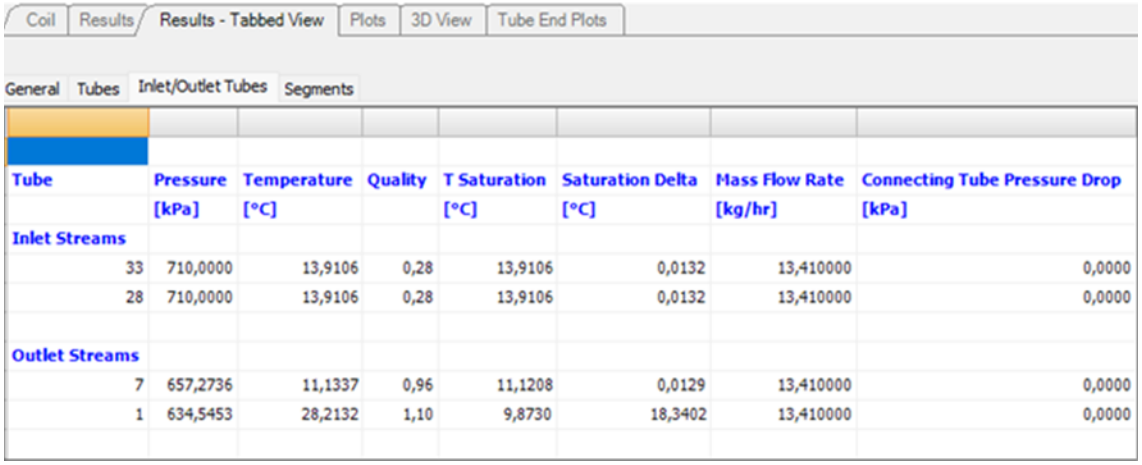


Figure 80: First trial geometry treatment heat exchanger

As we can see, the circuit in the upper part of the exchanger consists of 18 tubes, while the one below in 14 tubes. This means that, in evaporator operation, the first circuit, having more tubes, has higher overheating because, having greater heat exchange area, it can exchange more heat and therefore in the last tubes there is mainly superheated steam. In the second, you will have fewer tubes that can overheat and then the evaporation temperatures will be lower. When in an exchanger there is an imbalance in the circuits, there are negative effects in the operation.

By simulating the exchanger, the output temperatures and the saturation delta values of each circuit were sought, data that are not present at the time of data export and must be searched directly on the Coildesigner software, process that is not normally executed.



Tube	Pressure [kPa]	Temperature [°C]	Quality	T Saturation [°C]	Saturation Delta [°C]	Mass Flow Rate [kg/hr]	Connecting Tube Pressure Drop [kPa]
Inlet Streams							
33	710,0000	13,9106	0,28	13,9106	0,0132	13,410000	0,0000
28	710,0000	13,9106	0,28	13,9106	0,0132	13,410000	0,0000
Outlet Streams							
7	657,2736	11,1337	0,96	11,1208	0,0129	13,410000	0,0000
1	634,5453	28,2132	1,10	9,8730	18,3402	13,410000	0,0000

Figure 81: Inlet-outlet fluid table from CoilDesigner

As we can see the output temperatures in the first circuit (the one above) are equal to 28.2 °C and in the second of 11.3 °C. We see that the software calculates a saturation delta completely unbalanced, all the SH is processed in the first circuit with 4 more tubes.

Having identified the problem, we thought about the possible solutions:

- Redesigning the exchanger from the beginning
- Adjusting the imbalance on the exchanger already mounted on the unit

Both options have been completed. The exchanger has been redesigned with a more symmetrical geometry.

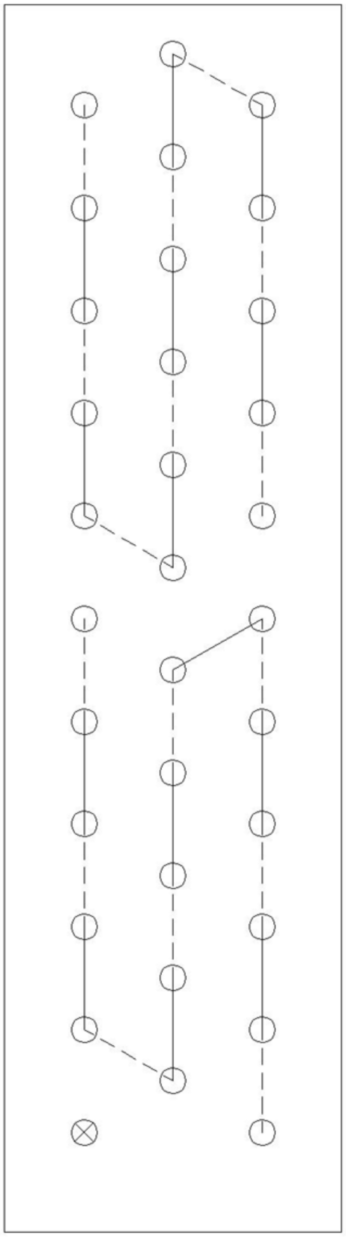
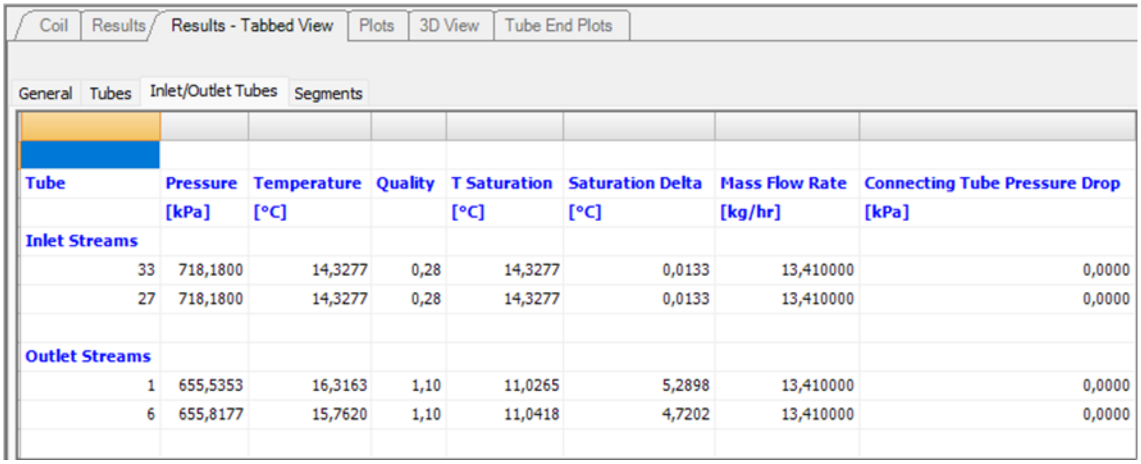


Figure 82: Redesigned geometry treatment heat exchanger

We see that the geometry above of two circuits counts 16 tubes, each making the exchanger more balanced: the first circuit consists of 5-6-5 tubes per rank and the second of 5-5-6 tubes per rank.

The results from software are also promising from the point of view of temperature in the circuits:



Tube	Pressure [kPa]	Temperature [°C]	Quality	T Saturation [°C]	Saturation Delta [°C]	Mass Flow Rate [kg/hr]	Connecting Tube Pressure Drop [kPa]
Inlet Streams							
33	718,1800	14,3277	0,28	14,3277	0,0133	13,410000	0,0000
27	718,1800	14,3277	0,28	14,3277	0,0133	13,410000	0,0000
Outlet Streams							
1	655,5353	16,3163	1,10	11,0265	5,2898	13,410000	0,0000
6	655,8177	15,7620	1,10	11,0418	4,7202	13,410000	0,0000

Figure 83: Inlet- outlet fluid table for redesigned geometry

In the image above, the temperatures are more balanced and the SH differs slightly between the two circuits.

As we know, the realization times are long and, in the meantime, it has been decided to try to modify the unbalanced exchanger mounted on the unit so that we can conduct tests and try to obtain better results. Through the experience of the technicians, it was decided to modify the exchanger going to change the input of the circuit above to obtain more tubes in the lower circuit in the exchanger. To make this operation as low impact as possible, the new exchanger is modified as follows:

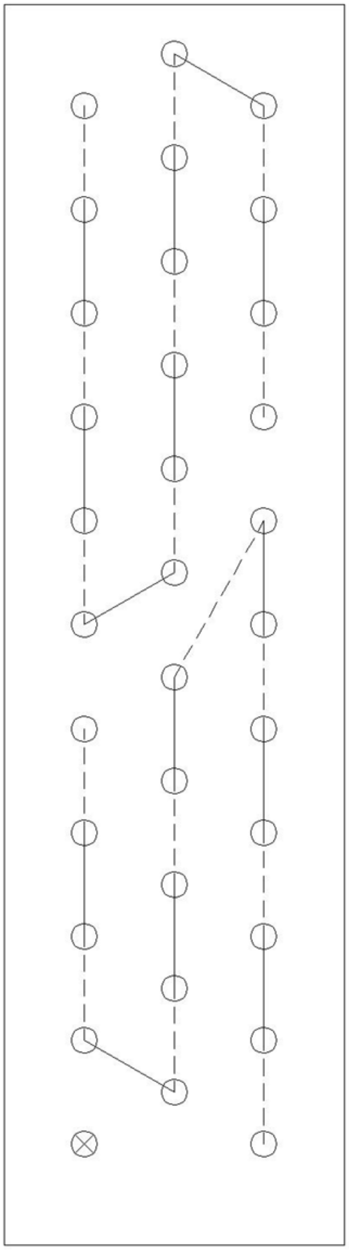
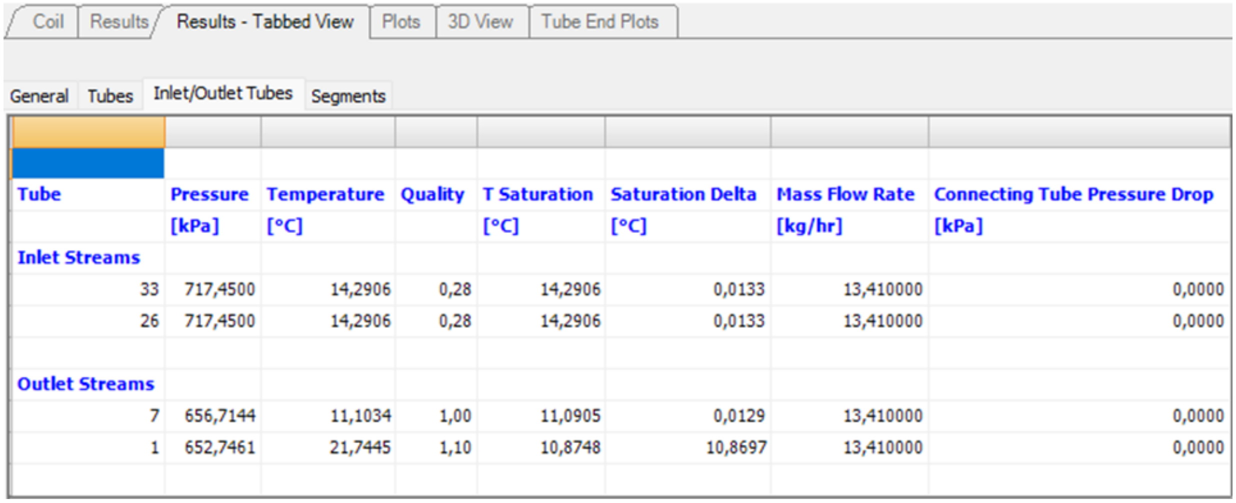


Figure 84: Modified geometry treatment heat exchanger

Therefore, we obtain two circuits of 16 tubes each but composed as follows:

- first circuit 6-6-4
- second circuit 4-5-7

This configuration is not final, and the circuitry is not symmetrical, but allows us to get from software the following results:



Tube	Pressure [kPa]	Temperature [°C]	Quality	T Saturation [°C]	Saturation Delta [°C]	Mass Flow Rate [kg/hr]	Connecting Tube Pressure Drop [kPa]
Inlet Streams							
33	717,4500	14,2906	0,28	14,2906	0,0133	13,410000	0,0000
26	717,4500	14,2906	0,28	14,2906	0,0133	13,410000	0,0000
Outlet Streams							
7	656,7144	11,1034	1,00	11,0905	0,0129	13,410000	0,0000
1	652,7461	21,7445	1,10	10,8748	10,8697	13,410000	0,0000

Figure 85: Inlet-outlet fluid table for modified geometry

As we can see the heat exchanger is still unbalanced, but the difference in the outlet temperatures of the refrigerant has decreased of 10 °C and the saturation delta has dropped by 8 °C respect to the first designed heat exchanger.

6.3.1.2 Second trial

The tests were performed on the modified exchanger seen in the previous part.

COOLING	Capacity [kW]	P_{in} [kW]	P_{fan} [kW]	T_{cond} [°C]	SH [K]	SC [K]	EEV valve opening steps
Test	1.6	0.53	0.06	50.63	2.39	10.10	160

Table 21: Cooling operation test data resume

As we can see in the cooling operation, the target power is not reached, moreover the temperatures, read by the temperature probes applied on the exchangers, detect a large temperature difference in the treatment heat exchanger of about 11°C between the two circuits. Although it was a temporary solution, in view of the more balanced heat exchanger, it was not expected that it would not reach the target power.

6.3.1.3 Third trial

Failing to get enough power to reach the target exchange power, we decided to take a step back and use the heat exchanger used on the native unit working at R32 for the treatment heat exchange while the source heat exchanger have a good performance. The external geometry is the same as before, it is composed of 3 circuits, the circuitry is shown below:

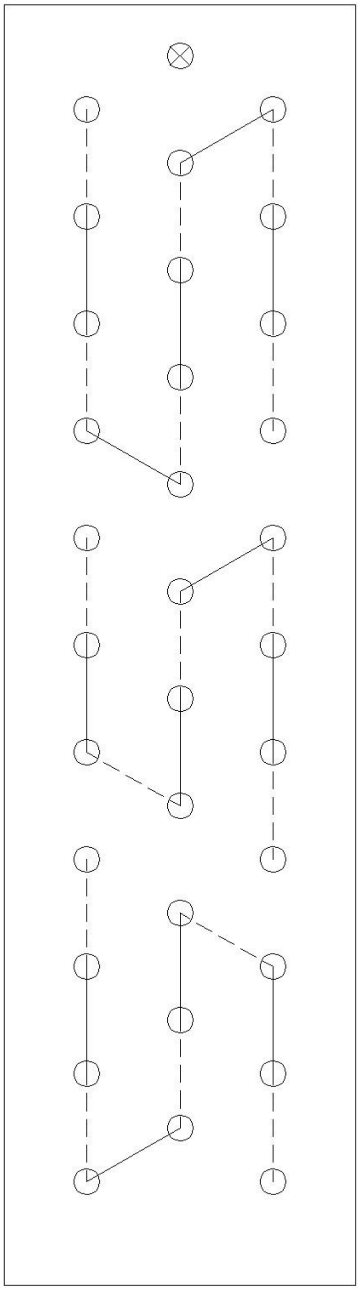


Figure 86: 3 circuits geometry treatment heat exchanger

Tests in heating condition show that the unit can reach the target power with a refrigerant charge equal to 150g. The main values of the unit in the heating test are shown in the table below.

HEATING	Capacity [kW]	P_{in} [kW]	P_{fan} [kW]	T_{cond} [°C]	SH [K]	SC [K]	EEV valve opening steps
Test 1	1.87	0.45	0.06	42.53	7.48	10.24	90
Test 2	1.88	0.48	0.06	39.95	3.23	6.23	98
Test 3	1.87	0.47	0.06	38.81	1.60	4.55	104

Table 22: Heating operation test data resume

The treatment heat exchanger works as a condenser and the source heat exchanger as an evaporator.

All tests were conducted under the same conditions and the desired capacity is reached. The only variable among the various tests was the degree of opening of the lamination valve called EEV. We see that, as the steps increase and therefore the valve opening, SC and SH go down and more and more reach acceptable values without affecting the power exchanged. The condensation temperature drops as the valve opens. The power of the fans remains unchanged keeping the air flow to the exchangers constant.

