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Calculation and comparison of different supermarket refrigeration systems

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Master of Science Thesis

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RESUME

Ce projet a pour but de comparer différents systèmes de réfrigérations fonctionnant au CO2 et de déterminer leurs potentiels en termes de performance et de consommation au regard de systèmes standards de réfrigération.

Cinq supermarchés fonctionnant au CO2, offrant différentes solutions techniques de réfrigération, et un système conventionnel fonctionnant au R-404A sont précisément analysés. Ces supermarchés sont équipés de l'instrumentation nécessaire pour mesurer températures et pressions. Les données collectées couvrent une période allant de septembre 2008 à aout 2010.

Cette étude démontre les différences de performances entre les fabricants de compresseurs, les améliorations apportées aux évaporateurs au cours des années ainsi que l'effet de la surchauffe.

Des simulations Excel ont permis d'étudier la solution technique de réfrigération en elle même et l'influence de chaque paramètre comme les niveaux de pression, les degrés de surchauffe,… afin de les classer par ordre d'importance.

Les études expérimentales et théoriques menées dans cette étude prouvent que certains systèmes fonctionnant au CO2 peuvent être des solutions efficaces pour remplacer le R-404A.

ABSTRACT

This projects aims to compare CO2 based supermarkets refrigeration systems to determine their potential in term of performance and consumption regarding standard refrigeration systems.

Five supermarkets using CO2 as refrigerant offering different cooling system solutions and one conventional indirect supermarket running with R404-A as refrigerant are deeply analyzed. These supermarkets have the complete instrumentation necessary to measure temperatures and pressures. The collected data cover a period from September 2008 to August 2010.

This study has demonstrated the performance differences between compressor manufactures, the improvements made in the cabinets over the years and the superheating and subcooling effects.

Excel simulations have permitted to study the cooling solution and the influence of each parameter like the pressure levels, the degrees of superheating… in order to classify them by order of influence.

The experimental and theoretical studies reported in this thesis prove that some CO2 based system solutions investigated can be efficient solutions compared to standard R404-A refrigeration systems.

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First I am really pleased to express my sincere gratitude to my supervisor, Samer Sawalha, without whom this thesis work would not have been possible. I sincerely thank you for all the time you spend to help, support or advice me all along my stay in your department. It was really pleasant to work with you.

Furthermore I would like to extend my gratitude to Professor Björn Palm, Applied Thermodynamics and Refrigeration Division of Energy Technology in the Royale Institute of Technology (KTH) who gave me the interest through his lectures to do a master thesis in supermarket refrigeration.

Finally I want to thank my French supervisor, Rémi Revellin, who supported and followed me all along this thesis work.

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NOMENCLATURE

Roman

COP	Coefficient of performance [-]
ΔT	Temperature drop [K]
DX	Direct expansion
FA	Freezer unit
h_{fg}	Latent heat of vaporization [kJ/kg]
KA	Chiller unit
LMTD	Logarithmic Mean Temperature Difference [K]
LR	Load ratio
LR_{corr}	Load ratio correction, fixed value
$\dot{m}$	Mass flow [kg/s]
PPM	Parts per Million
q_v	Volumetric refrigeration effect [kJ/m3]
SC	Subcooling [K]
SH	Superheat [K]
T	Temperature [°C]
$\dot{V}$	Volume flow [m3/s]

Greek

ρ	Density [kg/m3]
η_{is}	Isentropic efficiency [-]
η_v	Volumetric efficiency [-]
η_{is}	Total efficiency [-]
v	Specific volume [m3/kg]

Subscript

amb	Ambient
booster	Booster system
chillers	Chillers
comp	Compressor
freezers	Freezers
in	Inlet
out	Outlet
sat	Saturation

DEFINITIONS

BLEVE: A **b**oiling **l**iquid **e**xpanding **v**apor **e**xplosion occurs when a vessel containing a pressurized liquid above its boiling point ruptures

CFC: **C**hloro**f**luoro**c**arbon is an organic compound that contains carbon, chlorine, and fluorine, produced as a volatile derivative of methane and ethane

GWP: **G**lobal-**w**arming **p**otential is a relative measure of how much heat a greenhouse gas traps in the atmosphere. It compares the amount of heat trapped by a certain mass of the gas in question to the amount heat trapped by a similar mass of carbon dioxide

HCFCs: **H**ydro**c**hloro**f**luoro**c**arbon**s** is a common subclass of the chlorofluorocarbon but it contains also hydrogen

HFCs: **H**ydro**f**luoro**c**arbon**s** is an organic compound that contain only one or a few fluorine atoms

IDLH: **I**mmediate **d**anger to **l**ife and **h**ealth is defined as an exposure to airborne contaminants that is likely to cause death or immediate or delayed permanent adverse health effects or prevent escape from such an environment

ODP: The **o**zone **d**epletion **p**otential of a chemical compound is the relative amount of degradation to the ozone layer it can cause with trichlorofluoromethane (R-11 or CFC-11) being fixed at an ODP of 1.0

Source: http://en.wikipedia.org, 26th June 2011

1. INTRODUCTION

Environment concern in design and development of industrial products has increased in the recent years. Global warming is predicted to rise between 1.5 and 4.5 K in the next 100 years (Campbell, et al., 2007) and this is mainly due to the greenhouse gas emissions into the Earth's atmosphere. Supermarket refrigeration is contributing to this global warming in two ways: directly with gas leakage of HFCs refrigerants through the pipes and indirectly because a lot of electricity is consumed to maintain the supermarkets running. The main challenges in supermarket refrigeration are thus to minimize the direct environmental impacts without increasing the power consumption. The actual trend is to analyze the feasibility of using natural refrigerants in comparison with standard working fluids.

1.1. *Historical trend of CO2 (R-744) as a refrigerant*

CO2 as a working fluid in refrigeration systems has a very long history (Figure 1 below). In 1850 Alexander Twining was the first to propose CO2 as refrigerant for vapor compression cycles (Padalkar, et al., 2010). In late 1800s CO2 was used for cold storage, display cabinets, food market and comfort cooling applications (hospitals, trains, passenger ships).

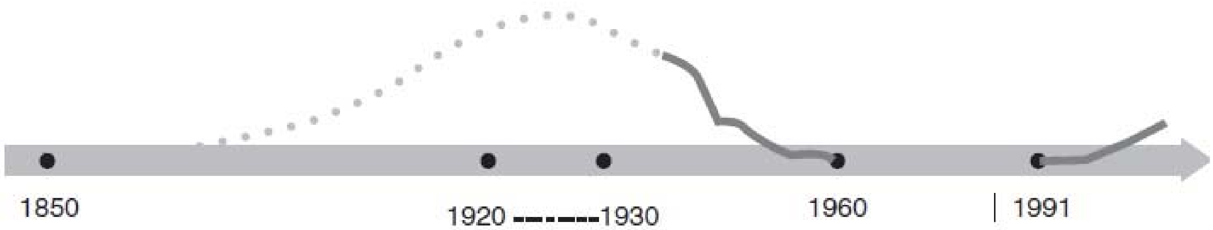

Figure 1: History of R-CO2 as a refrigerant (Campbell, et al., 2007)

However introduction of manmade synthetic chemical refrigerants in the 1930s (CFC, HCFC) had put an end to the use of CO2. The reasons were: better properties offered by chemical refrigerants, their lower working pressure and the inability to develop competitive components and systems for CO2. The use of CO2 has considerably reduced after 1940-50 (Padalkar, et al., 2010).

In 1987 during the Montreal Protocol, the use of CFC and HCFCs was definitely prohibited as they were contributing to the depletion of the ozone layer. CFCs have an ozone depletion potential (ODP) of 1 and HCFCs between 0.005 and 0.2 which is relative to the presence of chlorine in their chemical structure. To overcome this prohibition chemicals industry proposed a new type of synthetic fluid: the HFCs with an ODP of 0.

But even if HFCs are not harmful to the ozone layer their still have a high global warming potential (GWP). For example, R-404A, a refrigerant widely spread in supermarket refrigeration has a GWP of 3800 which is much higher than the reference fluid (CO2) with a GWP of 1. To reduce their impacts on the environment HFCs were integrated into the Kyoto Protocol in 1995 accompanied with various restrictive measures (Frechelox, 2009). Emissions of HFCs are nowadays controlled and amount of HFCs in refrigeration systems is limited to reduce direct gas leakage in the atmosphere (F-Gas regulation). The final objective is to prohibit the use of the HFCs in any kind of application.

Returning to natural refrigerants such as CO2, hydrocarbons (flammable) or ammonic (toxic) like in the early 20th century is seen as a good alternative to match the new legislation. New refrigerants are also created like hydrofluorocarbons (HFO-1234) but there is still no perfect refrigerant and it will probably not be the case in the future. A general overview of all the refrigerants actually available is given on Figure 2.