There are no large imbalances between the outlet temperatures from the exchangers. In fact, as we can see from the table below, the output temperatures of the treatment heat exchanger are around 36 ºC and the source heat exchanger temperatures are around 3.5 ºC.

HEATING	Test 1		Test 2		Test 3	
	T	S	T	S	T	S
Inlet temperature [°C]	41.44	4.33	39.06	5.05	37.95	5.19
	41.44	4.96	39.06	5.71	37.95	5.83
	41.00	5.40	38.62	6.10	37.50	6.19
Outlet temperature [°C]	41.44	4.16	39.06	2.92	37.95	2.70
	36.11	3.08	37.11	3.26	36.33	3.35
	32.35	4.99	34.33	4.79	34.72	4.59

Table 23: Temperature data test resume; T is for treatment heat exchanger, S is for source heat exchanger

Later, tests were conducted under cooling conditions. Remember that the charge in the cooling operation is usually higher than that in the heating condition because the source heat exchanger, which has twice the number of tubes compared to the treatment heat exchanger, in the operation in heating works as an evaporator, while in cooling it works as a condenser. If we look at the above equation in chapter 5:

$$Charge\ refrigerant = V\ \rho\ [kg]$$

We see that with the same volume, if the fluid evaporates it will have a lower density than in the case where it condenses.

The goal on this unit is to achieve the best possible performance with a refrigerant charge below 150g. Below are reported the values of the test conducted in the laboratory in cooling condition.

COOLING	Capacity [kW]	P_{in} [kW]	P_{fan} [kW]	T_{evap} [°C]	SH [K]	SC [K]	EEV valve opening steps
Test 1	1.94	0.56	0.06	11.00	8.13	8.46	112
Test 2	1.87	0.56	0.06	11.72	4.00	5.19	128

Table 24: Cooling operation test resume

The tests were conducted at stable room conditions where it is possible to simulate an external temperature of 35°C with 40% relative humidity and the conditions inside the room to be air-conditioned of 27°C with 50% relative humidity.

The two tests are similar to each other, to adjust the SC and SH the expansion valve is controlled, increasing the opening of the valve we see that the subcooling and superheating values drop significantly. A lower SC also involves a lower heat exchange to the evaporator because if we evaluate the power exchanged to the evaporator as:

$$Q = \dot{m}\ \Delta h\ [W]$$

we see that as the SC decreases, the Δh decreases and therefore also the heat exchange decreases. However, the opening of the expansion valve also allows to make the temperatures to the evaporator more homogeneous, below are reported the values of the inlet and outlet temperatures of each circuits of treatment heat exchanger working as evaporator and source heat exchanger working as condenser related to Test 1 and Test 2:

COOLING	Test 1		Test 2	
	Treatment H.E.	Source H.E.	Treatment H.E.	Source H.E.
Inlet temperature [°C]	12.02	56.50	12.83	52.27
	13.21	55.42	14.05	51.61
	12.83	55.80	13.70	52.34
Outlet temperature [°C]	22.44	45.88	22.78	44.92
	19.73	46.50	17.66	46.02
	12.35	38.39	13.16	43.12

Table 25: Temperature data test resume

It is possible to note that in general the last circuit of the source exchanger, which works as a condenser, tends to subcooling predominantly in Test 1, while in Test 2 the opening of the expansion valve allows to mitigate and thus make the last circuit less flooded. Remember that if in the condenser tubes we find a lot of liquid it means that that tube exchanges much less heat than the others of the circuit and this means that the power to be disposed of is divided between the remaining tubes of the circuit.

6.3.1.4 Temperature tracking test

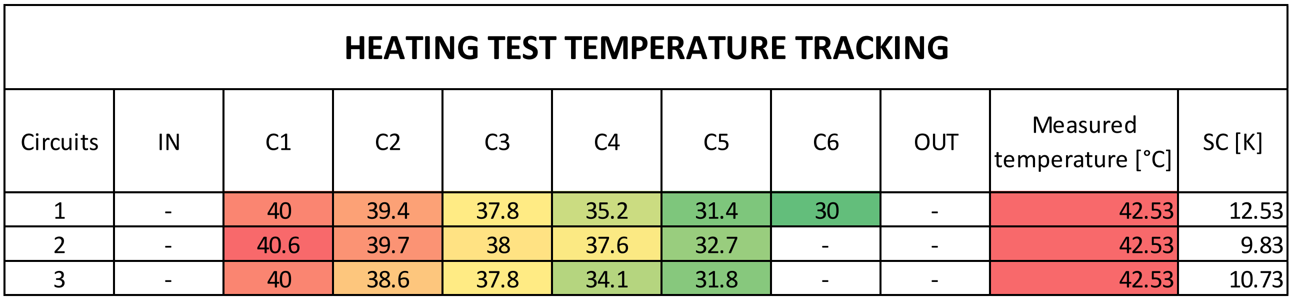
Tests were carried out to trace the temperatures of the heat exchangers in the heating conditions. As we can see from the images below, the tests were conducted in the configuration where both the treatment heat exchanger and the source heat exchanger have three circuits, which is the configuration on which comparisons will be made in chapter 7.

Temperature tracking on this unit has been complicated because both exchangers have one side adjacent to the unit wall and the other side is very close to other components. With a contact thermometer, it is difficult to do temperature detection, so we cut a part of the outer layer of EPP (Expanded polypropylene) so that you can access the curves that are between two consecutive tubes of each circuit.

Figure 87: Source heat exchanger view

The copper curves seen in the imagine above are the points where the temperatures of the refrigerant have been traced, with a contact thermometer, during operation in heating and reported here are in the table and later in the circuit diagram of the exchangers.

Treatment heat exchanger



HEATING TEST TEMPERATURE TRACKING										
Circuits	IN	C1	C2	C3	C4	C5	C6	OUT	Measured temperature [°C]	SC [K]
1	-	40	39.4	37.8	35.2	31.4	30	-	42.53	12.53
2	-	40.6	39.7	38	37.6	32.7	-	-	42.53	9.83
3	-	40	38.6	37.8	34.1	31.8	-	-	42.53	10.73

Figure 88: Temperature tracking data collection for treatment heat exchanger

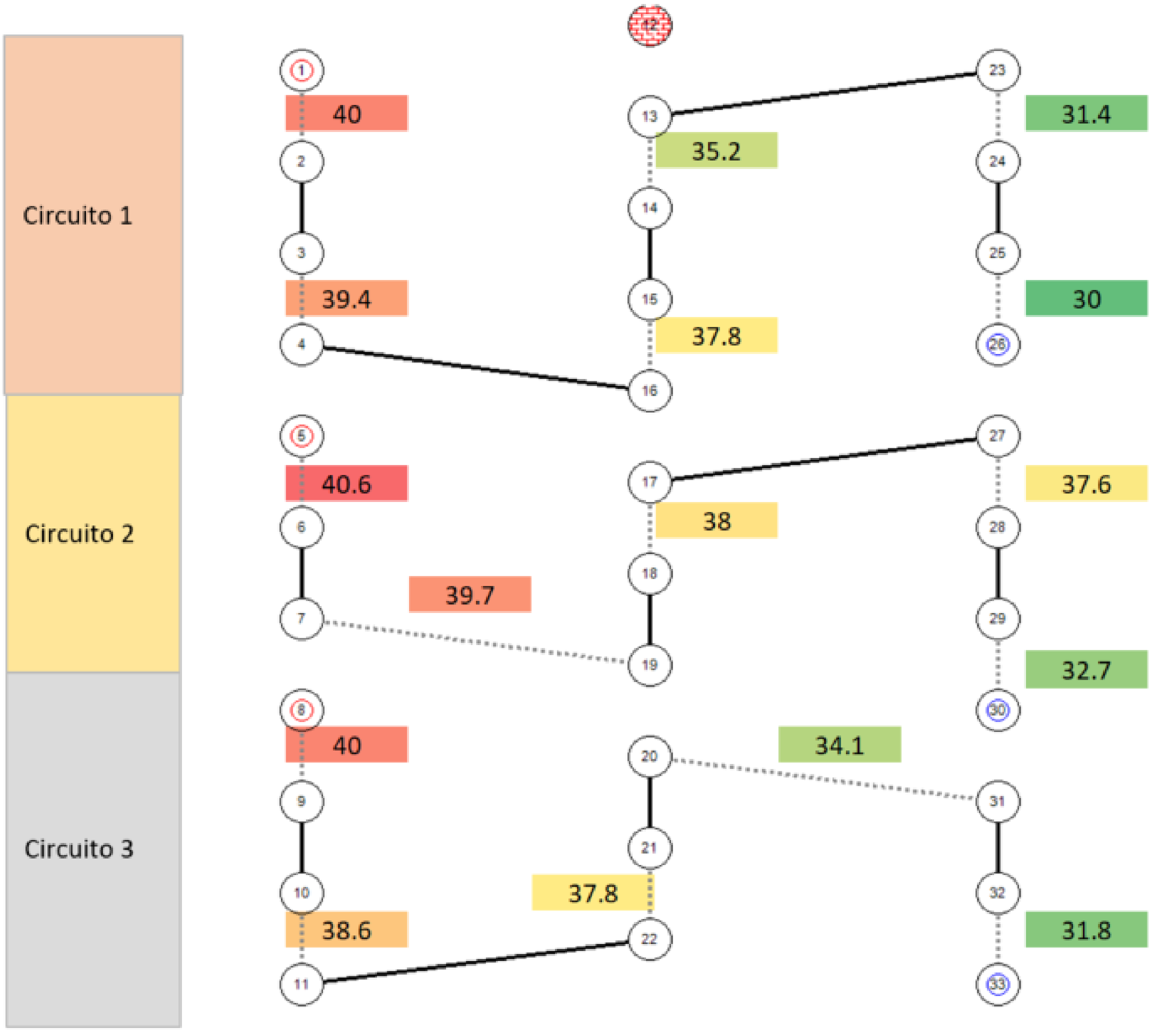
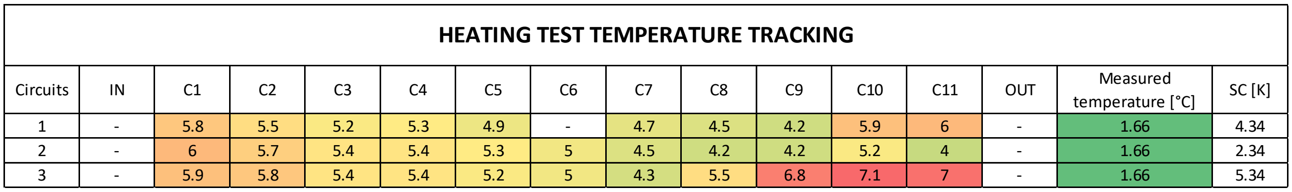


Figure 89: Temperature tracking data collection for treatment heat exchanger scheme

Source heat exchanger



HEATING TEST TEMPERATURE TRACKING

Circuits	IN	C1	C2	C3	C4	C5	C6	C7	C8	C9	C10	C11	OUT	Measured temperature [°C]	SC [K]
1	-	5.8	5.5	5.2	5.3	4.9	-	4.7	4.5	4.2	5.9	6	-	1.66	4.34
2	-	6	5.7	5.4	5.4	5.3	5	4.5	4.2	4.2	5.2	4	-	1.66	2.34
3	-	5.9	5.8	5.4	5.4	5.2	5	4.3	5.5	6.8	7.1	7	-	1.66	5.34

Figure 90:: Temperature tracking data collection for source heat exchanger

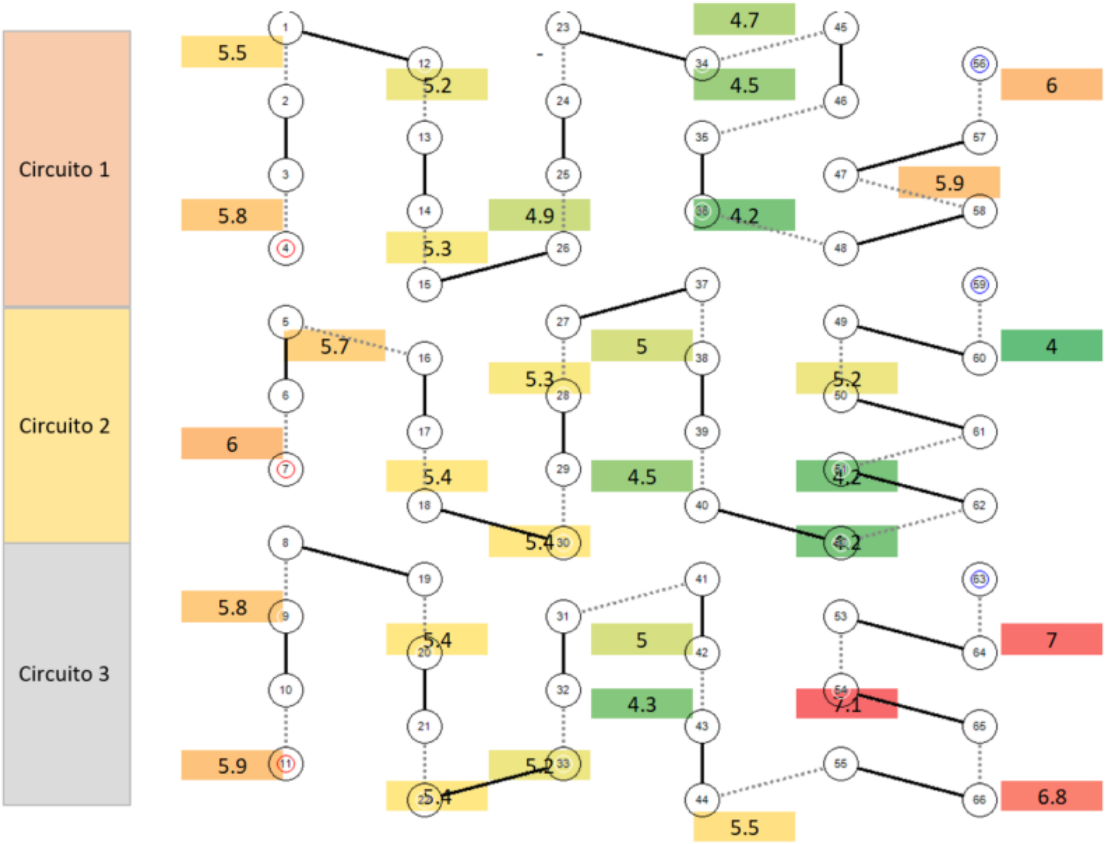


Figure 91: Temperature tracking data collection for source heat exchanger

These are the temperatures measured at every point accessible by the contact thermometer. It is possible to notice that the trends of the temperatures are almost similar between the circuits. In the source heat exchange, the third circuit reaches higher temperatures towards the exit than the other two. This difference is due to the different geometry of the last part of the circuit that allows exchanging more heat with the air.

6.3.2 Scroll air to water heat pump

For this unit we have reported only a small part of the data obtained from the reports of the various tests. The various tests were carried out under the conditions of heating and cooling and then with the heat exchangers respectively considered in evaporator and condenser.

Heating

Total heat load is calculated as follows:

The capacity, the input power and the fan gear relative to the test are read from the report or synoptic. The absorbed power of the fans is calculated with an experimental equation that links the gear (speed of the fan) to the power absorbed:

$$P_{fan} = 0.017 \exp(0.15 * gear) \ [kW]$$

The Total heat load equation is calculated by:

$$Total\ heat\ load = \frac{\left(Capacity - \left(P_{input} - 3P_{fan}\right)\right)}{2} [kW]$$

The P_{fan} value is multiplied by three because all the fans active on the unit are considered and everything is divided by two because during the calculations the simulations are conducted on the single heat exchanger.

The refrigerant pressure drop is calculated as the difference in refrigerant pressure between the inlet and outlet of the heat exchanger, usually the probes/sensor return the measurement in bars.

The evaporating temperature is calculated as the average of the saturated evaporation temperatures of the exchangers.

The average refrigerant saturation delta is calculated as the difference between the average evaporation temperature and the previously calculated T_evap.

Cooling

As for the tests carried out in heating, here the parameters of interest are also calculated.

Total heat load is calculated as:

$$Total\ heat\ load = \frac{\left(Capacity + \left(P_{input} - 3P_{fan}\right)\right)}{2}[kW]$$

Refrigerant side pressure drop is calculated as above as the pressure difference between inlet and outlet from the exchangers.