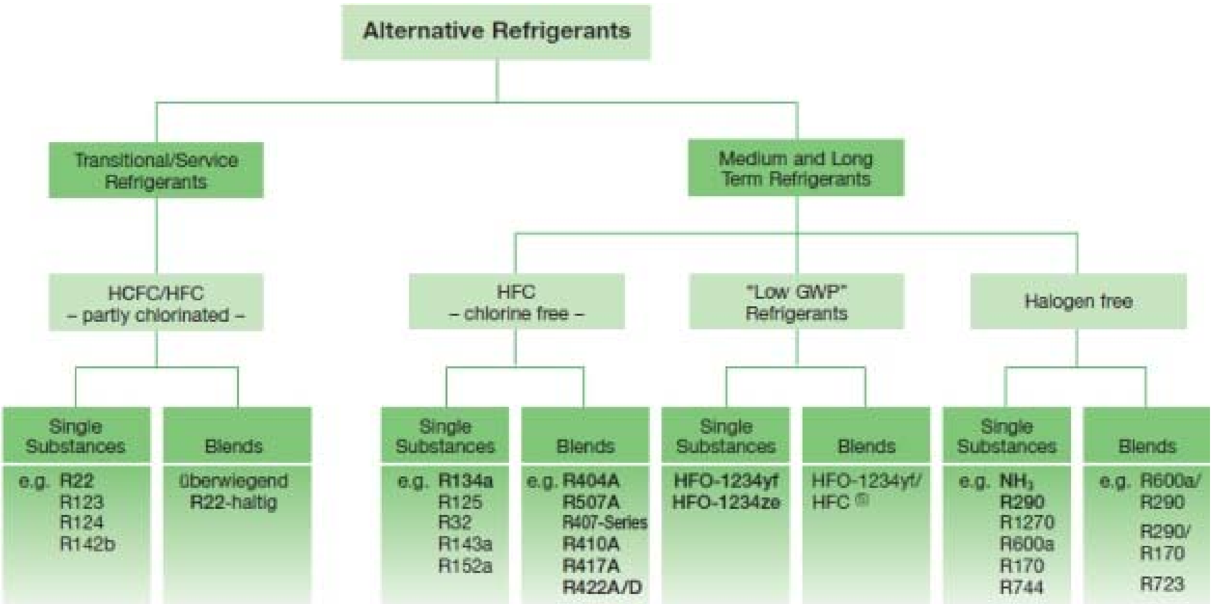

Figure 2: Alternative refrigerants to HFCs (Bitzer, 2010)

By using CO2 the environmental challenge is accomplished but with a potential increase of the energy consumption especially in hot countries. The real problematic nowadays concerning CO2 as refrigerant is to understand and take advantage of its special thermodynamic properties, to improve the technical solutions already existing and to give a proper design to all the refrigerating components in order to match or even surpass the performance of common solutions running with HFCs.

1.2. Energy usage in Swedish supermarkets

The number of stores in 2003 in Sweden was approximately 6100 with an average sale area of 2000m². The supermarket chain ICA had 45% of the total turnover in the total grocery market and both Kooperationen and Axfood had about 22% of the turnover (Arias, 2005). This share may have slightly changed in 2011 as the trend is to build more but smaller supermarkets such as Lidl or Netto with an average sale area of 1600 and 700 m² respectively.

Approximately 3% of the electricity consumed in Sweden is used in supermarkets (1.8 TWh/year). The total energy consumption in a hypermarket is about 326 kWh/m² per year whereas the total energy consumption in small neighborhood shops is about 471 kWh/m² per year (Arias, 2005). Usually the bigger the supermarket is the less it is going to consume energy per square meters.

Figure 3 and Figure 4 below show the cost structure and the energy usage for a standard supermarket in Sweden. Product costs are taking the most important part of the total turnover with 76% of the share. The cost of energy for this supermarket in only 1% of the total turnover but the profit is rather low also with 3%. It means that if the energy consumption is reduced by 50% the profits will increase by 25% (if all the other costs are kept constant) which is quiet significant.

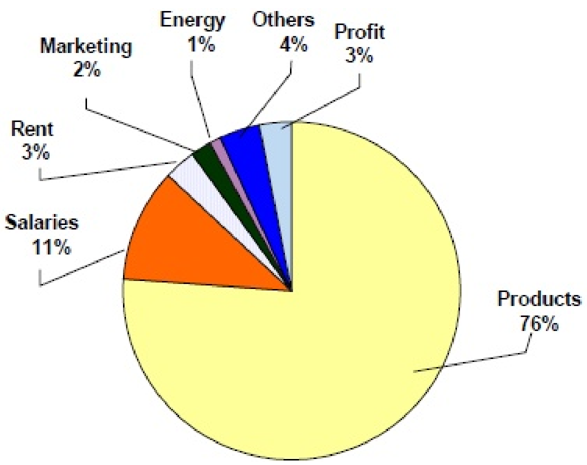

Figure 3: Cost structure for a standard supermarket in Sweden (Arias, 2005)

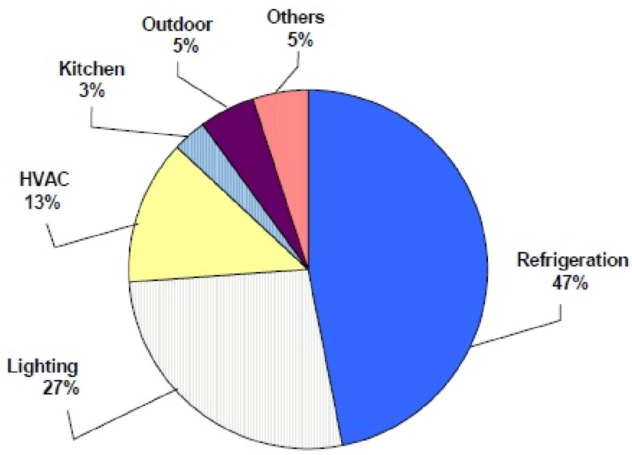

Figure 4: Energy usage for a standard supermarket in Sweden (Arias, 2005)

Finally the energy used in a supermarket comes principally both from the refrigeration system and the lighting. Usually when efficiency improvements are conducted in a supermarket like the refrigeration systems, lightning and HVAC are involved. This shows a great potential of improvement in energy systems.

2. OBJECTIVES

The main objective of this master student thesis is to compare different solutions in supermarket refrigeration running with CO_2 or R-404A/R-404C as refrigerant. These supermarkets present different cooling system configurations as they are not run by the same companies or provided by the same manufactures. The aim is to point out the reasons why they differ in term of efficiency or energy consumption.

2.1. Background

This project is well anchored in the actual context of reducing the greenhouse gases as it deals with CO_2 as a refrigerant in supermarket refrigeration systems.

This master thesis is a continuation of the work that has been done on this subject in the previous years by (Frechelox, 2009), (Tamilarasan, 2009), (Johansson, 2009), (Sawalha, 2008) dealing with CO_2 in supermarket and conducted by KTH and Sveriges Energi & Kylcentrum[1]. By a matter of fact, most of the supermarkets analyzed in this study have already been partially investigated.

However, due to the big amount of data accumulated through the years, it was finally possible to compare these supermarkets all together and start deeper analysis on some critical refrigeration components such as compressors or evaporators. Normalization processes has been conducted in order to make the different supermarkets comparable.

In this study CO_2 is used in transcritical direct expansion (DX) systems as a refrigerant and R-404A/R-404C are both used in indirect systems with a brine loop at the medium temperature level. For CO_2 systems two different configurations are analyzed: booster and parallel system solution.

This project highlights the main differences between standard and CO_2 refrigeration systems and to some extend explains these results with specific analysis. The final aim is to know if a neutral refrigerant such as CO_2 can offer similar performances than standard refrigeration systems and thus become a good alternative for the oncoming future due to his low GWP.

[1] http://www.iuc-sek.se

2.2. *Project description*

Several supermarket installations will be investigated with different CO2 systems and R-404A/R-404C systems in this project. The measuring equipments already exist and summary templates and raw data are available to avoid running new calculations especially concerning cooling capacity, power consumption and indeed COPs.

The project has followed the following objectives:

- Documentation on CO2 and the previous work linked with this topic
- Study of the compressors performance
- Macro Excel for data treatment
- Study of the cabinets performance and external superheating
- Data normalization
- Simulations using excel templates
- Discussions

All the conclusions given in this report are based on concrete measurement made in Swedish supermarkets spread all around the country, the supermarkets locations are pointed out on the map in (Figure 5).

Figure 5: Location of the supermarkets included in this study

3. CO2 TECHNOLOGY

CO2 is a naturally occurring gas, colorless, odorless, non flammable, non explosive, chemically inactive and relatively non toxic. CO2 is present in the atmosphere at a concentration in the range of 350-400 PPM. It's a greenhouse gas playing a key role for the sustainability of plants and the stabilization of the temperature on the earth above 0 °C.

It is environmentally friendly as it has an ODP of 0 and a GWP of 1 which are good requirements for his use in refrigeration. CO2 has very good thermo-physical and transport properties as it will be shown below.

3.1. Properties

The refrigerant properties are very important for the design of refrigeration systems in order to be energy efficient. A Mollier diagram for CO2 is available in Appendix A.

On of the main challenges when using CO2 is to manage to deal with the working pressure and especially on the high pressure side. The saturated pressure is around 5 to 10 times higher than other standard refrigerants (Figure 6). For example the high pressure side for CO2 will be around 1000 bar for a condensing temperature of 40 °C and 10 times lower if R-134a is used instead.

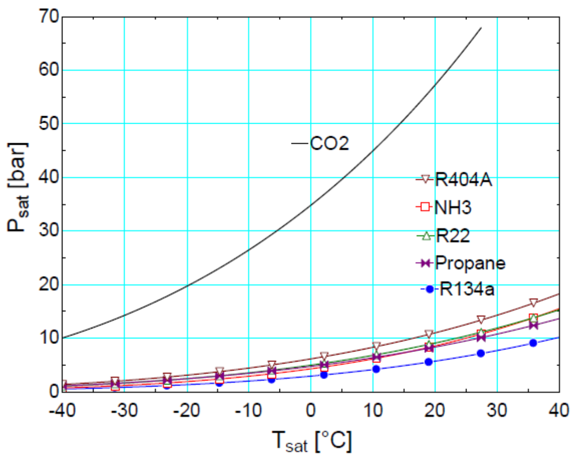

Figure 6: Saturation pressure versus temperature for selected refrigerants (Sawalha, 2008)

This implies for the manufacturers to develop special components like compressors, evaporators, tubes which are scaled to work on these levels of pressure. A good experience in CO2 refrigeration is thus required in the field of brazing or welding.

The saturated vapor density (Figure 7) is higher for CO2. This is obviously an advantage because the volume flow will be lower and it will decrease the refrigerant charge in the system.

Concerning the latent heat of vaporization or condensation, h_{fg}, shown on Figure 8, it is similar to other standard refrigerants (except ammoniac) in a range of 200-400 kJ/kg.