In cooling, we prefer to calculate the average refrigerant saturation delta as the difference between average refrigerant outlet temperature and condensing temperature.

6.3.2.1 Data

The tests were carried out inside an ATEX certified room in stable conditions, and then the values are taken in stable conditions: the unit is turned on, once it reaches the steady-state condition, the tests of nominal duration of 30 minutes are made. After that, the test recording is turned off and then turned off the unit. Usually the tests, in order to be considered valid, are conducted for a sufficient time to ensure that the unit and the conditions of the test room are stable. The dates are updated instant by instant and several hundred values are returned for each probe. The values reported are the average of these data.

Among the various tests conducted, below are reported the parameters of interest from each test. The parameters reported are only those useful for comparison that will be performed in chapter 7.

HEATING	Capacity [kW]	P_{in} [kW]	P_{fan} [kW]	T_{evap} [°C]	SH [K]	Air flow rate [m^3/h]	Δp [kPa]
(20.25)	51.58	14.49	1.1336	2.51	3.23	34500	16.75
(23.72)	61.75	17.72	1.1336	1.91	3.68	34500	25.00
(28.36)	78.53	23.31	0.466	2.15	5.61	28000	14.60
(28.97)	77.33	21.55	0.7229	0.95	4.92	31400	33.00
(30.27)PH01	80.34	24.39	1.53	-2.50	2.63	36400	38.58
(31.08)	82.85	26.88	2.0656	1.32	5.33	38828	38

Table 26: Data collection from practical heating tests

COOLING	Capacity [kW]	P_{in} [kW]	P_{fan} [kW]	T_{cond} [°C]	SC [K]	Air flow rate [m^3/h]	Δp [kPa]
(22.70)	38.54	11.06	1.40	40.50	3	38000	3
(35.48)	59.41	19.95	1.4	42.63	5.15	38800	8
(45.64)	68.89	24.55	0.723	45.61	4.40	31400	10.69
(45.58)PC01	69.42	26.22	1.53	45.89	4.59	36400	10.85
(53.57)	85.22	24.85	0.98	46.26	5.28	33000	12

Table 27: Data collection from practical cooling tests

As we can see in this case the capacity values do not only have a target to reach, in fact, multiple operating points have been tested where they varied both the target power required and the speed and therefore the power absorbed by the fans.

When we consider the power of the fan and the air flow rate, we reason in terms of air flow and power per single fan, as there are three mounted on the unit. The capacity values are related to the heat exchanged of both heat exchangers that exchange heat with the air, while in the simulations reported in Chapter 5 and later in Chapter 7, they refer to values halved because in the software is studied a single heat exchanger and then doubled the values of power to obtain the final values.

6.3.2.2 Temperature tracking

Below are reported the temperature tracking tests of the heat exchangers in the cooling conditions at nominal load condition.

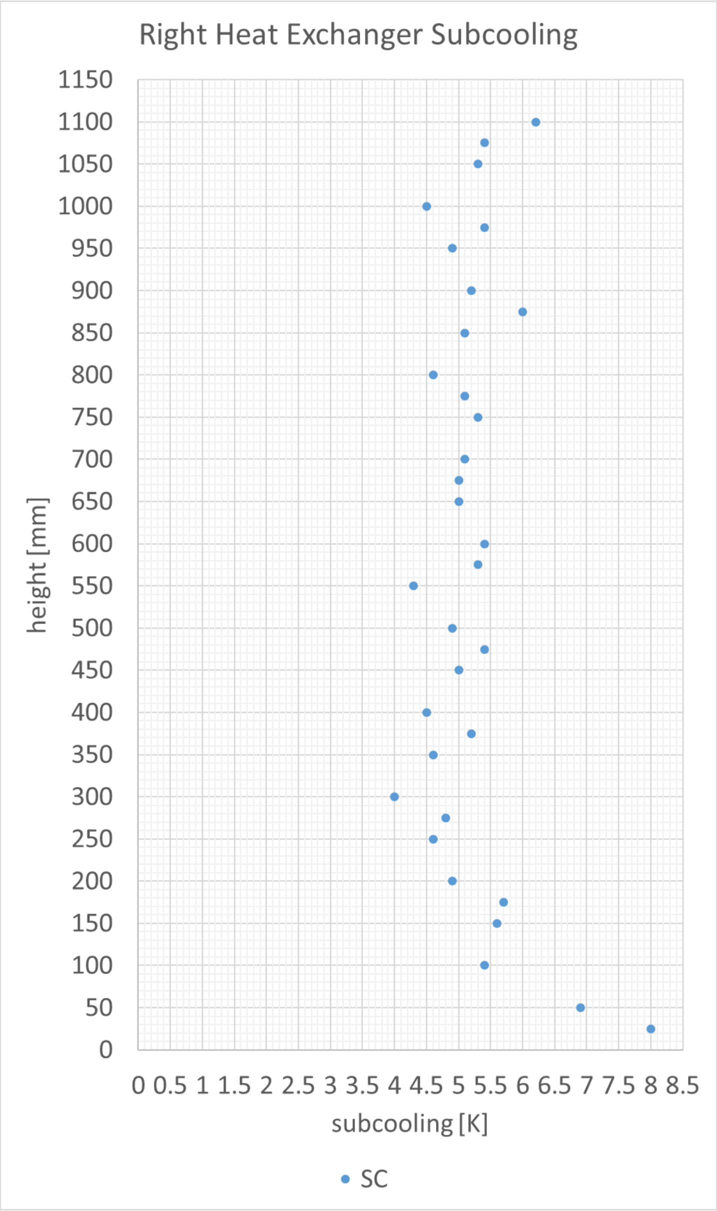


Figure 92: Right heat exchanger SC temperature distribution at full load

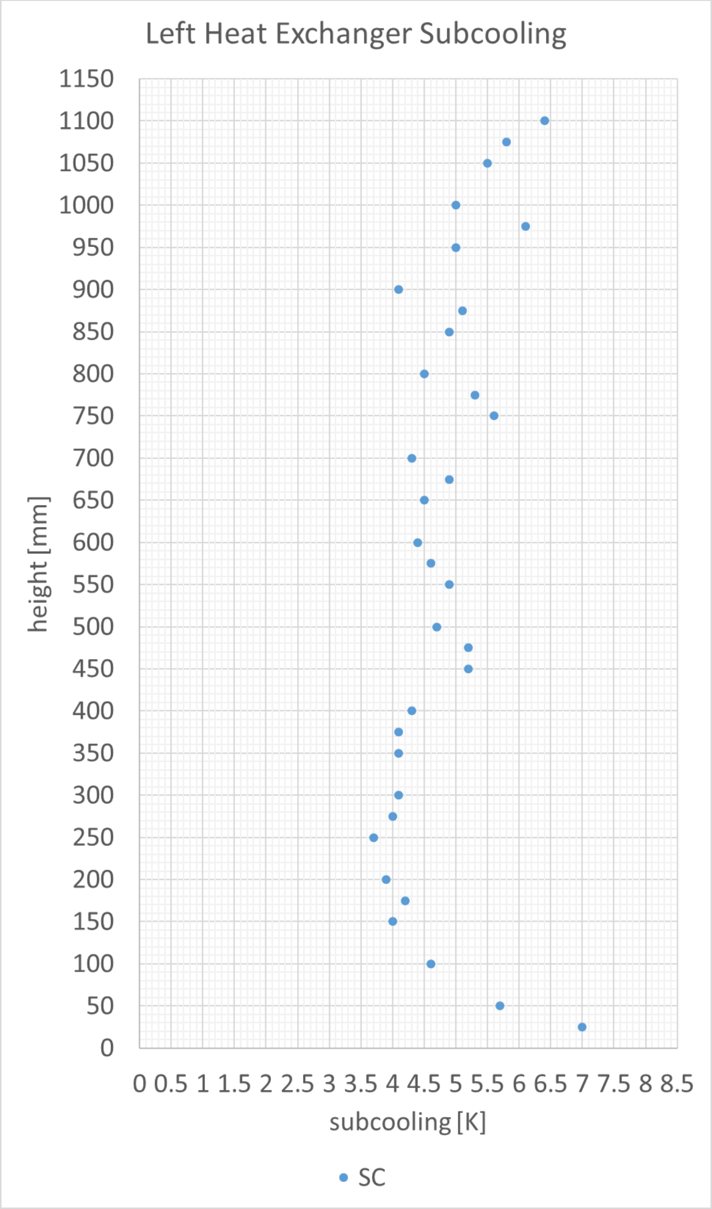


Figure 93: Left heat exchanger SC temperature distribution at full load

As we can see these two graphs are related to the condensation's operation and therefore in the conditions of cooling. Subcooling was plotted as a function of the height of the exchanger. As we can see the temperature trend of the exchangers is quite homogeneous, the exchanger on the right works slightly better in terms of uniformity of the values. The highest values in the lower part of the exchangers are perfectly normal. In fact, the last circuits at the bottom are composed of more tubes than the others (central circuits 3 tubes each, first circuit at the top and the two circuits in the lower part are composed of 5 tubes each). This promotes an increase in subcooling because a longer length of the circuit will increase the area of the heat exchange and therefore the heat exchanged by the circuit. On the other hand, the last circuits are reached by a lower amount of air flow, at speeds significantly lower than the one that reach the first circuits. This does not promote heat exchange. To reason in terms of heat exchange efficiency we can refer to the central circuits excluding both the first circuits placed in the upper part and those placed in the lower part. A homogenous distribution usually also involves a distribution of the refrigerant flows evenly divided between the various circuits and in general is defined a "good" heat exchanger, obviously the tracking of temperatures is not enough alone to determine if the exchanger is actually good, but at least it can be defined balanced and without problems in the distribution.

The same test was performed at partial load to see how the heat exchangers would behave when the exchanged power was lower.

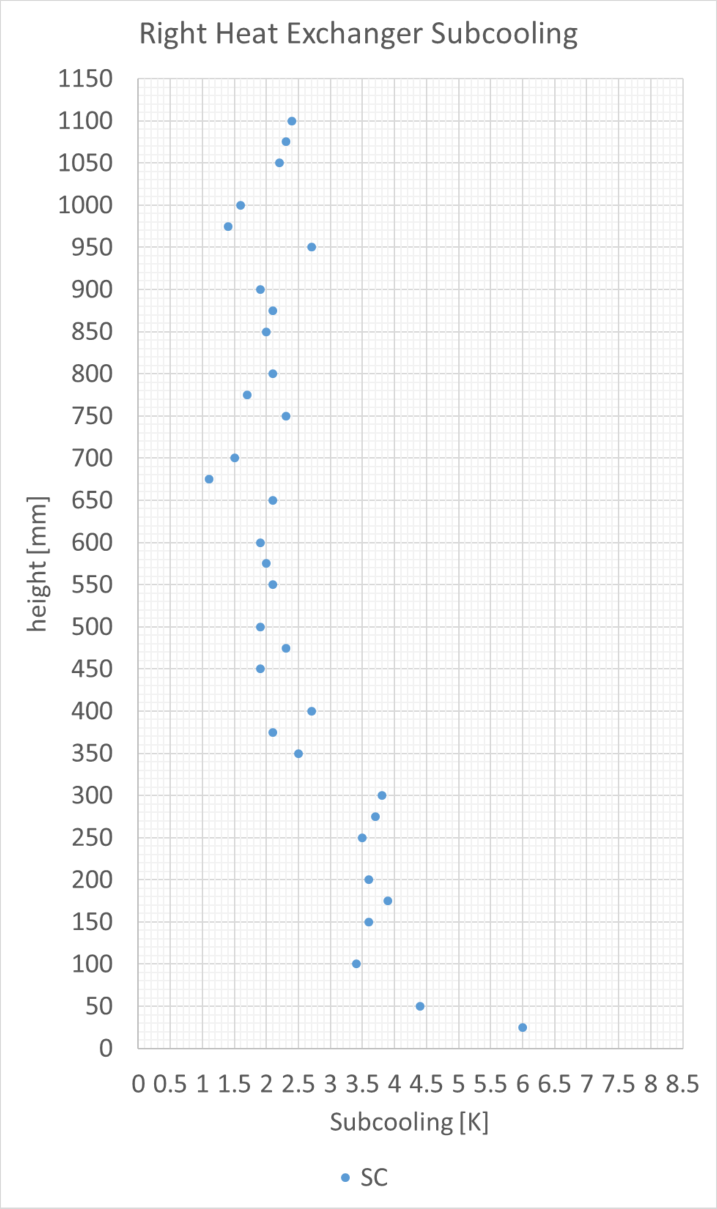


Figure 94: Right heat exchanger SC temperature distribution at partial load

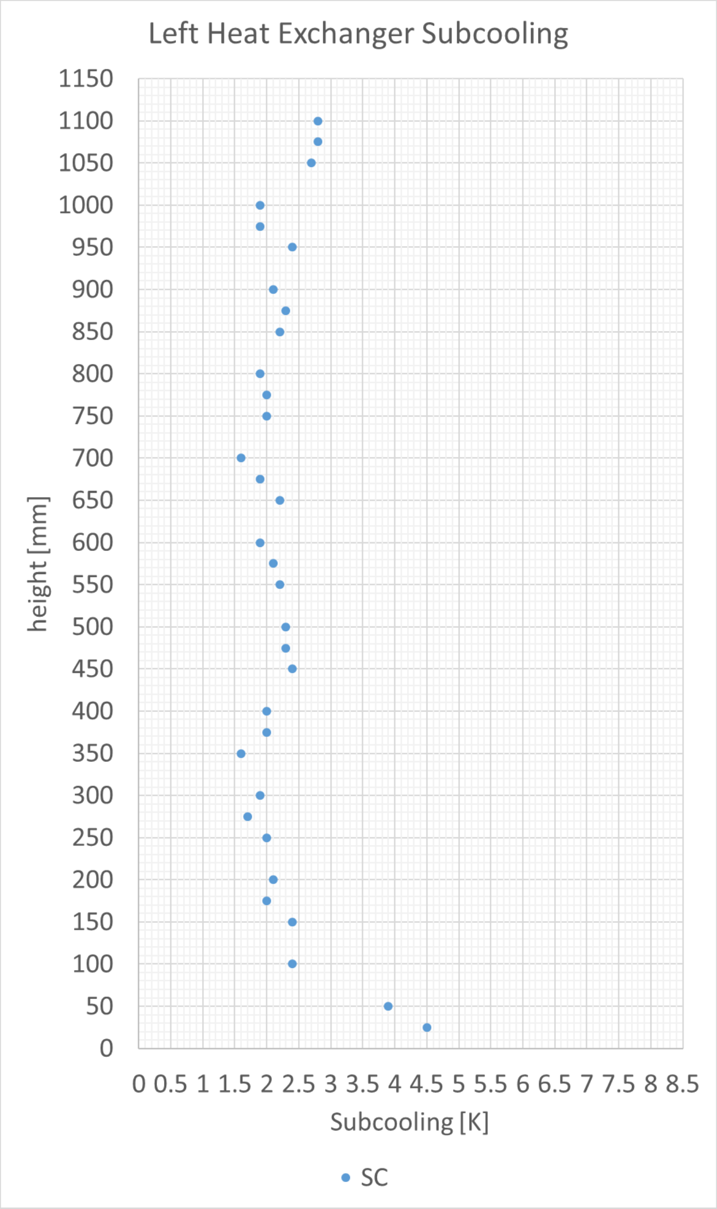


Figure 95: Left heat exchanger SC temperature distribution at partial load

At partial load, subcooling values are lower and more evenly distributed than in the full load case. We see that in the right heat exchanger the points are more scattered than the left, tracing the trend already seen at nominal load. In general, the exchangers have a good distribution of the subcooling points and therefore we can assume that the charge distribution is also uniform between the various circuits that make up the exchanger.

Chapter 7: Comparison of practical data and simulation via software

7.1 Introduction

After performing and reporting all the theoretical and practical tests in the previous chapters, let's now see the data comparisons for the two units: the VMC heat pump (A-A) and the Scroll heat pump (A-W). This procedure is used to understand how to optimize the fin and tube heat exchangers operating with propane starting from a theoretical sizing performed with CoilDesigner.