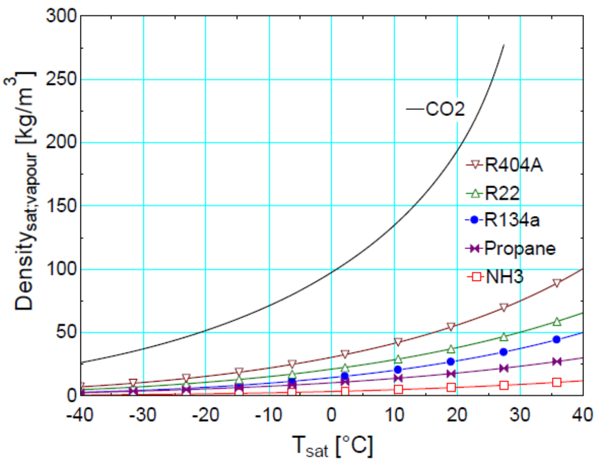

Figure 7: Saturated vapor density for selected refrigerants (Sawalha, 2008)

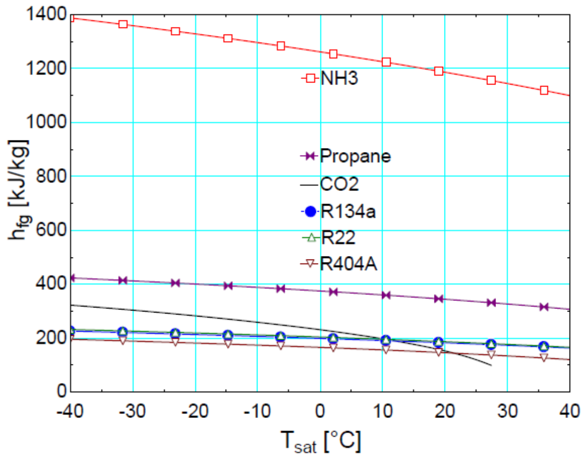

Figure 8: Latent heat of vaporization for selected refrigerants (Sawalha, 2008)

An important parameter in refrigeration is the volumetric refrigeration capacity which is the synthesis of the two figures above. It is defined with the following formula:

(1) $q_v = \rho_{sat.vap.} \cdot h_{fg}$ with $\rho_{sat.vap}$ the saturated vapor pressure

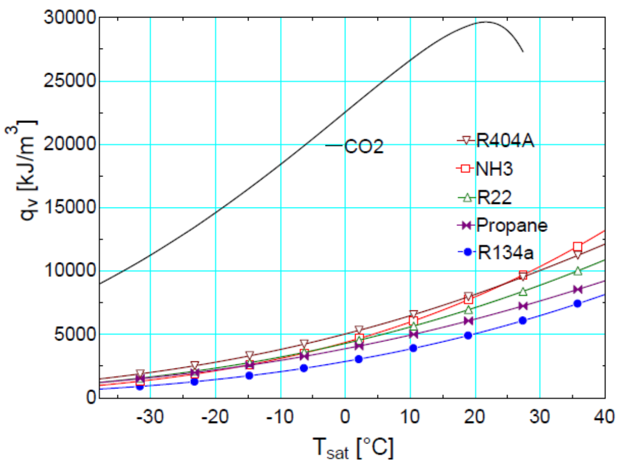

Figure 9: Volumetric refrigeration capacity for selected refrigerants (Sawalha, 2008)

Figure 9 shows that the volumetric refrigeration capacity is much higher for CO2 than other refrigerants. It means that for a fixed cooling capacity the size of the components (compressor, evaporators) will be smaller. Furthermore the tubing size will also decrease as the pressure drop is inversely proportional to the volumetric refrigeration capacity.

The liquid to vapor density ratio (Figure 10) is rather low for CO2 and gives a high momentum for the vapor phase, better shear force between liquid and vapor flow and it will result in a more homogeneous flow.

The surface tension (Figure 11) of CO2 is also lower. A lower surface tension will increase the wetting of the tubes and thus improve the heat transfers. The amount of heat required to convert liquid bubbles to vapor bubbles is reduced. However on the downstream side of the evaporator the increased rate of conversion will have a negative impact on the heat transfers.

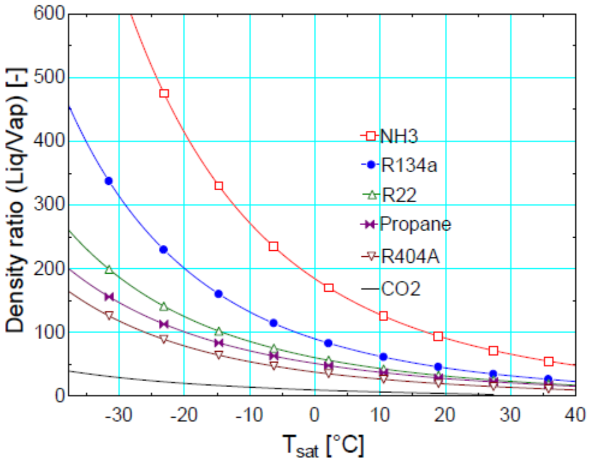

Figure 10 : Liquid to vapor density ratios for selected refrigerants *(Sawalha, 2008)*

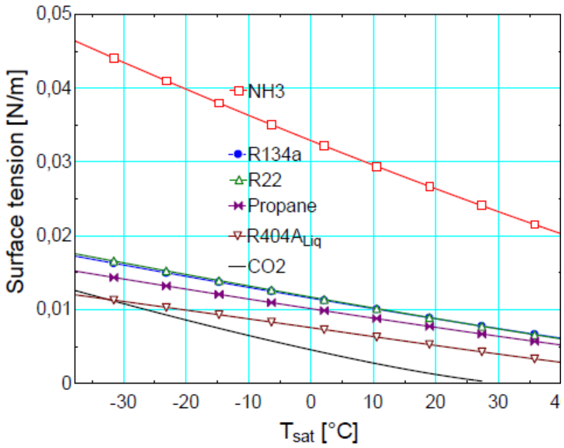

Figure 11: Surface tension for selected refrigerants (Sawalha, 2008)

3.2. Transcritical CO2 vapor compression cycle

In a normal subcritical cycle, the heat rejections and absorptions are standard refrigerant change processes. However, for CO2, the critical temperature is equal to 30.9 °C which is close to the ambient temperatures in some parts of the world. It means that in supercritical regions the pressure is independent of the temperature and the cooling is made at a constant pressure but without phase change (Figure 12).

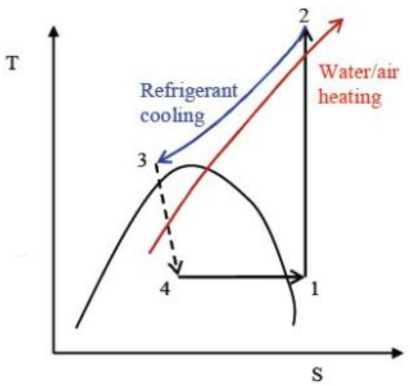

Figure 12: Transcritical vapor compression (Padalkar, et al., 2010)

By a matter of facts, the pressure in the condenser can increase to a maximum pressure for a constant gas cooler outlet temperature. The energy consumption will increase but also the cooling capacity and it will finally result in a COP increase. However, over the maximum pressure the gain in cooling capacity will not compensate the energy consumption anymore. This is because the constant temperature lines become vertical as the gas cooler pressure increases. The pressure control in the gas cooler plays thus a key role in the efficiency of the system for CO2 systems.

Due to a low critical point, CO2 systems will suffer a loss of performance in transcritical regions. But the glide temperature available in gas cooler can also be used for heating applications and may result in a competitive combined COP.

3.3. *Safety issues with CO2*

The natural percentage of CO2 in air is 0.036 %. CO2 is known as a narcotic agent and a cerebral vasodilator (pressure reduction in blood). It also affects the red cells because when they get saturated they will not be able to exchange CO2 for fresh oxygen.

If the concentration goes up by more than 10 % then it will cause a coma. The immediate danger to life and health (IDLH) concentration level is 4 % by volume but the industry is used to work with refrigerants much heavier and toxic than CO2 like ammoniac and it is not considered as a serious issue to use CO2 as a refrigerant (Padalkar, et al., 2010).

As shown in Figure 6, CO2 has 5 to 10 times more vapor pressure than other refrigerants. There will thus always be questions about the safety of CO2 and thickness of materials required to hold this gas in receivers, heat exchangers or tubes.

Explosion or rupture strength of pressure vessels depend on the following parameters:

- Malfunctioning of safety equipments
- Overcharging or overheating
- Incorrect operation conditions
- Structural weakness in components such as corrosion

If an explosion occurs, the severity of casualty depends on the explosion energy (refrigerant charge in individual components, vapor fraction, local temperature and pressure) and the boiling liquid expanding vapor explosion (BLEVE) (Padalkar, et al., 2010).

4. Field installations – Description and layout

In this study five supermarkets running with CO_2 as refrigerant are analyzed against one standard indirect system running with R-404A. They are spread all over Sweden (Figure 5). All the supermarkets have a special given name:

- TR1, TR2, TR3, TR4 and TR5 for the CO_2 systems
- RS1 for the R-404A system

These supermarkets belong to different Swedish supermarket retail.

Table 1 on page 11 below gives a brief description of all the supermarkets (operation date, unit numbers, load, compressor types…). The systems were not in operation; the data retrieval started in 2007 for TR1 and RS1, in 2008 for TR2 and in 2010 for the rest. Hence, the trend over the months will be harder to analyze for the last supermarkets in operation.

Each supermarket is separated into LT- and MT- or FA- and KA- stage which correspond to low and medium temperature stage. The low temperature stage includes the freezers and on the other hand the medium temperature stage includes the chillers. Cold storages are also used to stock bigger amount of food and may belong to LT- or MT- stage depending on the food.

The total load is different from one supermarket to another. For example TR4 is a small local supermarket with a total maximum load of 31 kW and TR1 which is the biggest one has 290 kW of total load.

The complete layouts of the supermarkets are given in Appendix B. It is good to keep these figures in mind in order to understand the refrigeration cycles and also because it is conditioning the results due to the different technological solutions used.