7.2 VMC heat pump (A-A)

As already seen in the previous chapter, in the practical tests to obtain the target power under the conditions of maximum charge legal limits of 150g inside the unit, we were forced to remove the capillaries to which the pressure probes were connected. The only valid tests for comparison are those where:

- the refrigerant charge does not exceed 150 grams
- target exchanged power is reached
- refrigerant pressure data are not known

We will therefore focus more on comparing temperatures.

Under these hypotheses, the practical tests will be compared with the values returned by the software simulating the conditions reported in the reports of the individual tests during operation in heating and cooling.

7.2.1 Heating operation

The heating tests previously seen in chapter 6, were simulated on Coildesigner. The boundary conditions are set by the control room and therefore constant:

- External air temperature = 7°C
- Relative air humidity = 87%
- Recovery air temperature= 20°C
- Recovery relative air humidity = 50%
- External air flow rate= 257-261 m^3/h
- Exhausted air flow rate = 273-275 m^3/h

As already seen in Chapter 6, the tests were conducted by increasing the opening of the EEV expansion valve. With these simulations, we go to search as precisely as possible two parameters: heat exchanged and SH. Focusing these two parameters on the software we can evaluate the evaporation temperature returned by the software compared to the real one.

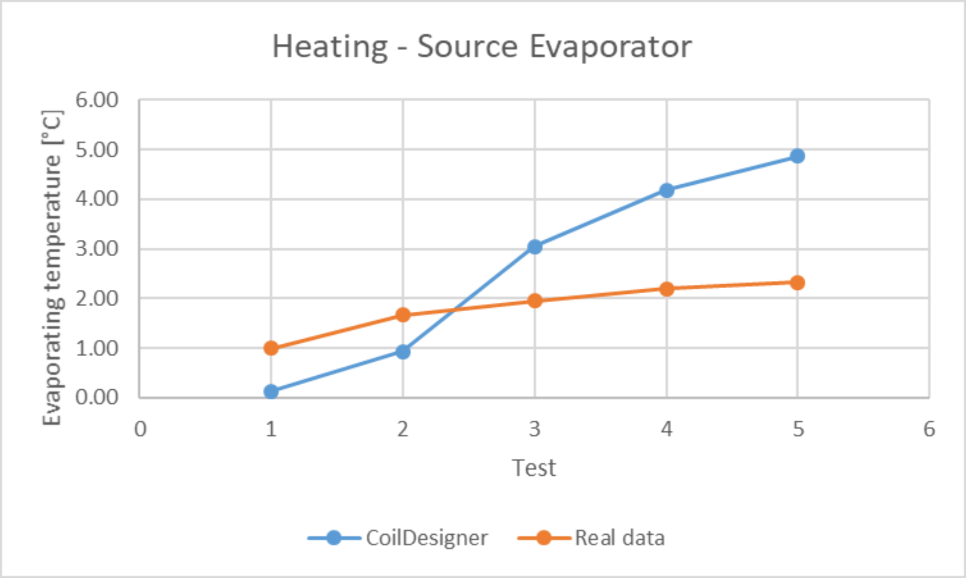


Figure 96: Comparison evaporating temperature between real data and CoilDesigner source H.E. of VMC heat pump heating operation

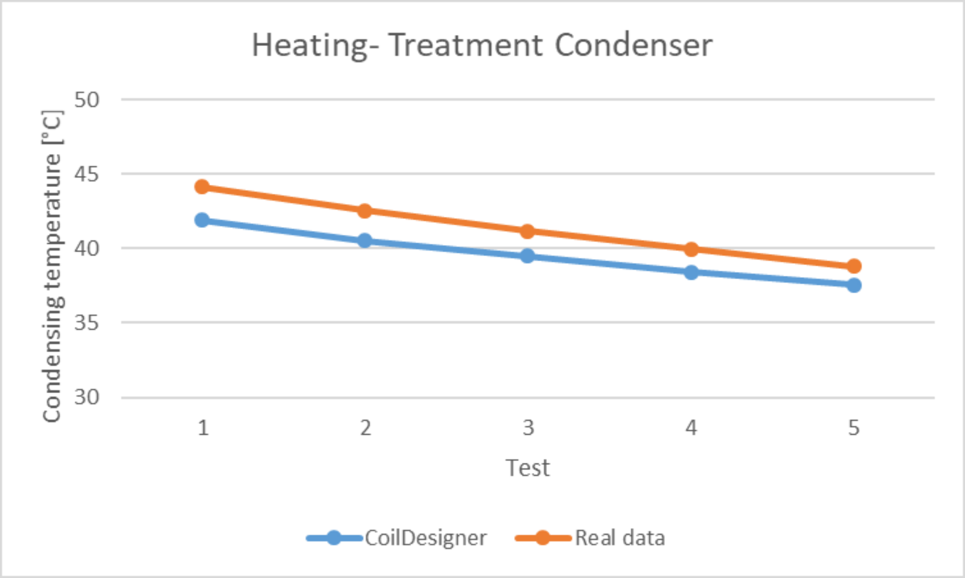


Figure 97: Comparison condensing temperature between real data and CoilDesigner treatment H.E. of VMC heat pump heating operation

In the above two graphs the evaporation and condensation temperatures measured during the tests were compared with the simulations.

The heat exchanger source in the real case has an almost constant trend, while the software initially underestimates the evaporation temperature and then go to overestimate the temperatures. In the first two tests CoilDesigner's behaviour is unusual. The causes of such irregularities will be analysed later. The behaviour from the 3 test onwards can be considered in line with what was expected. As the evaporation temperature increases, the software tends to increase the deviation from the actual data obtained during laboratory tests.

As for the treatment heat exchanger, you can notice that the software tracks the condensing temperatures slightly lower than the real case, with a slight deviation, but in a constant way. We can see that as the condensation temperature decreases, this difference becomes smaller and smaller.

7.2.2 Cooling operation

We follow the same reasoning performed for operation in heating, using the following parameters for the cooling operation:

- External air temperature = 35°C
- Relative air humidity = 47%
- Recovery air temperature= 27°C
- Recovery relative air humidity = 50%
- External air flow rate= 253 m^3/h
- Exhausted air flow rate = 331-332 m^3/h

The treatment heat exchanger will work as evaporator and the source heat exchanger will work as condenser. Unlike before, in the cooling conditions we go to center the exchanged power and the subcooling given by the practical tests and then evaluate the difference in condensation temperature between theoretical and real. For organizational reasons only two tests are reliable and usable for comparison with software.

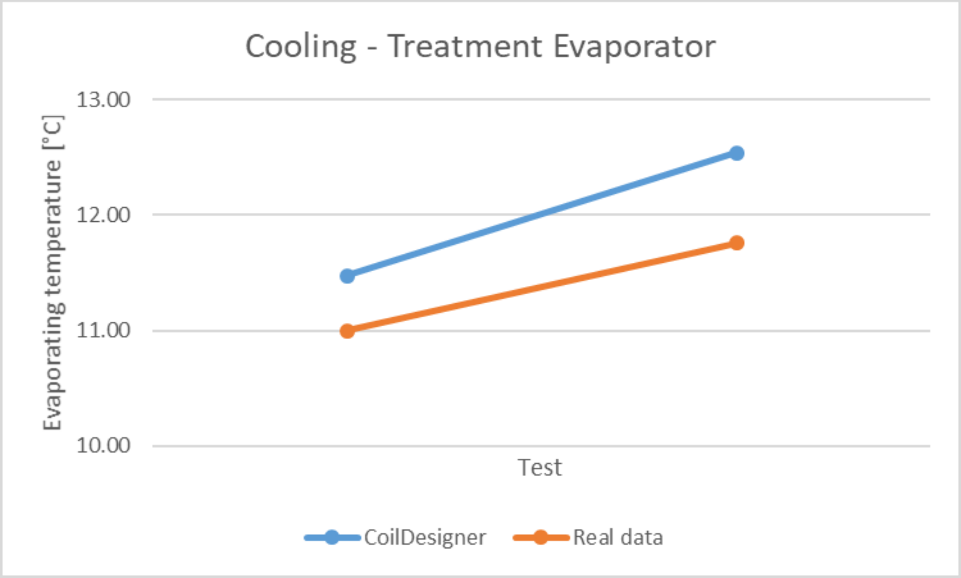


Figure 98: Comparison evaporating temperature between real data and CoilDesigner treatment H.E. of VMC heat pump cooling operation

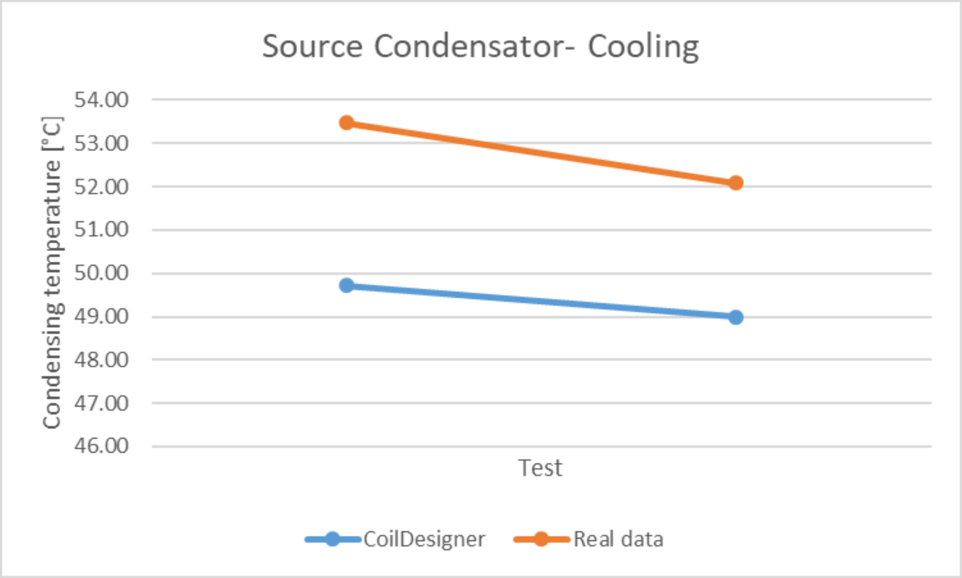


Figure 99: Comparison condensing temperature between real data and CoilDesigner source H.E. of VMC heat pump cooling operation

In cooling operation, as the evaporation temperature increases the deviation between CoilDesigner and real data increases slightly where the software tends to overestimate the evaporation temperature. It is difficult to predict a trend but in general, we can think that the deviation will tend to increase with increasing evaporation temperatures.

In the condenser, we see that the software underestimates the condensation temperature by various degrees. Also, in this case, it is not easy to predict the trend but in general we have seen in other cases that as the condensation temperature decreases, the difference is thinner.

7.2.3 Analysis of the results

From the comparison between experimental tests and simulations we can see that, using the parameters derived from the laboratory tests, CoilDesigner tends to overestimate the evaporation temperature and underestimate the condensation temperature. Then we assume a thermodynamic cycle like this:

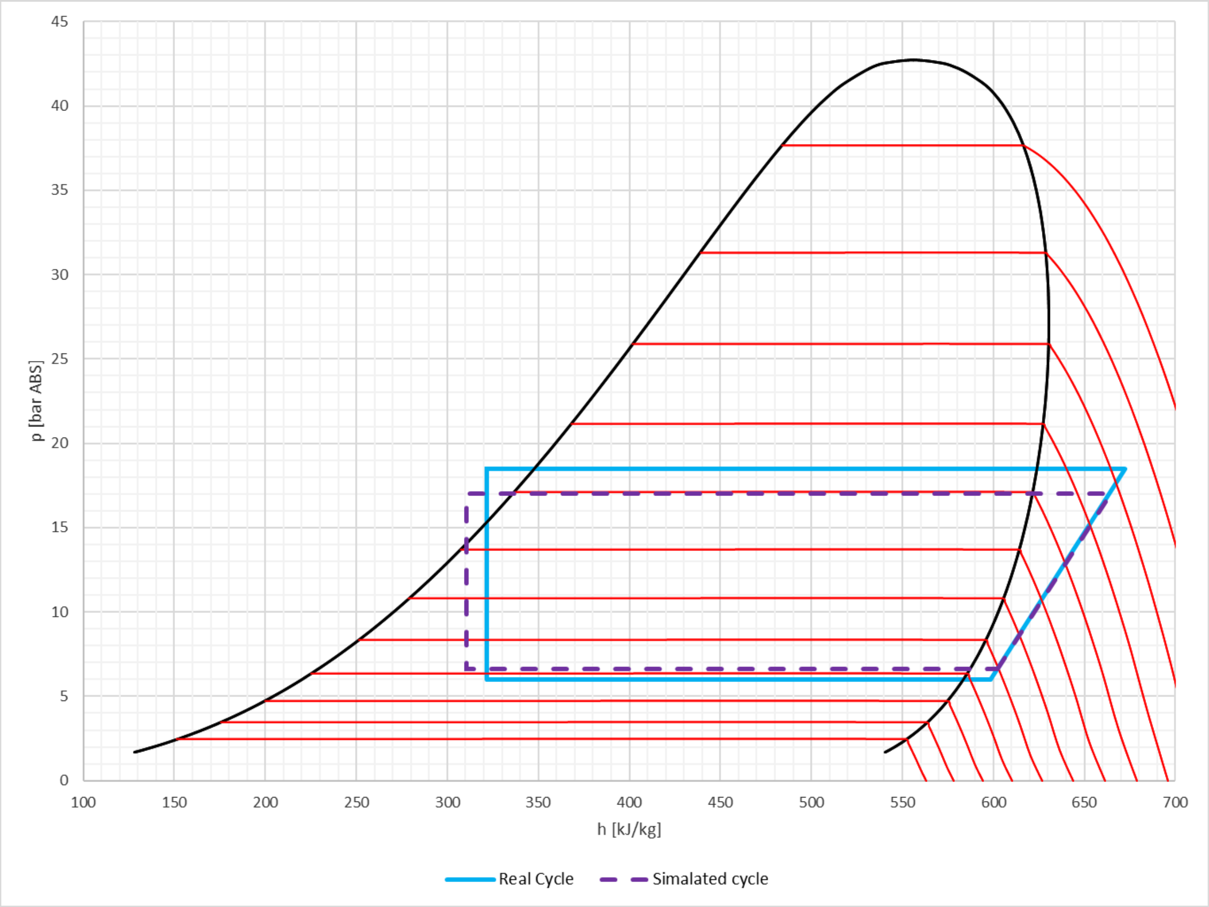


Figure 100: Comparison R290 real cycle and simulated

In the summer case, under the same conditions of power exchanged, superheating and subcooling, the program assumes a cycle with better efficiency with higher evaporation temperature and lower condensation temperature. If we apply the first principle of thermodynamics:

$$COP_{cooling} = EER = \frac{Q_1}{L} = \frac{T_1}{T_2 - T_1}$$

Where:

- Q_1 is the heat exchanged in the evaporator

- L is compressor work equal to the difference between heat exchanged at the condenser and heat exchanged at the evaporator
- T_1 is the evaporating temperature
- T_2 is the condensing temperature

We see that, if the evaporation temperature rises and the condensation temperature drops, the EER numerator increases, while the denominator drops, causing the overall efficiency of the heat pump in cooling operation to increase. Similarly, the same reasoning can be applied to the case of heating condition. Consequently, we can say that Coildesigner in general is very accurate in evaluating the characteristics of the exchangers, but tends to slightly overestimate their performance. Therefore, in the theoretical cycle, the compressor requires less power than the real.

It is acceptable to think that the software has a certain deviation from the real data because the analysis of CoilDesigner stops at the boundary of the exchanger without considering: the potential losses inside the various tubes of the circuit, the efficiency of the other exchanger, expansion valve, the type and the efficiency of the compressor and any temperature and humidity fluctuations that occurred during the tests.

7.3 Scroll air to water heat pump (A-W)

The tests related to the Scroll heat pump were performed under heating and cooling conditions in different operating conditions to test as much as possible the operating points of the unit. Unlike the VMC heat pump, we have no refrigerant charge limits and this allowed us to use pressure probes to track the pressure difference of the refrigerant in and out of the heat exchangers. The data derive from practical tests were simulated in CoilDesigner using so the real cycle conditions and evaluating, for the same power and SC or SH, the pressure drop on refrigerant side of the exchangers. As we have seen in previous chapters, the pressure drop of the refrigerant in large heat exchangers is important from many reasons and therefore obtaining from the parameters comparable between theoretical and practical allows us to evaluate the exchanger in its entirety.