Supermarkets	TR1		TR2		TR3		TR4		TR5		RS1	
	LT-Stage	MT-Stage	LT-Stage	MT-Stage	LT-Stage	MT-Stage	LT-Stage	MT-Stage	LT-Stage	MT-Stage	LT-Stage	MT-Stage
Operation date	Autumn 2007		August 2008		February 2010		May 2010		May 2010		October 2008	
Refrigerant	CO2		CO2		CO2		CO2		CO2		R-404A	
System	Transcritical (parallel)		Transcritical (booster and single stage)		Transcritical (booster)		Transcritical (booster)		Transcritical (booster)		Indirect with a brine loop	
Unit numbers	2	2	2	3	2	2	2	2	2	2	1	1
Maximum load (kW)	60	230	50	200	30	100	6	25	21	55	18	87
Compressor types	2 Dorin TCDH 372 B-D	4 Dorin TCS 373-D	2 Dorin SCS 362	3 Dorin TCS 373-D	1 Dorin SCS 340-D 1 Dorin SCS 362-D	5 Dorin TCS373-D	2 Bitzer 2MHC-05	2 Bitzer 4MTC-7K	4 Bitzer 2HHC-2K	5 Bitzer 4HTC-20KI	2 Bitzer 4VCS-6.2	2 Bitzer 4J-22.2Y
Heat recovery	Yes		Yes		Yes		Yes		Yes		No	
Sub-cooler	No		Ground heat sink		No		Heat exchanger		Heat exchanger		Heat exchanger	
Oil Cooler	Yes		Yes		Yes		No		No		No	

Table 1: System description for all the supermarkets

5. MEASUREMENTS AND EVALUATION METHODS

Data retrieval is obtained through software connected to internet or directly via web pages. Three different solutions were used for the data acquisition as they are not controlled by the same companies:

- IWMAC for TR1, TR2 and TR3 (Iwmac, 2010)
- Long Distance Service (LDS) for TR4 and TR5
- ViSi+ for RS1

Sensors installed in the supermarkets were not especially installed for this study because they are needed to operate the systems and are essential regulation elements. Most of the sensors used are manufactured by the Danfoss Company. The most common types are AKS or HSK according to their pressure range (Danfoss, 2010).

5.1. *Data retrieval*

Each parameter is not taken at the same time for the supermarkets operated through IWMAC. An Excel macro has been created in order to filter the data and to be able to run calculations at a specific time (every five minutes). Figure 13 below shows the filtered data in red.

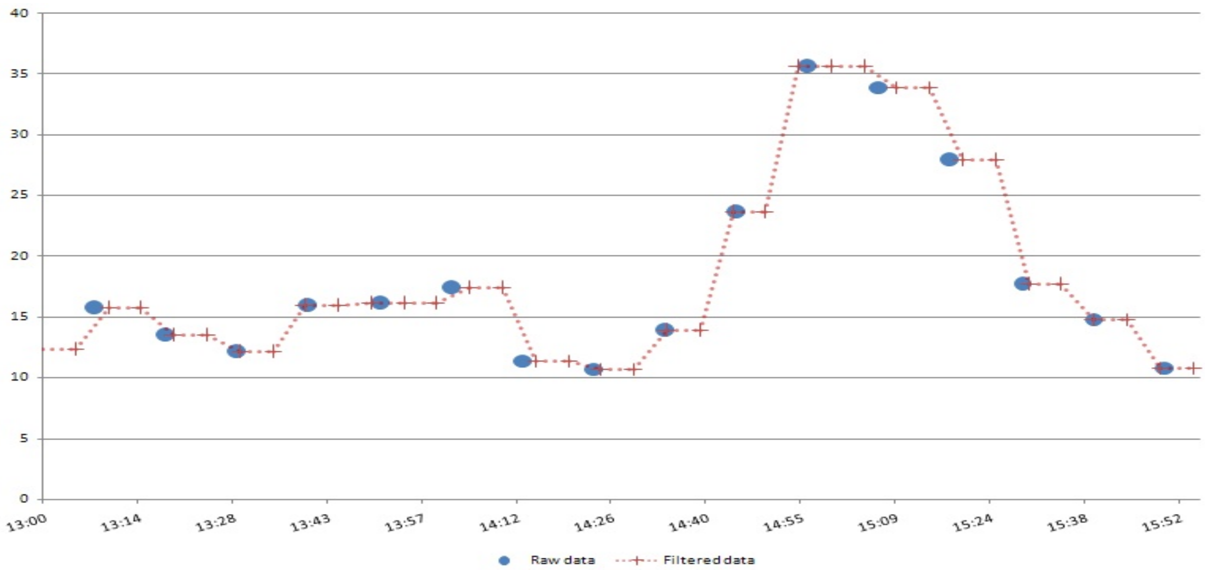

Figure 13: Data retrieval and filtering with Excel

5.2. Mass flow evaluation

Mass flow evaluation is one of the most sensible parameter and there is usually not any mass flow measurement installed by the system manufacture.

However, several methods are available to calculate the mass flow such as the Dabiri's method (Dabiri, et al., 1981), the energy balance method and the volumetric efficiency method. These methods are described in David Frechelox report (Frechelox, 2009) and the final conclusion explains that the method based on the volumetric efficiency is more reliable.

The volumetric efficiency method is based on calculations and manufacture data. Calculations take as inputs the pressure and the temperature inlet of the compressor in order to determine the specific volume. To determine the mass flow the equation (2) is used:

$$(2)\ \dot{m}_r = \frac{\eta_V \cdot \dot{V}_S}{v_{comp\_in}}$$

with

$\dot{m}_r$ [kg/s]: mass flow of refrigerant

η_V [-]: volumetric efficiency (manufacture data)

$\dot{V}_S$ [m^3/s]: Swept volume flow (manufacture data)

v_{comp_in} [m^3/kg]: specific volume (calculated)

As shown in Table 1 two different compressor manufactures are used (Dorin[2] and Bitzer[3]). They are both specialized in CO2 compressors and the manufacture data used in the equation above is relative to these companies and the compressor's type.

5.3. COP Calculation

One of the most important evaluation parameter in refrigeration systems is the coefficient of performance (COP). It gives the ratio between the cooling capacity and the energy consumption of the system. Hence, the higher this value is the better is the performance of the system. The general equation to determine the COP is:

$$(3)\ COP = \frac{Cooling\ capacity}{Electrical\ power\ consumption} = \frac{Q_0}{E_{comp}}$$

As different solutions are used such as parallel, booster or indirect systems it is necessary to explicit the way of calculating the total COP.

[2] http://www.dorin.com/
[3] http://www.bitzer.de/

5.3.1. Parallel systems

COP calculation for a parallel system is rather easy because each unit is separated. The formula used to calculate the total COP will be:

$$(4)\ \ COP_{parallel} = \frac{Q_{o\_freezers}+Q_{o\_chillers}}{E_{freezers}+E_{chillers}}$$

The COP for the two medium stages will be easily defined as:

$$(5)\ \ COP_{chillers} = \frac{Q_{o\_chillers}}{E_{chillers}} \text{ and } COP_{freezers} = \frac{Q_{o\_freezers}}{E_{freezers}}$$

5.3.2. Booster systems

It becomes more complicated for a booster system as the low stage is included in the medium stage. Hence, the flow will be divided into two parts (freezers and chillers) and linked with the following formula:

$$(6)\ \ \dot{m}_{total} = \dot{m}_{freezers} + \dot{m}_{chillers}$$

However the COP will be expressed in the same way as before:

$$(7)\ \ COP_{booster} = \frac{Q_{o\_freezers}+Q_{o\_chillers}}{E_{freezers\_FA}+E_{chillers}}$$

But the formula of the COP for LT- or MT- stage will differ and the energy used in the medium compressor for the medium stage has to be defined ($E_{chillers\_for\_KA}$).

$$(8)\ \ E_{chillers\_for\_KA} = E_{chillers} \cdot \frac{Q_{o\_chillers}}{(Q_{o\_freezers}+Q_{o\_chillers}+E_{freezers\_FA} \cdot \eta_{tot\_FA})}$$

The electricity consumption of the low stage will thus be the summation of the compressor power of the low temperature stage and the energy used in the medium compressor for the low temperature stage.

$$(9)\ \ E_{freezers} = E_{freezers\_FA} + E_{chillers} - E_{chillers\_for\_KA}$$

Finally the expressions of the COP for the medium stage are:

$$(10)\ \ COP_{chillers} = \frac{Q_{o\_chillers}}{E_{chiller\_for\_KA}} \text{ and } COP_{freezers} = \frac{Q_{o\_freezers}}{E_{freezers}}$$

5.4. *Load ratio normalization*

As seen before in Table 1 the total load is not identical for all the supermarkets. Considering the load for the low and the medium stage temperature; a ratio can be introduced to see the share between the medium and the low stage temperature. Usually the load for the medium stage temperature is higher and the formula used to quantify this ratio is:

$$LR = \frac{Q_{o\_chillers}}{Q_{o\_freezers}} \quad (11)$$

However, to be able to compare the different solutions, the load ratio should be normalized at a fixed value. A common value for European supermarkets is 3 which means that the cooling capacity for the medium stage temperature is three times higher than for the low stage temperature.

In order to correct the COP according to a fix load ratio of 3 (LR3) an equation has been especially developed (Frechelox, 2009). The total COP for a fixed load ratio of 3 will be thus calculated by:

$$COP_{tot\_LR3} = \frac{Q_{o\_chillers} \cdot \left(\frac{1+LR3}{LR3}\right)}{\frac{1}{LR3} \cdot E_{freezers\_FA} \cdot \frac{Q_{o\_chillers}}{Q_{o\_freezers}} + E_{chillers}} \quad (12)$$

The full demonstration of this calculation is given in Appendix C.

6. Results from the measurments

As explain before on page 4 both CO2 and R-404A have already been studied over the years. The aim was mainly to determine the total COP of each system and his trend over the months.

In Figure 14 below is plotted the total COP with a load ratio of 3 against the condensing temperature for all the systems.