7.3.1 Heating operation

During operation in heating fin and tube heat exchangers, which are two arranged in "V", operate as evaporators. The conditions used in the software are:

- External air temperature = 7 °C
- External air relative humidity = 87%
- Air flow rate= 30000-38000 m^3/h for each heat exchanger

The values used and shown in the graphs below are referred to the single heat exchanger.

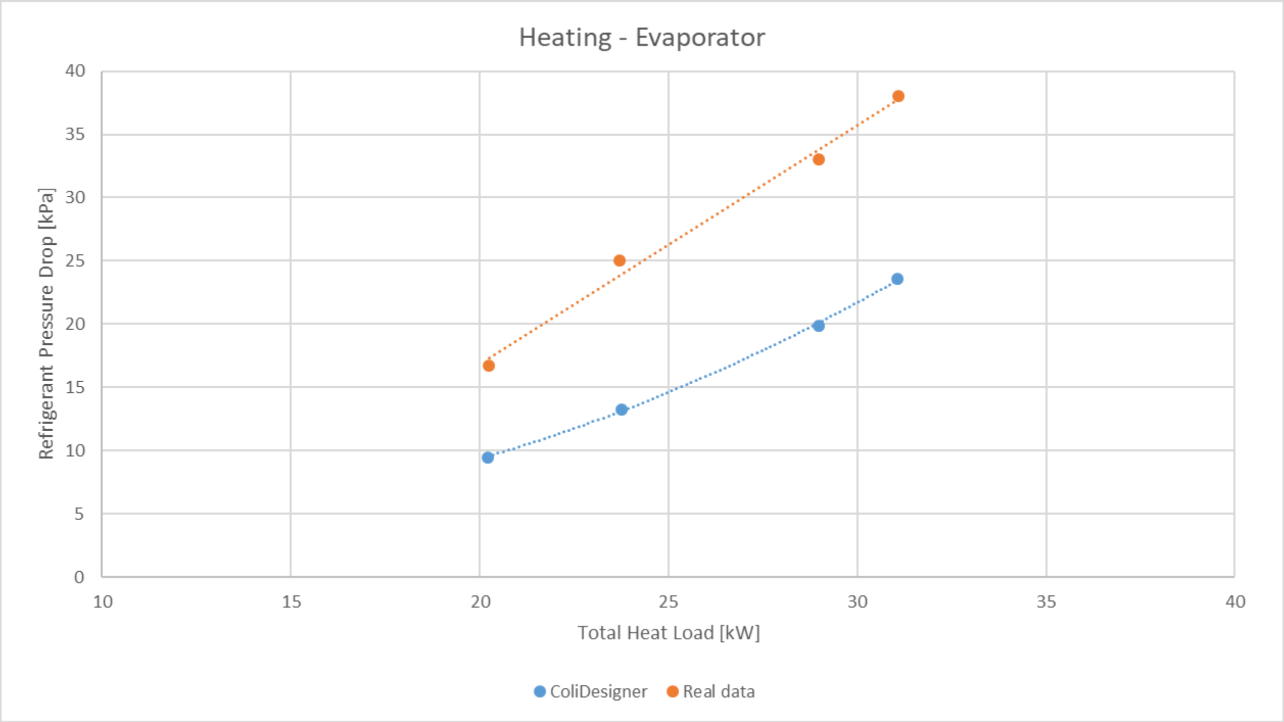


Figure 101: Refrigerant pressure drop - Total heat load graph for evaporating H.E. scroll unit

Plotted values in the graph are the pressure drop refrigerant side values related to the relative heat load. In orange are indicated the values obtained from the practical tests and in blue the values returned by the software. We see that CoilDesigner returns significantly lower pressure drop values than real cases, the difference between real and simulated varies in a range 7-14kPa that increases slightly with increasing total heat loads.

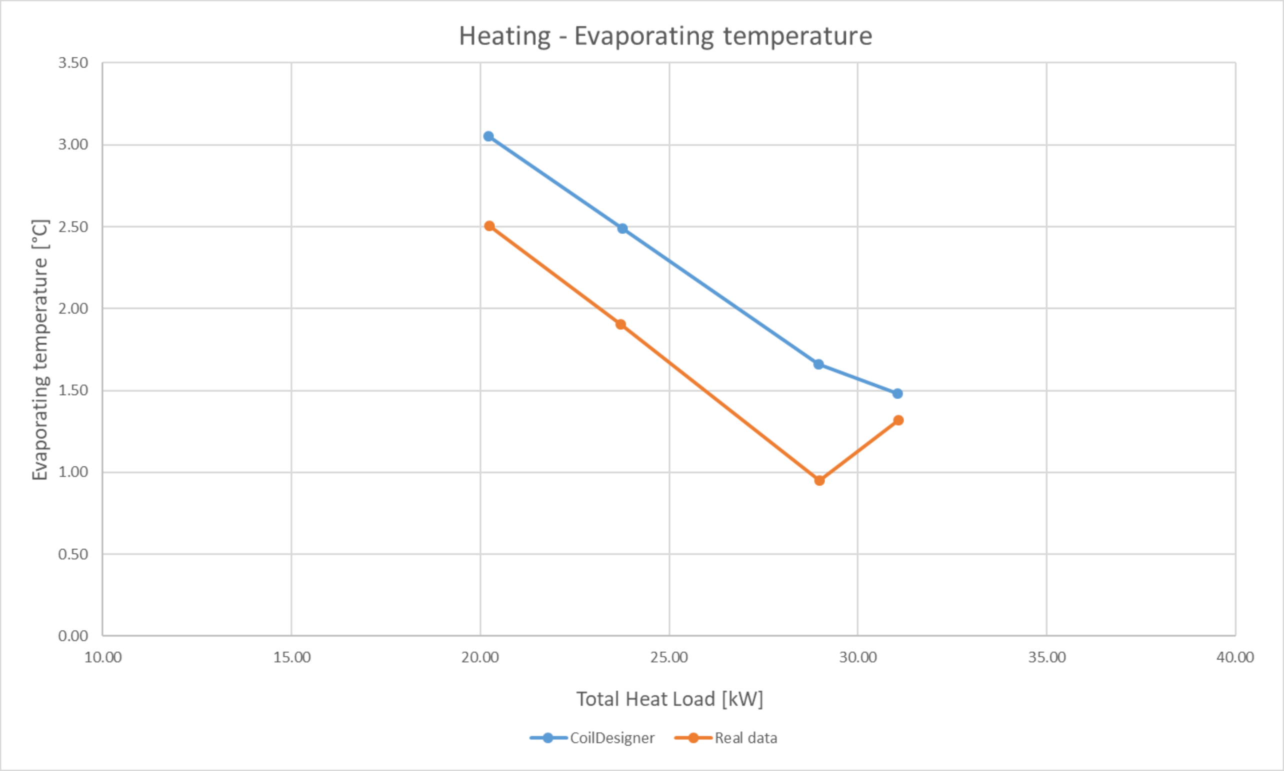


Figure 102: : Evaporating temperature - Total heat load graph for evaporating H.E. scroll unit

The graph refers to the variation of evaporation temperatures between CoilDesigner and practical tests in each test. In general, the difference remains in a range between 0.16 and 0.71 °C, where the software slightly overestimates the evaporation temperature in the exchanger. In the condition more similar to the nominal one, which is to the one with a heat load of 31.08kW for evaporator, we see as the deviation of the temperatures assumes the minimum deviation of 0.16 °C.

7.3.2 Cooling operation

In cooling conditions the exchangers have the function of capacitors. The conditions resulting from the practical tests used in the software are:

- External air temperature = 35 °C
- External air relative humidity = 49%
- Air flow rate= 31400-38828 m^3/h for each heat exchanger

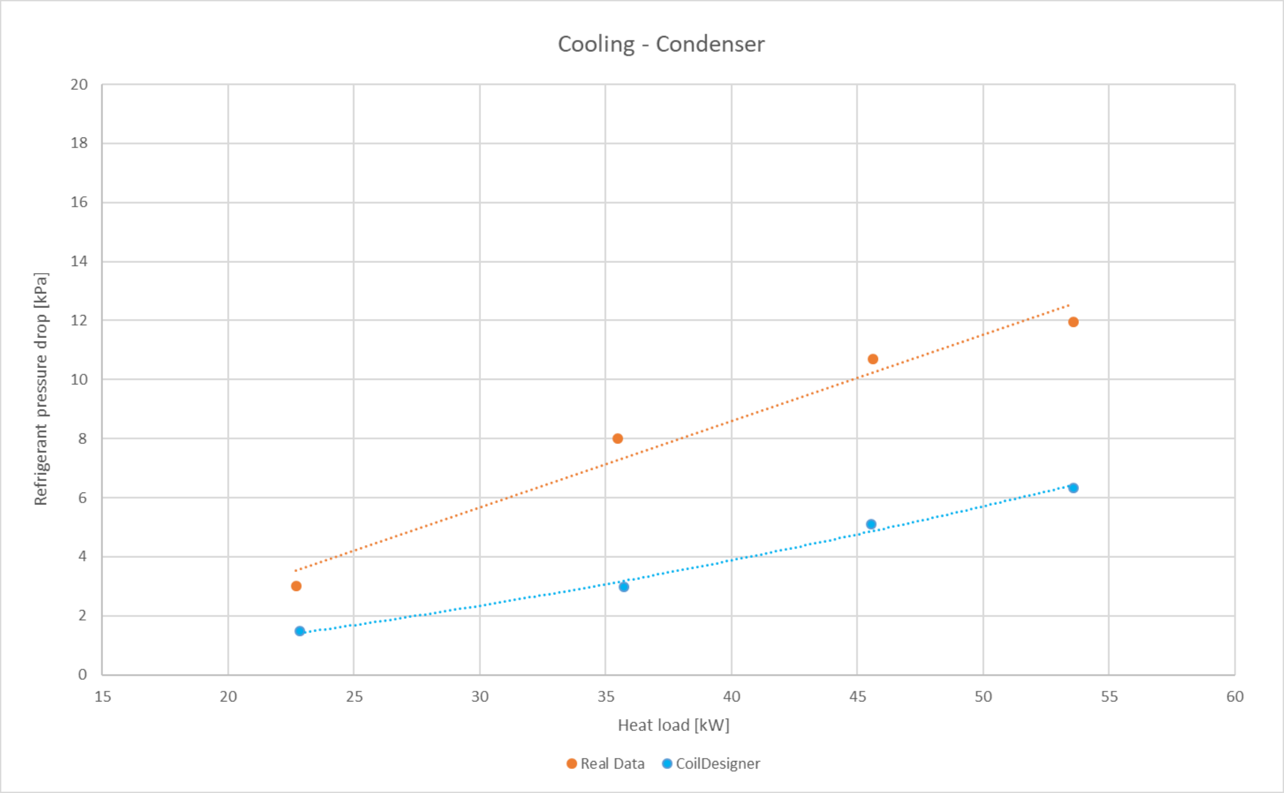


Figure 103: : Refrigerant pressure drop - Total heat load graph for condensing H.E. scroll unit

From the graph, we see how the difference between the real and calculated pressures tends to increase slightly as power increases in a range of 1.5-5.6 kPa, values significantly lower than in heating operation. The two curves have very similar trends, the software understates pressure losses. As we have already seen, the pressure drop values of the refrigerant in operation by condensers are much lower than the evaporator operation, therefore the deviation of CoilDesigner is greater in the case of condensers than in the case of evaporators.

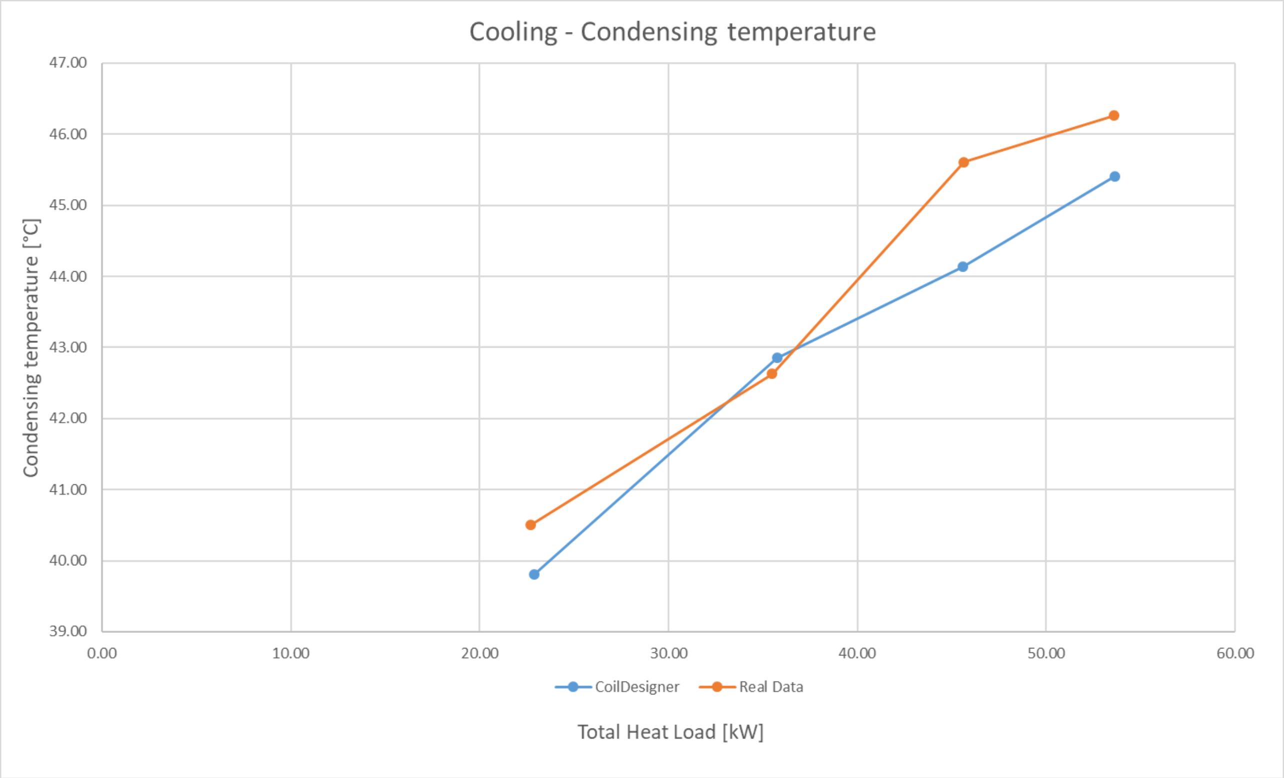


Figure 104: Cooling temperature - Total heat load graph for condensing H.E. scroll unit

We see that in general the software can find values quite close to reality.

7.3.3 Analysis of the results

The data obtained from the comparison between practical data and simulations for the scroll heat pump confirmed what we have already seen with the VMC heat pump for the evaporation temperature and for the condensation temperature. In general, the software overestimates the evaporation temperature and underestimates the condensation temperature. So, for the same heat exchanger, SH and SC, consider the heat exchanger better than it actually is in terms of performance. A further confirmation comes from the comparison on the pressure drop refrigerant side that is considered less than the reality, thus making the exchanger more ideal and so with less losses. We see how in general the distance between the pressure and temperature curves follows a mostly constant trend, a symptom of good linearity both by the program and by the software. The only point that would seem to deviate from these considerations is the power of 35.48 kW where there is an intersection between the condensing temperature curves. The causes can be multiple, for example a bad combination of the correlation factor related to the condensation phase used in the software or even an error in the instrument, or a punctual reading of the measurements during an unexpected transient regime during the recording tests.

In general, we can say that, the software is able to represent, with acceptable and constant errors, the real functioning of the heat exchangers regardless of their size, number of tubes, exchanged power. Using this software to size tube&fin heat exchangers operating at R290 is feasible, it must be taken into account that the software will return values with errors:

- A small overestimation of the evaporation temperature
- A small underestimation of condensation temperature
- An underestimation of refrigerant pressure drops

From a thermodynamic point of view, an underestimation of pressure losses means considering the exchanger better than it really is. From the mechanical point of view, however, if during the sizing you get pressure drops that ensure the minimum refrigerant speeds necessary for the return of the oil, in reality then the return of the oil will be ensured having higher losses and therefore higher refrigerant speeds.

It would have been interesting to compare the refrigerant side pressure losses between the two units, but for the reasons explained in Chapter 6 and subsequently also at the beginning of Chapter 7, this was not possible.

7.4 Modification of correction factor for correlation coefficient

As seen in chapter 7, in the graphs we notice a substantial difference between the values we get from the laboratory tests and the simulations on CoilDesigner. The goal would be to make the software return the values as similar as possible to the real values. As seen in Chapter 5 during the explanation of the fin and tube heat exchanger simulations procedure, there are three correction factors used to "change" the heat exchange and three to adapt the pressure drop.

Correlation coefficients are various and are based on research studies during the decades. Correlations were introduced in the calculations of the heat transfer coefficients when it was understood that from the theoretical calculation to the practical case there was a large deviation in the results. Over the years, many researchers have published their correlations classes of fluids, conditions of motion and geometric configurations.

To modify these coefficients would mean modify the work of researchers who have created these correlations by performing multitudes of tests and studies. However, the program allows us to insert in addition to the correlation factor also a correction factor that allows us to use pre-existing coefficients adapting them to our needs.

The idea is then to simulate the tests and change the correction factor, which is normally set to one, until you get the same values as the practical tests. Initially we focus on the tests performed on the heat exchanger used on the Scroll heat pump (A-W) during the operation in heating of the unit because we have the most complete values both for what concerns the pressure data and for those relating to evaporation temperatures.

7.4.1 Heating condition Scroll heat pump (A-W)

Choosing the most representative test, we increase the correction factor related to the refrigerant two-phase pressure drop correlation. Each increase corresponds to a simulation performed. After hitting the pressure drop factor that allows us to obtain results similar to the values obtained from laboratory tests, we focus on the heat transfer coefficient factor. In this case, we have to go below the unit value until we get an evaporation temperature as close as possible to the real one.

Below are reported the value obtained during the test in order to perform the best matching of the correlation factor.

Correction factor two phase pressure drop correlation	1	1.25	1.75	1.75	1.75
Correction factor two phase heat transfer correlation	1	1	1	0.9	0.6
Refrigerant pressure drop from practical data [kPa]	25	25	25	25	25
Refrigerant pressure drop after modification [kPa]	13.70	16.38	21.65	21.77	22.32
Evaporating temperature from practical data[°C]	1.51	1.51	1.51	1.51	1.51
Evaporating temperature after modification [°C]	2.62	2.52	2.34	2.25	1.88

Table 28: Variation of evaporating temperature and pressure drop varying the correction factors

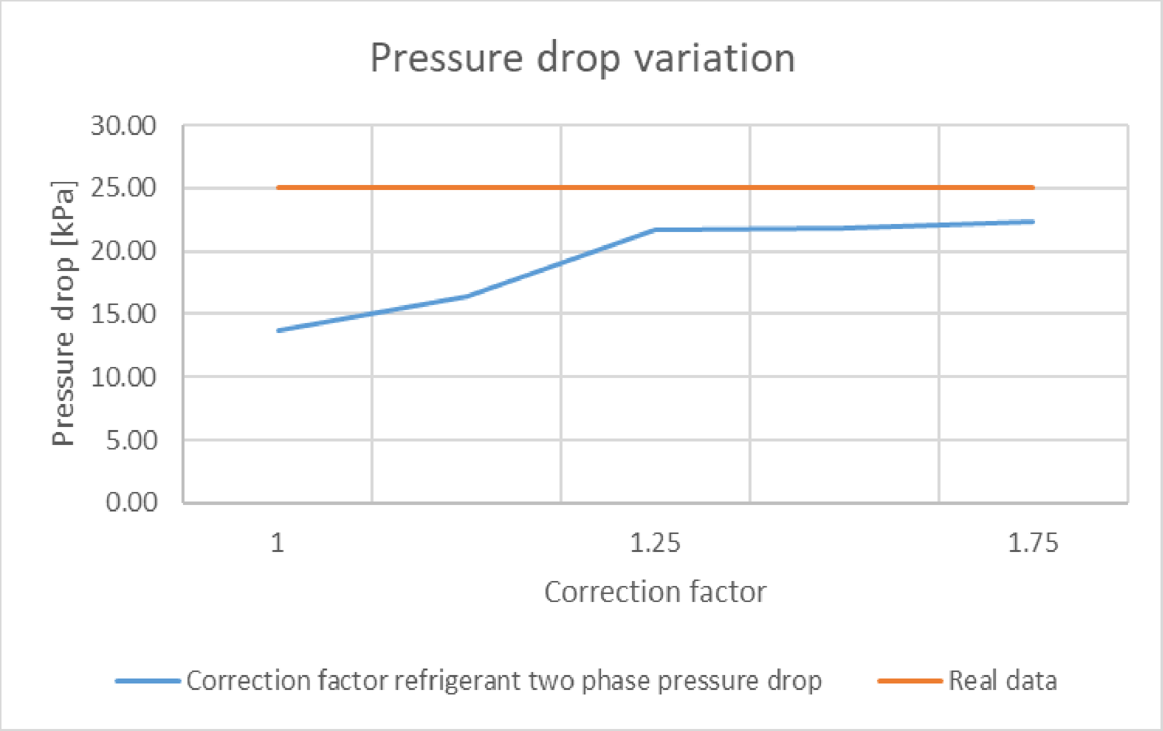


Figure 105: Pressure drop refrigerant side variation varying the correction factor

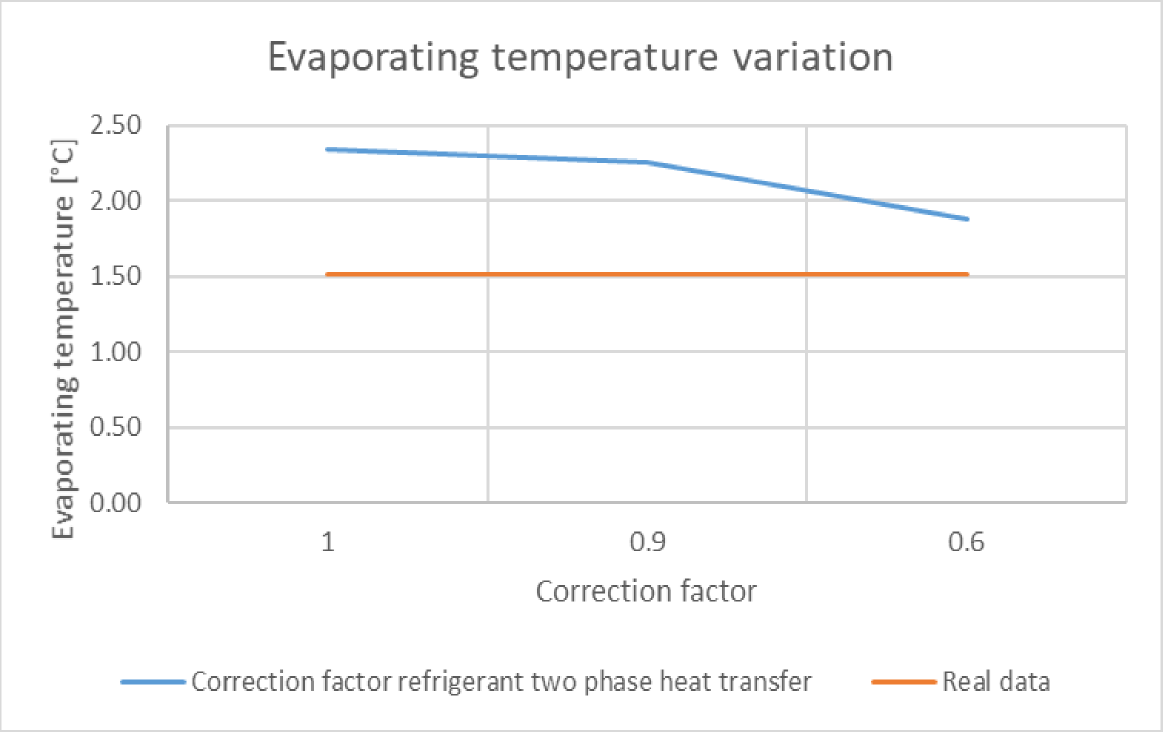


Figure 106: Evaporating temperature variation varying the correction factor

As we can see the difference on the pressure drop decreases significantly as the correction factor on the pressure drop increases, while the difference on the evaporation temperature decreases as the correction factor on the heat transfer coefficient decreases.

After finding these corrections for the most representative test, we apply the new correction factor to all the other tests to see if the behaviour remains mostly linear.

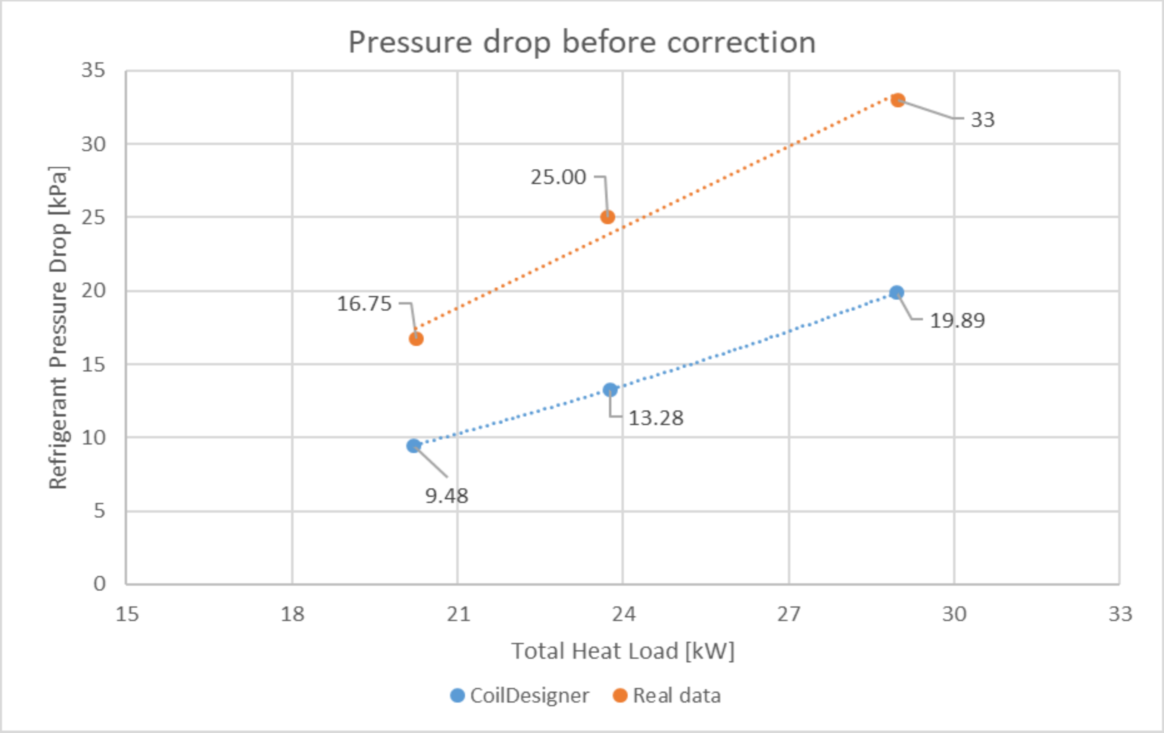


Figure 107: Comparison between pressure drops simulated without modification and real data

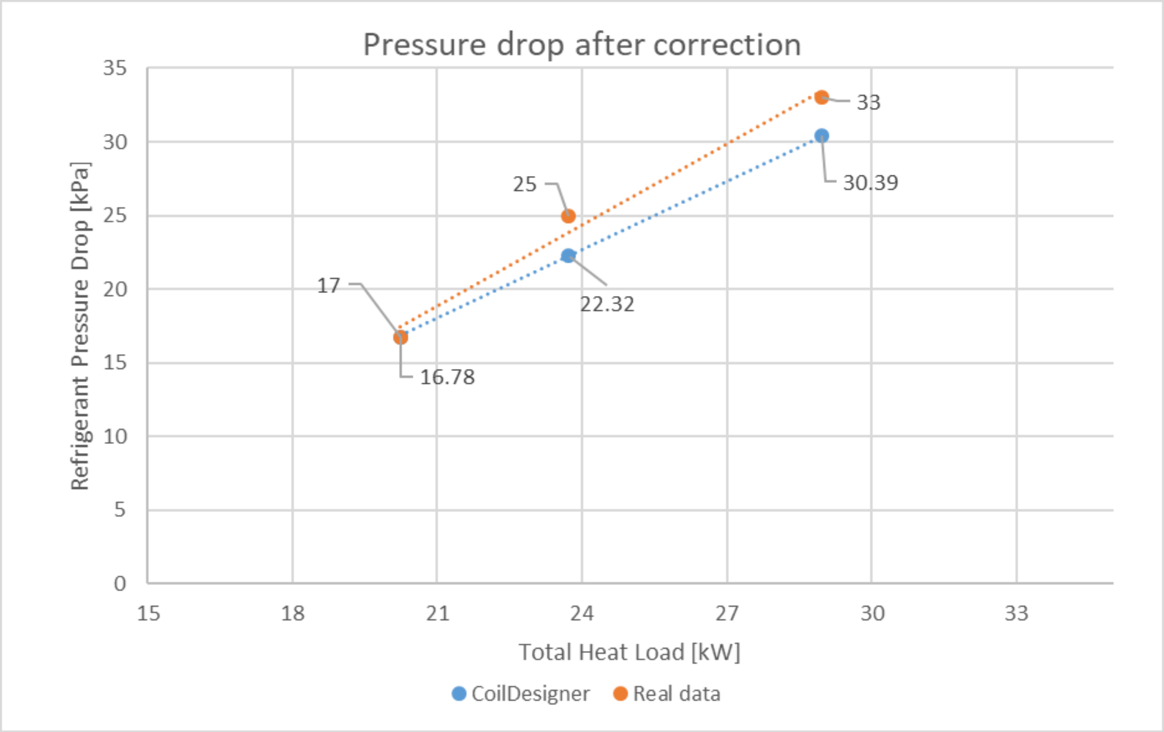


Figure 108: Comparison between pressure drops simulated with modification and real data

As we can see, the corrective factors chosen lead to a good matching of the pressure drop refrigerant side. In the three tests, we see a minimum deviation of 0.22 kPa and a maximum of 2.61 kPa. These values are in an acceptable range below 8% of the total value against the previous 40%.