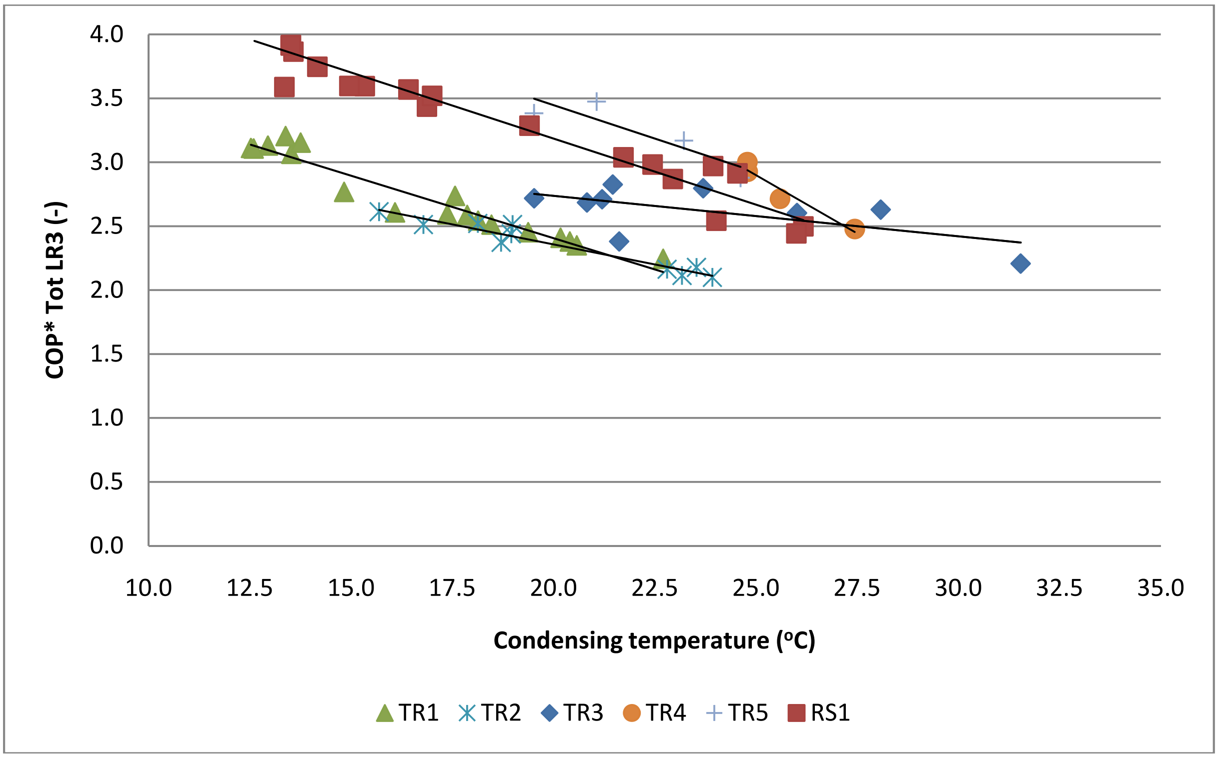

Figure 14: Field measurements of total system COP with a load ratio of 3

This figure will conditioned all the next analysis and it can be basically considered as a starting point. Already a lot of information is given in this single plot in term of efficiency. If it is assumed that a condensing temperature of 20 °C is fixed for the regulation (the case in most European countries) then TR1, TR2 and TR3 are operating with a lower efficiency than the R-404A system but, on the other hand, TR4 and TR5 are operating at a higher COP.

It is clearly shown that CO2 systems can have better performances than standard R-404A systems. All the next part of the report aimed to understand these differences by specific analyses such as for example the technological solution itself, the subcooling or the evaporating temperature.

Finally with all these specific analyses done a model developed in Excel will be used to uniform all the systems in order to perform calculations on a reference model.

7. REFRIGERATION SOLUTIONS

The different supermarkets differ by their technical solutions and the way they are regulated. The aim of this first part is to focus on the technological solutions themselves.

7.1. Mollier diagrams

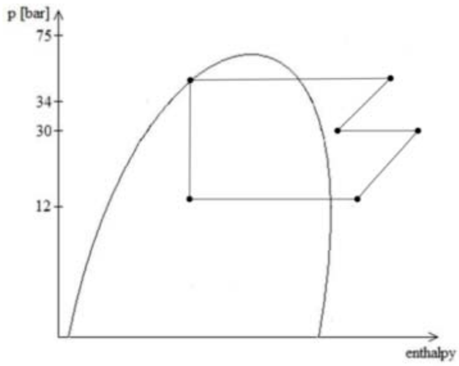

Figure 15: TR1 - FA stage

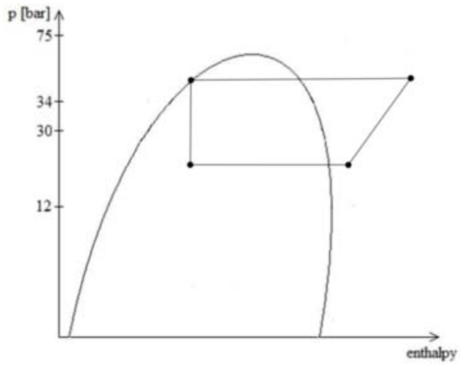

Figure 16: TR1 - KA stage

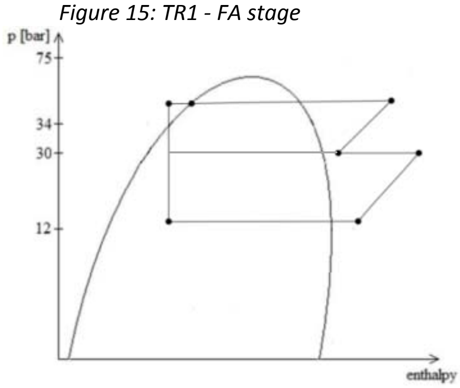

Figure 17: TR2 – KAFA stage

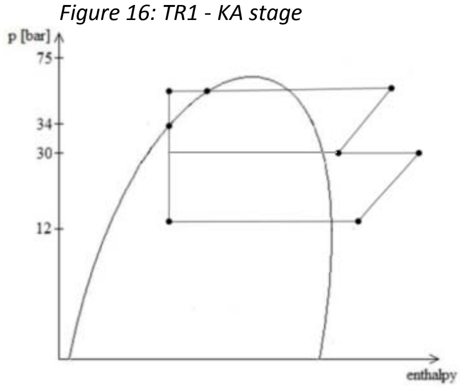

Figure 18: TR3 – KAFA stage

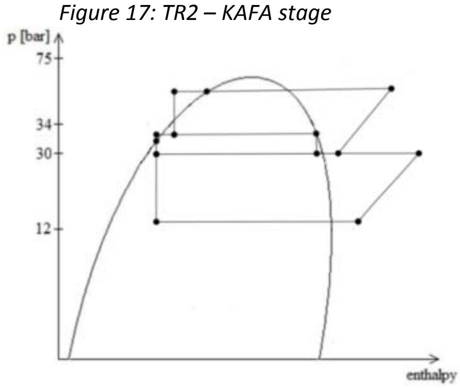

Figure 19: TR4&5 - KAFA stage

7.2. System cooling solution description

Only the technological solutions of each supermarket will be described in this part. Parameters like the pressure levels or the degrees of superheating will be discussed later on.

7.2.1. CO2 systems

TR1, TR2 and TR3 present different system solutions:

TR1 is a basic parallel system. It means that the medium and low stage temperatures are independents. The compression on the low stage temperature, to reduce the power consumption, is achieved with a double stage compressor and an intercooler. The practical advantage of this solution is in case of problems on the lines only one stage will be stopped and the other one will run properly.

TR2 and TR3 are both booster systems. The low and medium stage temperatures are linked together as the return line of the low and medium stage temperature are mixed together before entering the high pressure level compressor. The only difference between TR2 and TR3 is that TR2 is coupled with a borehole to add some subcooling before entering the expansion valve. TR2 has also one medium parallel stage temperature. Booster systems are less convenient in case of problem because if there is an incident on the medium temperature line then both units will be stopped.

TR4 and TR5, regarding their technological solutions, are exactly the same. However there is a heat exchanger and a return line added after the condenser. The enthalpy difference over the evaporators will be higher but the mass flow will be also reduced in the low and medium temperature lines. Hence, the cooling capacity in the medium temperature stage will be kept constant but it may increase the COP of the low stage temperature. The refrigerant charge is expected to be lower due, as explain before, to the return line. TR4 and TR5 are thus supposed to present better performances than the other CO2 systems in term of system cooling solution.

The big difference in total COP between TR1, TR2, TR3 and TR4, TR5 observed on Figure 14 may be somehow linked with the system cooling solution but this effect has to be quantified.

8. Compressor study

There are two types of compressor manufactures used in this study: Bitzer and Dorin compressors. Dorin compressors data used for the calculations come directly from the manufactures' information sheet. For Bitzer compressors, which combine CO2 and standard refrigerants, the data used for the calculation come from a special software developed by the Bitzer Company (Bitzer, 2010).

For all the calculations, the superheating is fixed at 10 °C with no subcooling and the CO2 systems are mainly working subcritical because most of the time the condensing temperature is not exceeding 30 degrees. All the results, the volumetric efficiency and the total efficiency for each compressor, are presented in Appendix D.

8.1. Pressure ratio

CO2 compressors operate at higher pressure levels and with a larger pressure differential than standard refrigerants. However the pressure ratio is lower as shown on Figure 20. Furthermore, for the case of R-134a, the piston displacement is around 7 times higher than CO2 for the same cooling capacity and the expansion losses are also lower.

Despite the high levels of pressure and the shape of its PV diagram, the negative effects of the pressure drop through the valves, CO2 compressors generally have better efficiencies.