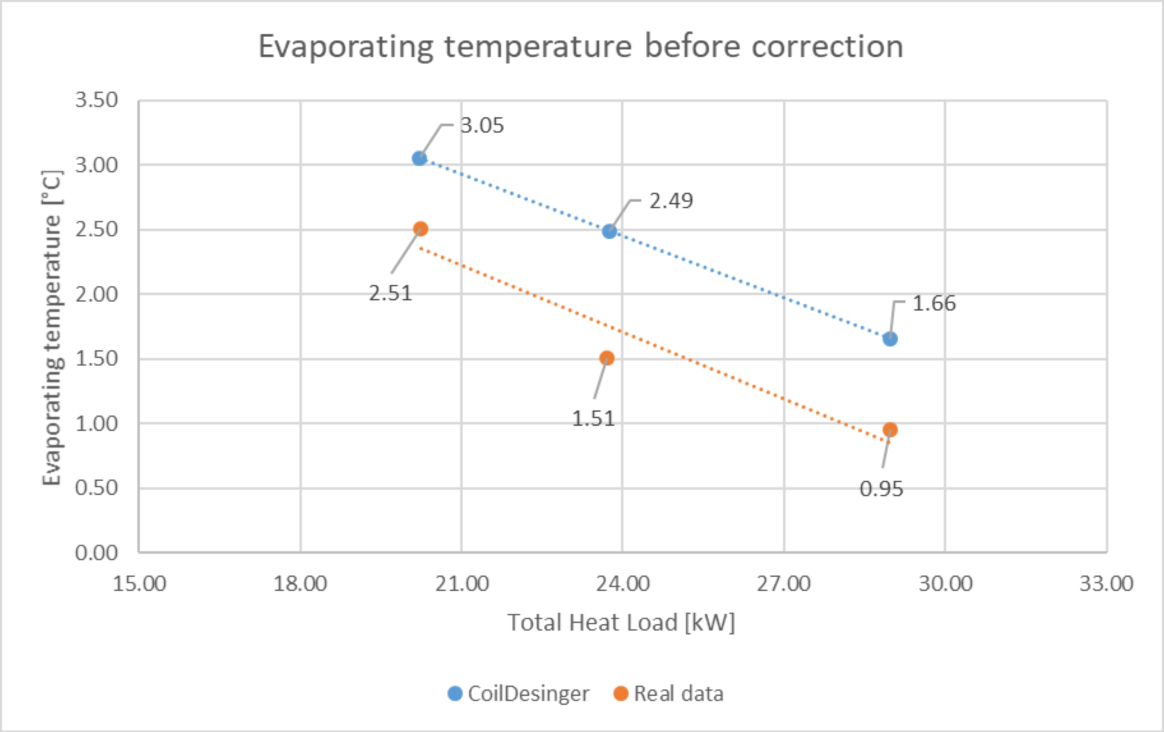


Figure 109: Comparison between evaporating temperature simulated without modification and real data

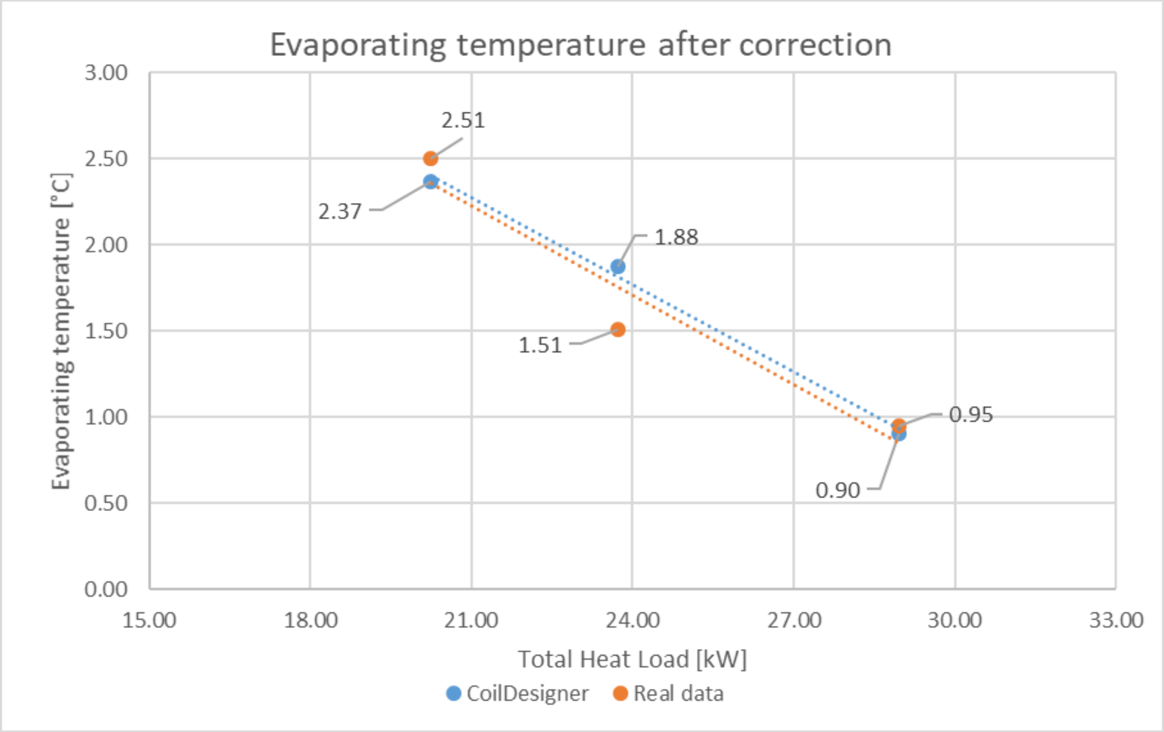


Figure 110: Comparison between evaporating temperature simulated with modification and real data

Before the correction of the coefficients, there was an error in the results of the evaporation temperature between 17.7% and 42.8%. After correction, we reduced the error in a range between 5.2% and 19.5%.

As we can see, the choice of correction factor for the single test is valid also for the other tests, maintaining a small detachment from the real values, but completely acceptable.

7.4.2 Cooling condition Scroll heat pump (A-W)

Now we move to the modification of the correction factor two phase pressure drop correlation and correction factor two phase heat transfer correlation for the tests in cooling condition.

As before, we take a single laboratory test, define as sample test, and initially change the corrective factor related to the pressure drop to get from CoilDesigner similar results to the data of the practical test, then we also try to get closer to the condensation temperature by changing the correction factor related to heat transfer correlation.

Refrigerant pressure drop from practical data [kPa]	11	11	11	11	11	11	11	11
Condensing temperature from practical data[°C]	46.89	46.89	46.89	46.89	46.89	46.89	46.89	46.89
Correction factor two phase pressure drop correlation	1	1.25	1.75	2	2	2	2	2
Correction factor two phase heat transfer correlation	1	1	1	1	0.6	0.4	0.25	0.22
Refrigerant pressure drop after modification [kPa]	5.21	6.12	7.89	8.77	9.19	9.54	10.03	10.12
Condensing temperature after modification [°C]	44.13	44.07	44.07	44.08	44.27	44.75	45.93	46.49

Table 29: Variation of condensing temperature and pressure drop varying the correction factors

From the data shown in the table above, we can see that increasing the corrective factor of the pressure drop and decreasing the corrective factor related to the heat exchange, the value of the pressure drop calculated by software is getting closer and closer to the value that we get from the practical test.

The decrease in the value of the correction factor two phase heat transfer correlation results in an increase in the condensation temperature.

The graphs below are related to all the simulations performed to obtain the results sought.

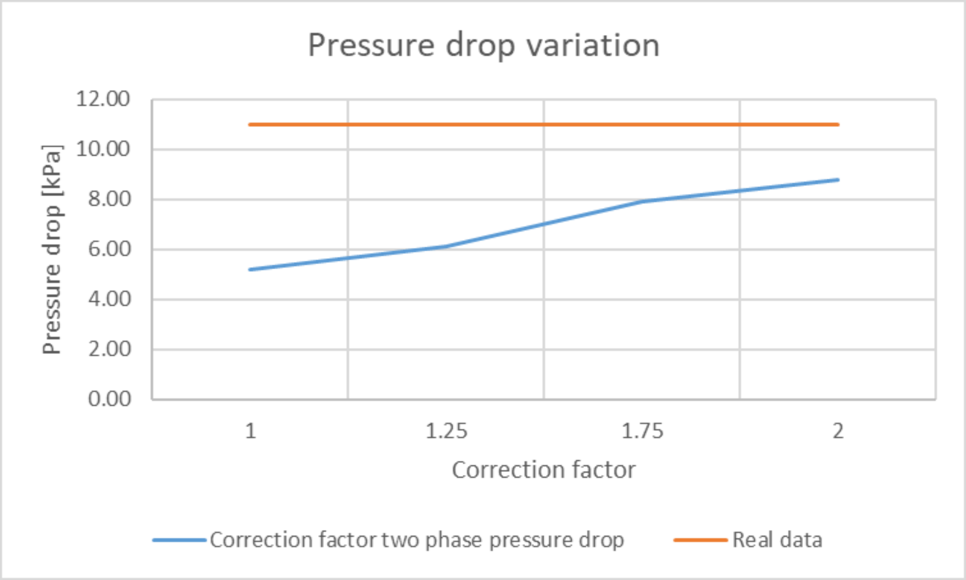


Figure 111: Pressure drop refrigerant side variation varying the correction factor

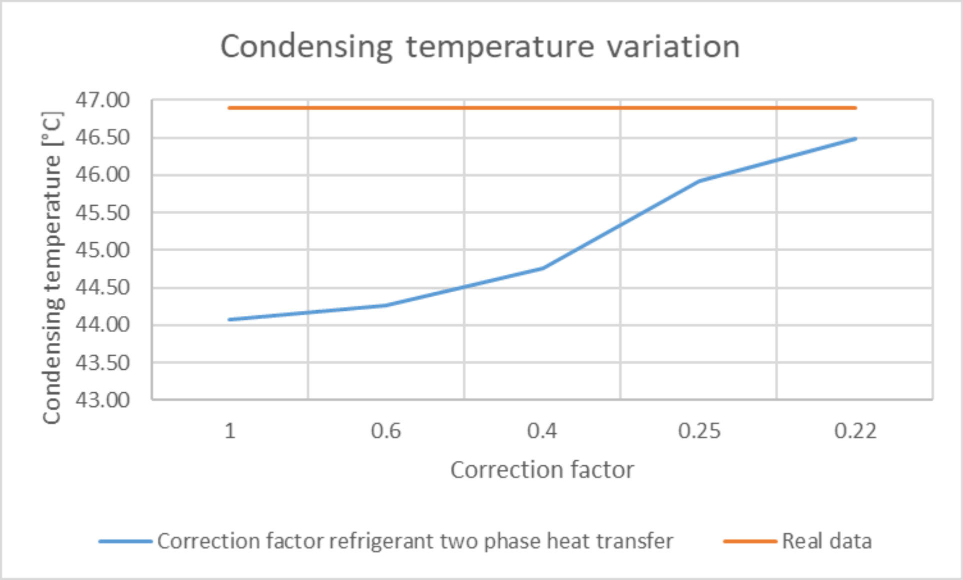


Figure 112: Condensing temperature variation varying the correction factor

Obtained the desired results on the individual sample test, we will apply the corrections to all the tests.

Below are graphs of pressure drops and condensation temperatures in the various tests before and after the introduction of corrective factors.

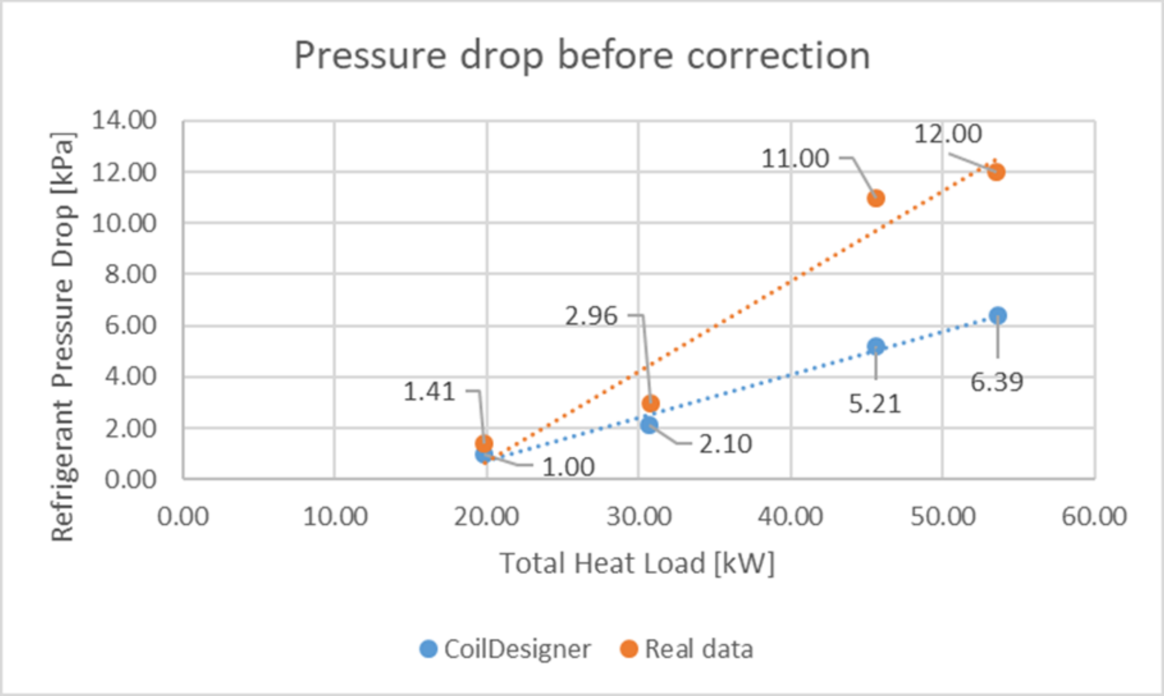


Figure 113: Comparison between pressure drops simulated without modification and real data

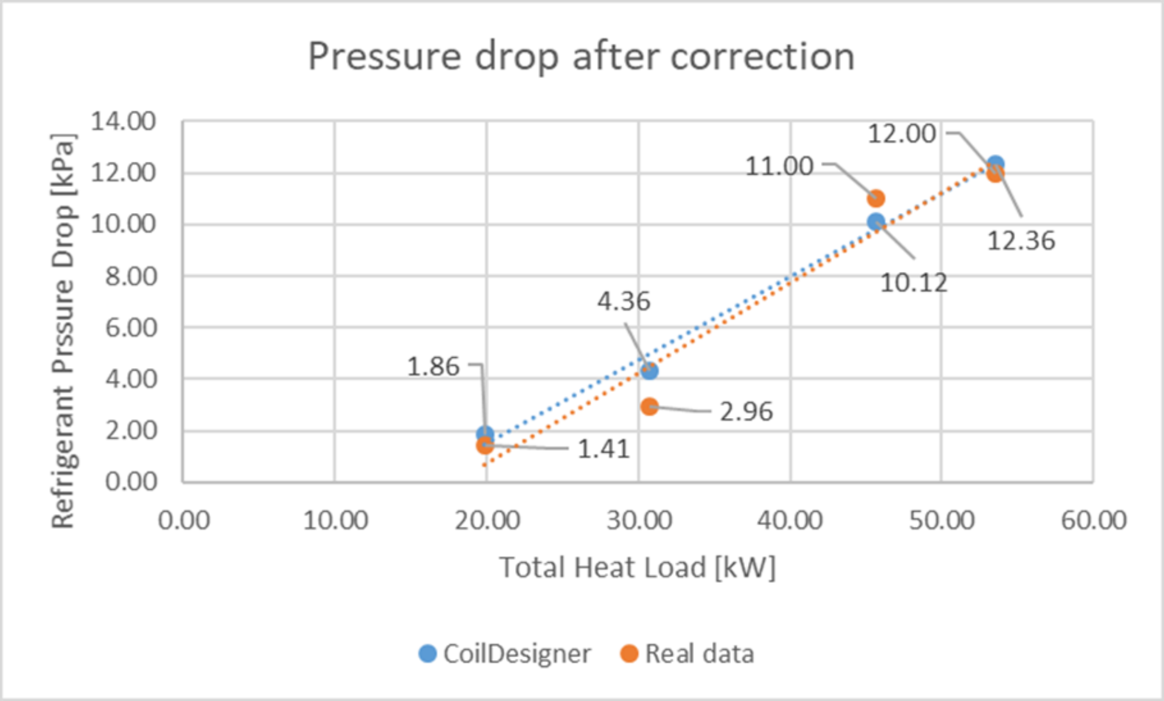


Figure 114: Comparison between pressure drops simulated with modification and real data

The graphs show the difference of the pressure drop before and after the introduction of the correction factors. From the first graph, already analysed in the previous chapter, to the increase of the heat load also increases the difference between the real values and those calculated by the software up to a maximum of 52.64% of error that corresponds to 5.8 kPa. The error that is obtained after correction is at most 32.56% in the second test, but in the remaining four tests the maximum is 24% which corresponds to 0.45 kPa. As the heat load increases, the values are matched more and more until an error of 8% equal to 0.36kPa.