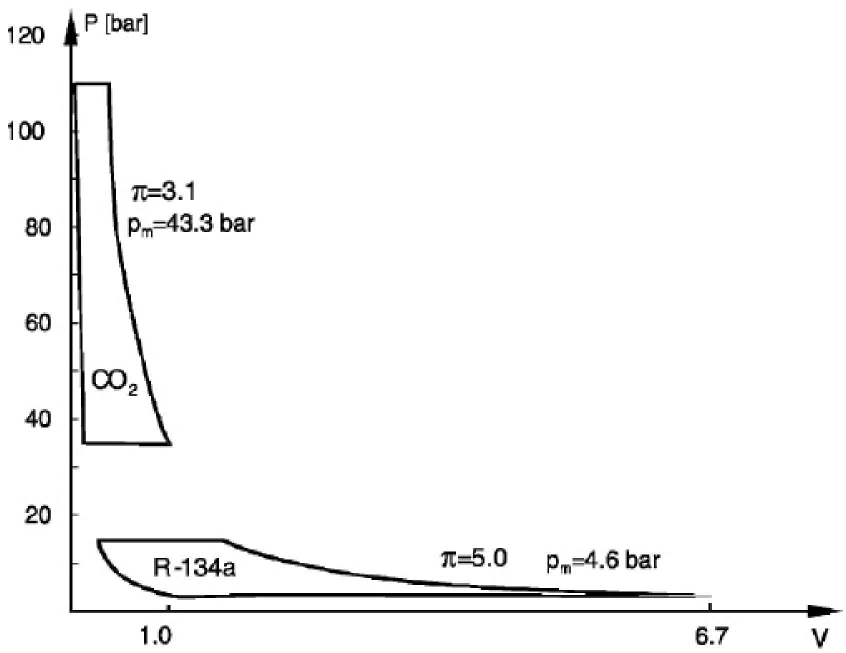

Figure 20: Pressure ratio (Kim, et al., 2003)

8.2. Volumetric efficiency

Figure 21 below shows the volumetric efficiency for the medium temperature stage and for all the supermarkets. Some compressors, even if they are different, have been regrouped because they present the same trend. An average working pressure ratio is highlighted in the figure to see their operating range. All the compressors have a volumetric efficiency in a range of 80 – 90 %. However TR1, TR2 and TR3 offer better volumetric efficiency than the other supermarkets.

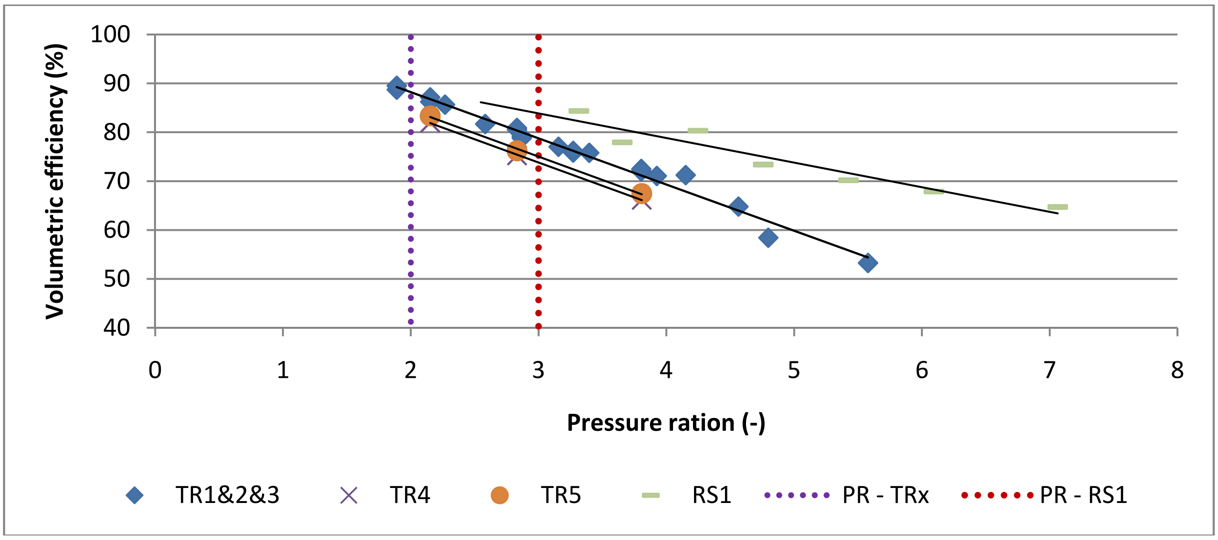

Figure 21: Volumetric efficiency - KA

Regarding the low temperature stage on Figure 22 the differences are slightly higher. TR1 has the best volumetric efficiency as it is a double stage compressor. Furthermore the CO2 systems offer better volumetric efficiencies than R-404A systems as they are operating at higher pressure ratio.

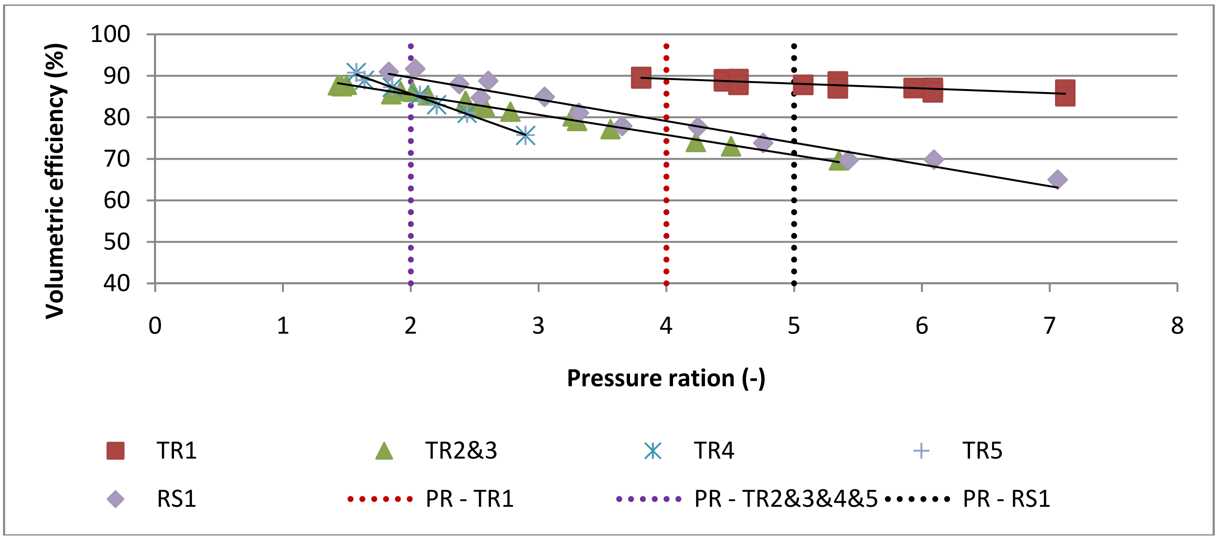

Figure 22: Volumetric efficiency – FA

8.3. *Total efficiency*

Figure 23 below shows the total efficiency for the medium temperature stage and for all the supermarkets. As before, some compressors have been regrouped because they present a similar trend. TR1, TR2, TR3 and RS1 are operating at almost the same total efficiency (60 %). Bitzer compressors used in TR4 and TR5 are operating at a higher value (70 %) which may have strong consequences on the losses of the compressors.

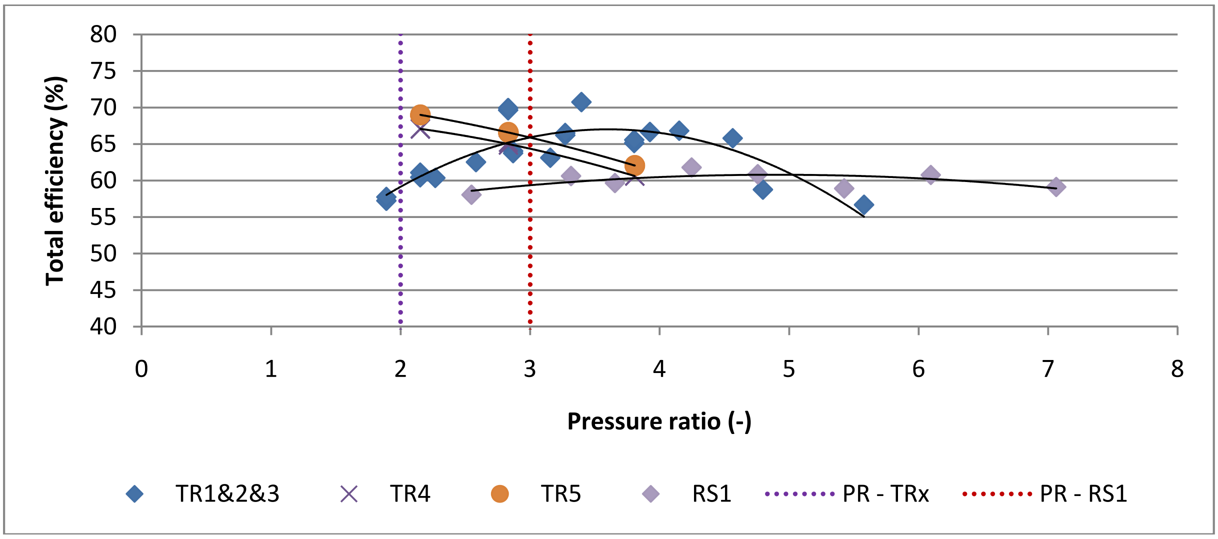

Figure 23: Total efficiency - KA

Regarding the low temperature stage on Figure 24, TR1, TR4, TR5 and RS1 are operating around a value of 60 %. However TR2 and TR3 have a really low total efficiency (45 %) which means that the losses of the compressors on the low stage temperature are rather high.

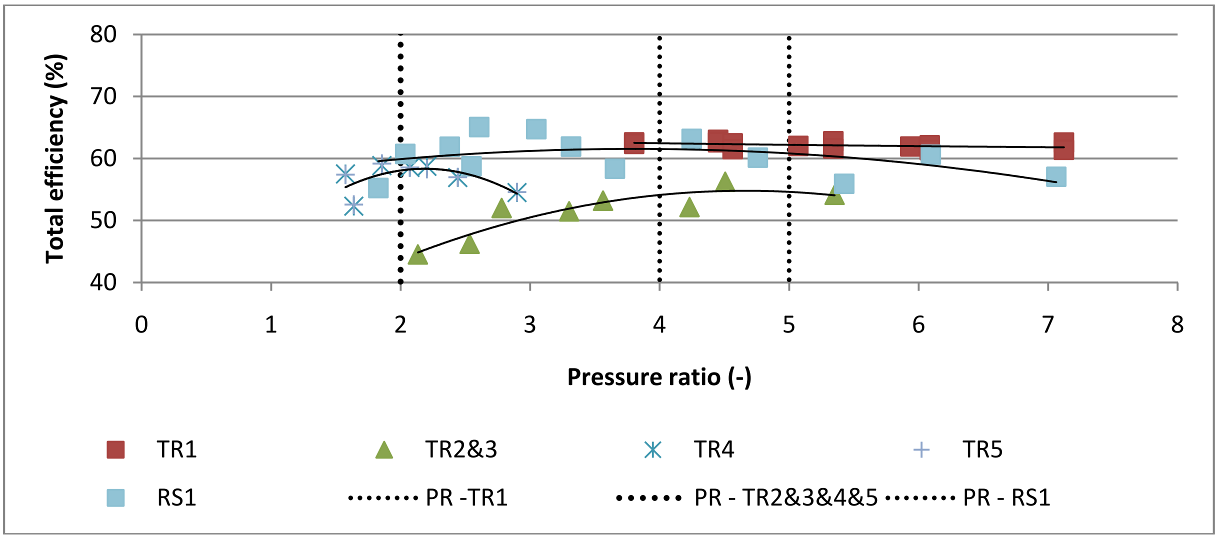

Figure 24: Total efficiency – FA

9. Cabinet study

The cabinets implemented in each supermarket are neither manufactured nor installed by the same company for TR1, TR2 and TR3. Table 2 below shows the data review for the five CO2 supermarkets. Due to the big amount of chillers and freezers; four freezers and six chillers has been taken into the calculations for each supermarket.

All the calculations made for the low and medium temperature stage are available on Appendix E. Some results appear to be strange for some chillers and a few freezers and, in this case, the values have not been taken into account for the plots. For example when the expansion valve is always open or almost always closed the results are not taken into the calculations.

Supermarket	Nb of chillers	Nb of freezers	Data
TR1	63	22	May, August, December
TR2	62	34	May, August, December
TR3	37	26	May, August, December
TR4	3	5	June, August, November
TR5	25	11	June, August, October

Table 2: Data review

9.1. Danfoss controllers

Control of the evaporator and thermostatic expansion valves is realized with Danfoss controllers. The main diagram on Figure 25 explains where the temperature measurements are done.

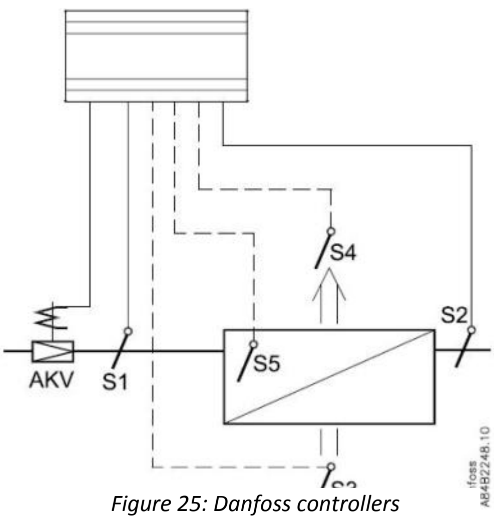

Figure 25: Danfoss controllers

The sensor notation is defined as:

S1 – Inlet of the evaporator
S2 – Outlet of the evaporator
S3 – Return air
S4 – Supply air
S5 – Mean temperature of the unit
Pe – Low pressure

Superheat is measured by using the temperature sensor S2 and the pressure sensor Pe which is a more accurate method than a measurement based on temperatures.

9.2. Averages calculation

The plots are directly based on averages. It is then necessary to assure than the values used are reflecting reality and that they are in some way controlled.

The main parameter which is influencing the averages (superheat, return air) is the defrost state. When the defrost mode is activated, the outlet temperature of the evaporator will increase and the temperature of the air in the cabinets will be influenced. Hence, as the supermarkets don't have the same defrost state frequency, all the averages in this study have to be based on a state where defrost is not occurring. It means that the points which are corresponding to a defrosting mode have to be removed from the calculations.

For chillers the reaction time when the defrost mode stops is almost instantly. Figure 26 below shows the evolution of the return air temperature in the evaporator for a standard chiller during one day. Defrosting modes is occurring around four times during in one day and is lasting almost 1 hour.

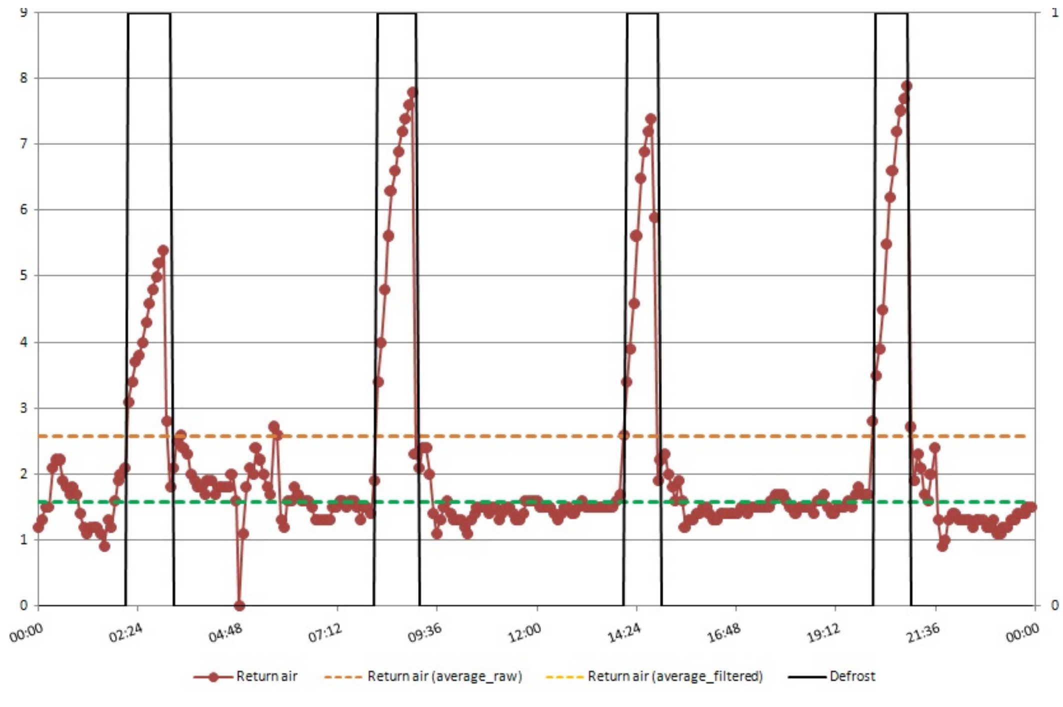

Figure 26: Return air temperature for a standard chiller during one day

As the reaction time is rather fast it is assumed in this study that for all the chillers when the defrost sensor is activated the values are not kept and when it is deactivated the values are taken into account for the averages.

The same plot for a freezer is shown on Figure 27.

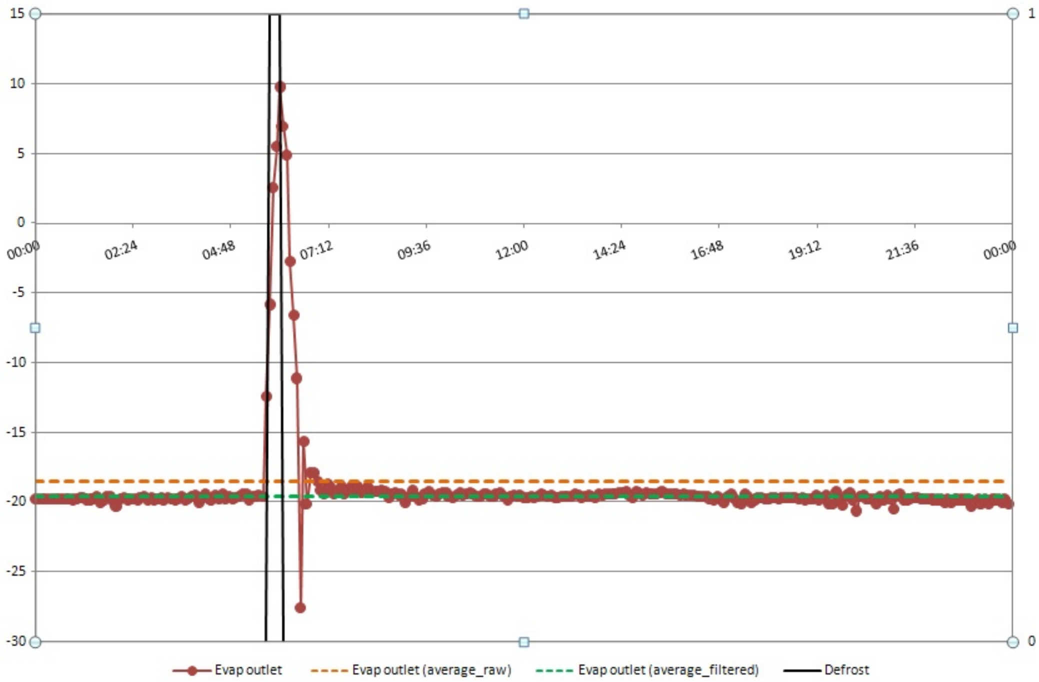

Figure 27: Outlet temperature of the evaporator for a standard freezer during one day

It can be noticed that the defrosting period is shorter for a freezer than a chiller (about 15 minutes which is four times faster than for a chiller). It is mainly due to the way the evaporator is heated up by electric heater while the medium temperature evaporator is defrosted by circulating the warm air with no refrigerant flow.