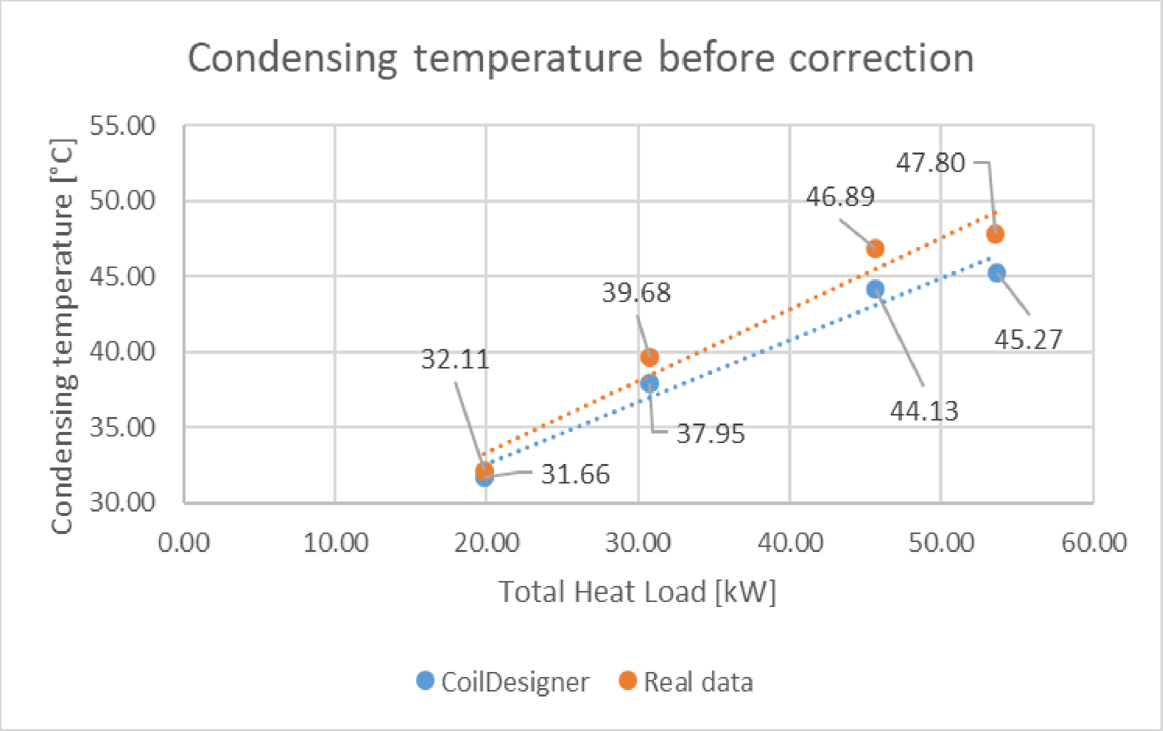


Figure 115: Comparison between condensing temperature simulated without modification and real data

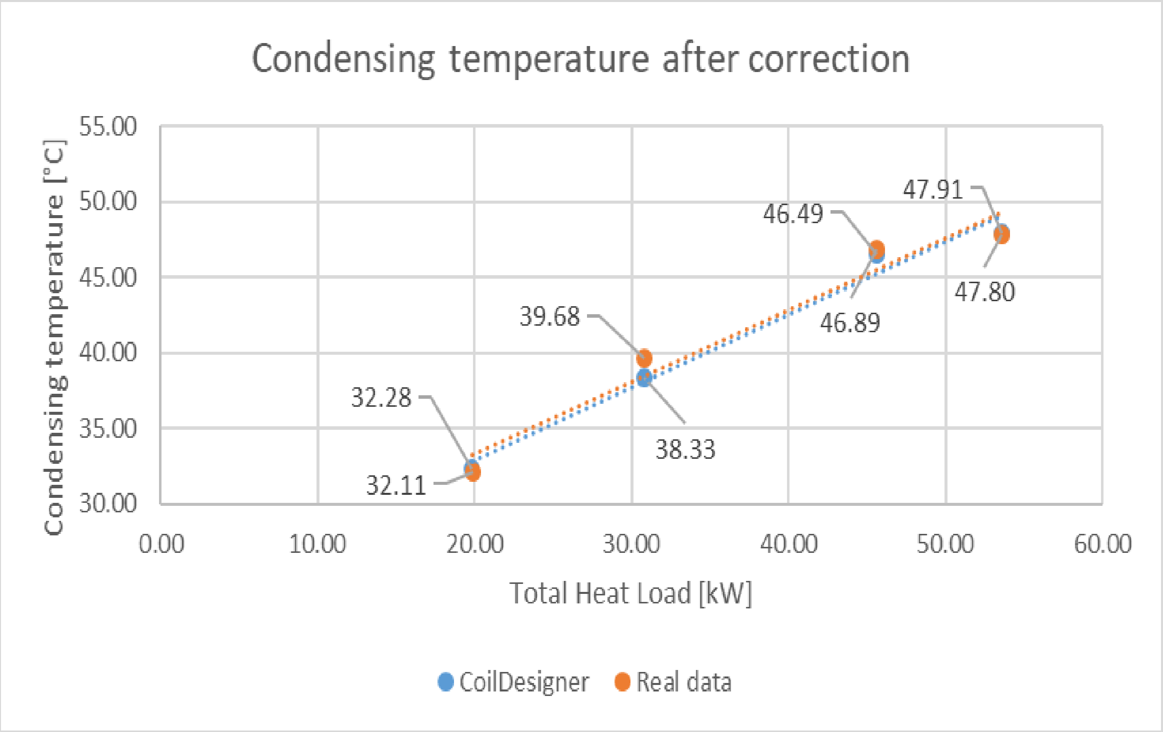


Figure 116: Comparison between condensing temperature simulated with modification and real data

As we can see, before the introduction of corrective factors, as the heat load increased, we noticed a divergence in the values of the condensation temperature up to a maximum deviation of 5.29% corresponding to 2.53 °C of the difference between real test and software simulation. A lower condensing temperature, involves considering the exchanger to achieve a higher and less realistic cycle efficiency. After the correction, the difference between the practical tests and the software is much narrower until a percentage difference between the calculated value and the real value of 0.22% corresponds to 0.11 °C.

7.4.3 Cooling condition VMC heat pump (A-A)

The above comparisons cannot be made for VMC heat pump (A-A) testing. However, simulations were conducted to partially test only evaporation and condensation temperatures. These figures are considered partial and therefore do not reflect the effectiveness of the choices made for correlation correction coefficients.

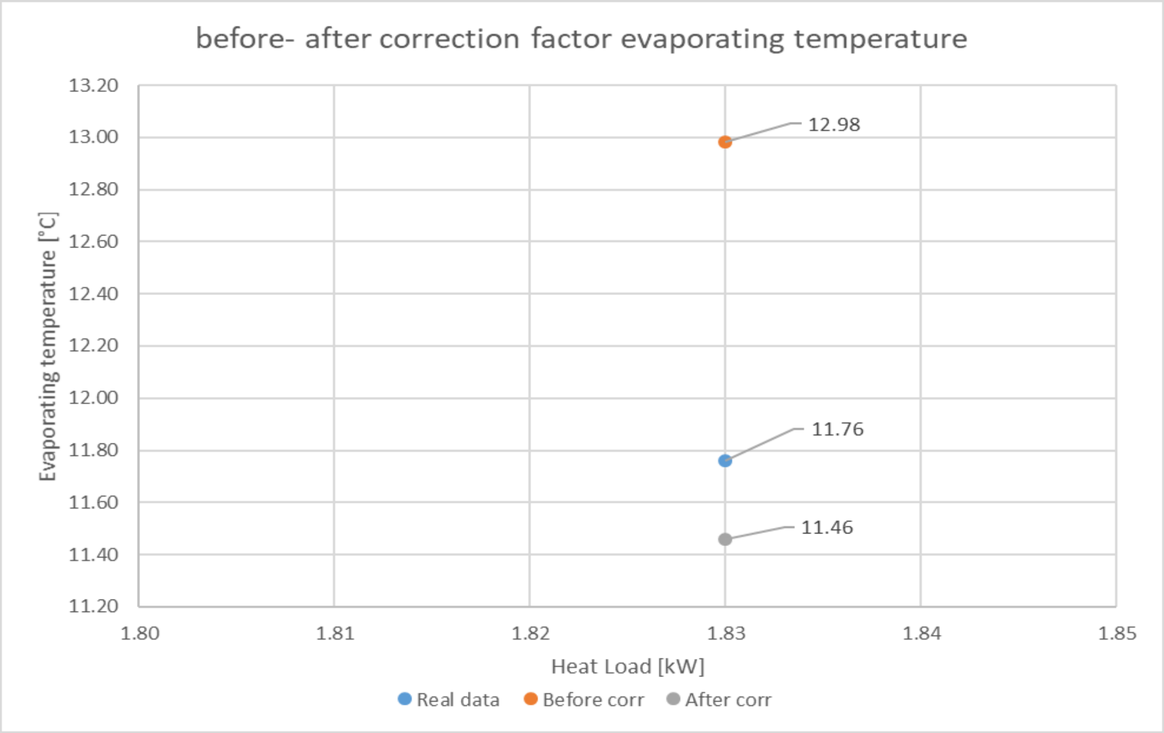


Figure 117: Comparison between evaporating temperatures simulated with, without modification and real data

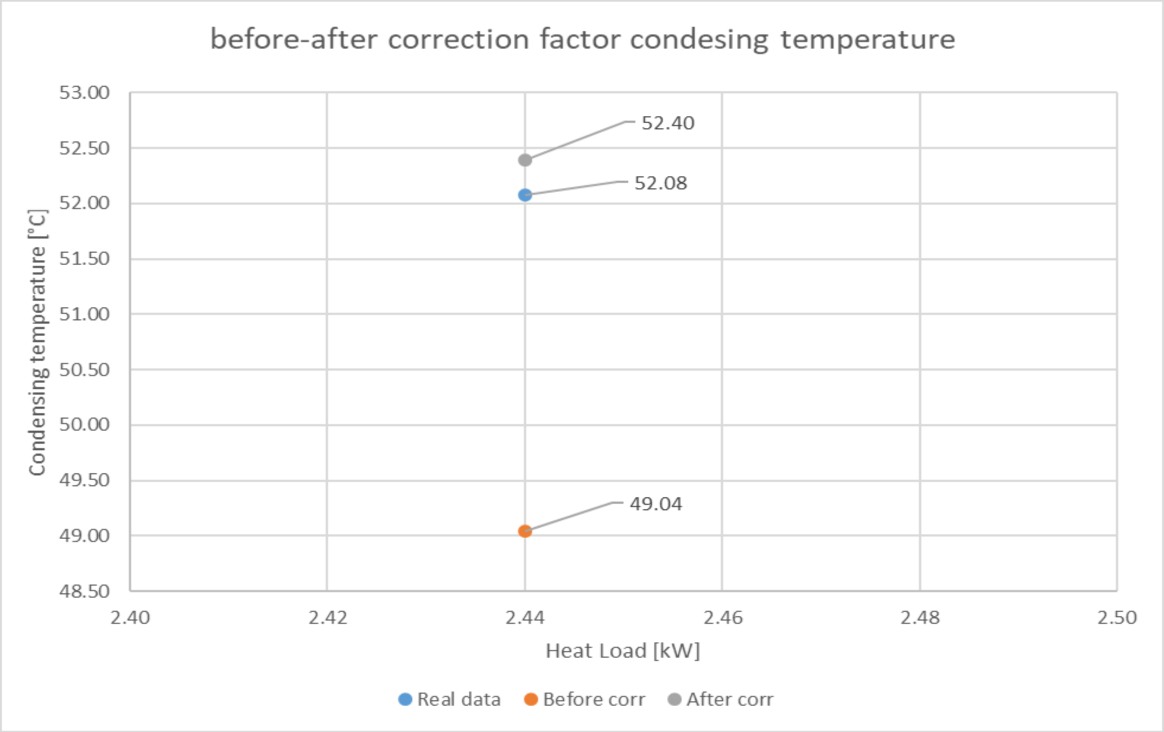


Figure 118: Comparison between condensing temperatures simulated with, without modification and real data

In the evaporation temperature graph, the difference between actual and simulated temperature before the insertion of corrective factors is 10.37% later lowered to 2.55% after correction.

The biggest difference between before and after the use of corrective factors lies on the condensation temperature. In fact, it goes from a 6% error to a 0.6% error. is a remarkable achievement, but as already mentioned does not demonstrate the effectiveness of the chosen values for the corrective coefficients.

7.4.4 Analysis of the results

In both operations of the Scroll heat pump (A-W) there was a difference between the real and calculated value quite high both for the pressure drop refrigerant side and for the evaporation/ condensation temperature. By performing simulations related to a single practical test used as a sample, the corrective factors for pressure drop correlation factor of the two phase state and for heat transfer correlation of the two phase have been modified, simulating the test for different values of the above. Having obtained acceptable results in the sample test, the other laboratory tests in the CoilDesigner software were simulated with the modified corrective parameters. From the values you get, it is noted that the values of the correction factors are correct and usable for all the tests without having to vary their value obtaining a good matching of the pressure drop refrigerant side and the condensing/evaporating temperatures between practical test and CoilDesigner. The values of the corrective factors used are as follows.

	Evaporation operation	Condensation operation
Correction factor for Pressure drop correlation two phase	1.75	2
Correction factor for Heat transfer correlation two phase	0.6	0.22

Figure 119: Final correction factors find to match CoilDesigner with real data

Chapter8: Conclusion

In this thesis, three different heat pumps operating with refrigerants R32 and R513A were initially studied. The goal was to size the fin and tube heat exchangers operating at R290 in order to optimize heat exchange. A theoretical study has been carried out on the differences between the cycles operating with the three refrigerants in order to be able to take knowledge of the main differences that should have been evaluated in order to operate with propane and the compatibility of the various components has been assessed unit.

A theoretical selection of the exchangers operating at R290 was made using a software called CoilDesigner in order to obtain a range of exchanger configurations that could be used on the three units. Starting from a fixed geometry, the parameters of interest related to this analysis were mainly the achievement of the heat exchange required by the project, the evaporation and condensation temperatures as optimal as possible, the verification of the minimum refrigerant speed that ensures the return of the oil to the compressor in order not to have mechanical failures and have been evaluated in function of the number of circuits for each exchanger. The operation of the three units at nominal load and at partial load was evaluated. Thanks to this analysis, the heat exchangers were realized.

Experimental tests were conducted in the heating and cooling operation at nominal and partial load of two of the three total units and then compared the practical results with the relative simulations where the values obtained from the tests were inserted into the input.

We have seen that the analysis of the fin and tube heat exchangers with CoilDesigner is valid, if used correctly the software is able to estimate most of the parameters of interest for the sizing of the heat exchangers.

In particular, for small size heat exchangers, it can also be interesting to analyse the results concerning the temperatures of the refrigerant inside the heat exchanger tubes to obtain a heat exchanger as balanced and efficient as possible. Even small imbalances between circuits lead to large distributions differences and therefore inefficiencies.

The CoilDesigner software tends to overestimate the evaporation temperature, underestimate the condensing temperature and refrigerant pressure drop for heat exchangers operating at R290 in fact idealizing the heat exchange. To have a good matching between simulations and tests on the unit you can act on corrective factors linked to the correlation of heat transfer and pressure drop. The optimum values were identified on one sample test and then applied to all the others. It has been seen that the chosen values returned a greater precision in all the simulations. Thanks to this change, the software error was reduced from 52.6% to 8% on the pressure drop, from 47.7% to 5.2% on the evaporation temperature and from 5.3% to 0.22% on the condensation temperature.

The results obtained are valid only for the scroll heat pump unit (A-W). The VMC heat pump unit (A-A) could not be fully analysed for the impossibility of reading the pressure drop of the refrigerant inside the heat exchangers and the practical tests related to the Screw heat pump (A-W) unit were not carried out in time with this master thesis. However, the three units are operational and reach the standards initially set by the project.

The implications related to these results are multiple in that, you can know the behaviour of the exchanger by software in any external and operational condition without having to conduct laboratory testing. Normally 30 to 90 minutes are needed to conduct a test, the test room must be free, booked well in advance and ATEX certified. Then it must be brought to temperature conditions other than those with considerable expenditure of energy, also need several operators who follow the test. Thanks to the software, it is possible to size and test the exchanger in its entirety before actually creating the unit, thus being able to optimize and test it in many conditions and configurations reducing therefore the times of complete development of the project.

Any future developments related to this thesis are related to the analysis and comparison of the tests that will be conducted on the screw heat pump unit (A-W). Try to use the corrective factor found previously and see if the software can do a good matching of the values of the heat exchangers as was the case for the scroll heat pump (A-W). Test the heat exchangers designed and optimized for the VMC heat pump (A-A) as seen in Chapter 6 and be able to determine if the corrective coefficients found can be used on all sizes of fin and tube heat exchangers.

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