On the other hand, the reaction time for a freezer is low compared to a chiller which means that it is not possible to keep the same data treatment as before.

The values keep for the calculations are in this case depending on the outlet temperature of the evaporator which is increasing when the defrost mode is activated. The values are used for the averages when the outlet temperature of the evaporator is included in a range of +/- 5 °C of the monthly average of this value.

The reaction time of the system to reach the reference working conditions when the defrost mode is deactivated is then well estimated and to base the calculation on the outlet temperature is more accurate than on the defrost sensor.

This is even more relevant for some supermarkets when the number of points increases when there are some fluctuations which give indirectly more weight on these points.

9.3. Temperature evolution on the air side

The inlet and outlet temperature on the air side correspond to the temperature of the air when it enters/leaves the evaporator. It is presented in the following sub-sections.

9.3.1. Low temperature stage

The supply and return air temperatures are the main important parameter as it fix the effective temperature in the freezer. The regulation is set to keep the products at a temperature lower than -18 °C in order to stop bacteria from forming and allow food to be stored for long periods of time.

All the freezers have a supply air temperature close to -27 °C and a return air temperature closed to -21 °C (Figure 28). TR2 is working with a lower return air temperature (-22 °C) and TR1 with a higher return temperature (18 °C). An efficient cabinet has a ΔT between the inlet and the outlet temperature of the air as low as possible. Due to this, TR1 cabinets present a lack of efficiency and some energy is lost.

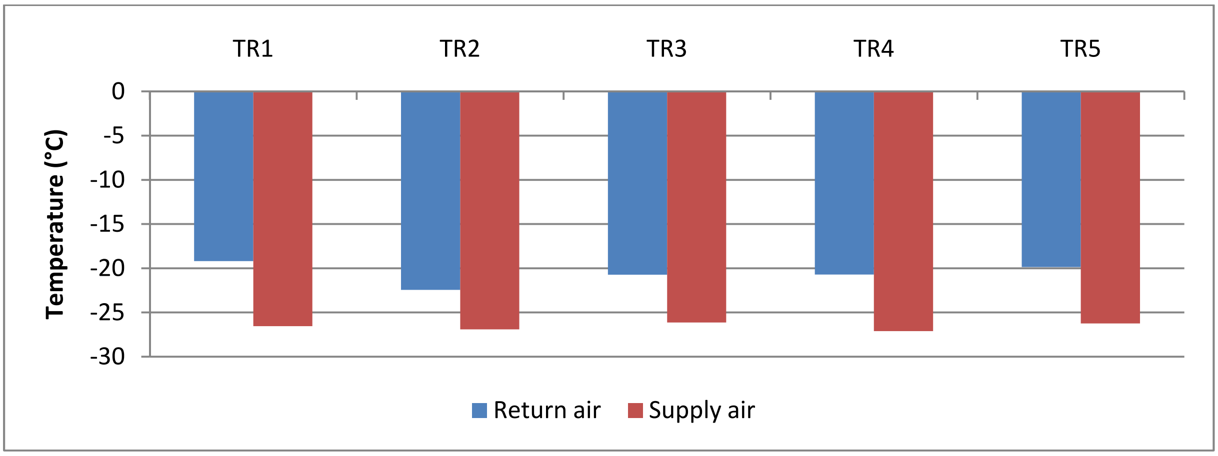

Figure 28: Inlet and outlet temperature on the air side of the low temperature evaporator in the different supermarkets

9.3.2. Medium temperature stage

Generally the analysis of the medium temperature stage is harder due to higher fluctuations among the chillers. For chillers the regulation is set to keep the products in a range of -1-4 °C.

As expected, on Figure 29, the return air is always positive but is different among all the supermarkets. It is around 4 °C for TR1 and TR2 and around 2 °C for TR3, TR4 and TR5.

Concerning the return air the differences are larger and the temperature can be either positive (TR1, TR2, TR3 and TR5) or negative (TR4). The supply air for TR4 is relatively low compared to the other systems (-1 °C).

These values have to be compared with the inlet and outlet temperature of the evaporator in order to try to find an explanation. It has also to be noticed that the scale of this plot is larger than for the freezers and may give bigger importance to smaller fluctuations.

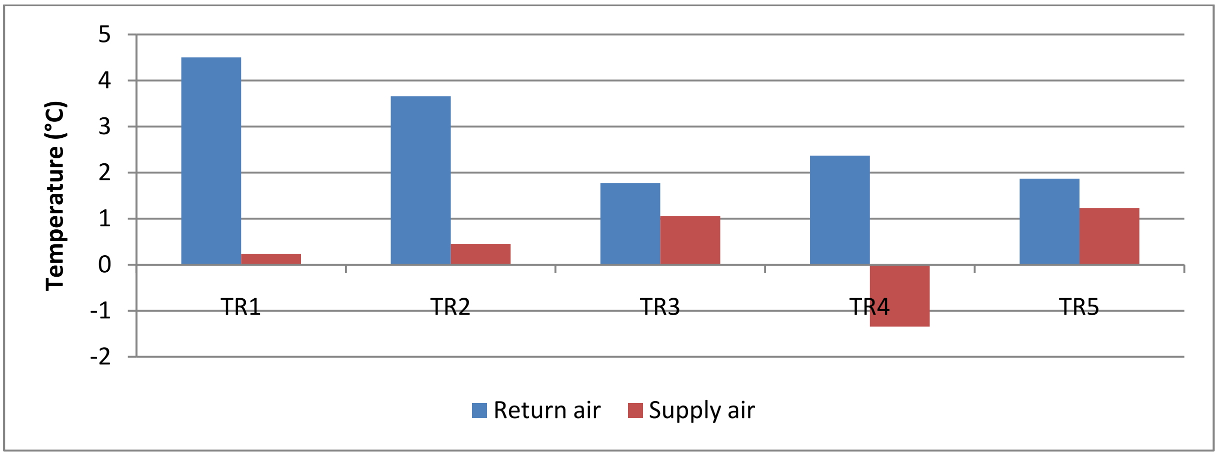

Figure 29: Inlet and outlet temperature on the air side of the medium temperature evaporator in the different supermarkets

9.4. Temperature evolution on the refrigerant side

The inlet and outlet temperatures on the refrigerant side correspond to the temperature of the refrigerant when it enters/leaves the evaporator. Hence, the inlet and outlet temperatures of the refrigerant will be respectively always lower than the supply and the return air temperature.

9.4.1. Low temperature stage

The inlet temperature can be averaged at -31 °C and the outlet temperature at -22 °C. TR1 and TR2 have a lower inlet temperature (-34 °C) than the other supermarkets.

However, they all have the same supply air temperature and indeed this difference could be explained by the evaporator efficiency or the rate of the air.

The ΔT between inlet and outlet of the evaporator (internal superheating) is equals to 12 K for TR1 and TR2 and equals to about 7 K for TR3, TR4 and TR5.

The ΔT with the air at the inlet of the evaporator is then in a range of 3-7 K and 1-4 K for the evaporator outlet. These variations will be explained in page 28 considering the LMTD value of each unit.

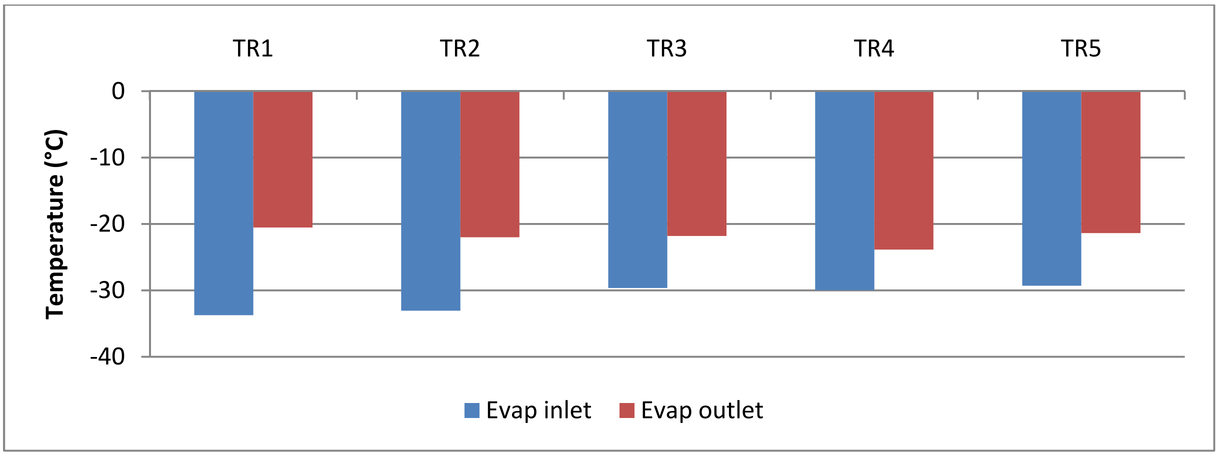

Figure 30: Inlet and outlet temperature on the refrigerant side of the low temperature evaporator for all the supermarkets

9.4.2. Medium temperature stage

The fluctuations on the medium stage for the chillers are smaller than on the air side but the plots are not sufficient to explain all the results.

Evaporator inlet is around -6 °C and evaporator outlet is around 3 °C for the supermarkets but only 0.5 °C for TR4. This plot is not helpful to explain TR4 case. The evaporator outlet temperature is low but the return air for TR4 is in the range of the other supermarkets. The same analysis can be drawn for the evaporator temperature inlet and the supply air.

The ΔT between inlet and outlet of the evaporator (superheating) is equal to 11 K for TR1, TR2 and TR3 and equal to 7 K for TR4 and TR5.

The ΔT with the air at the inlet of the evaporator can be averaged to 7 K and 1 K for the evaporator outlet.

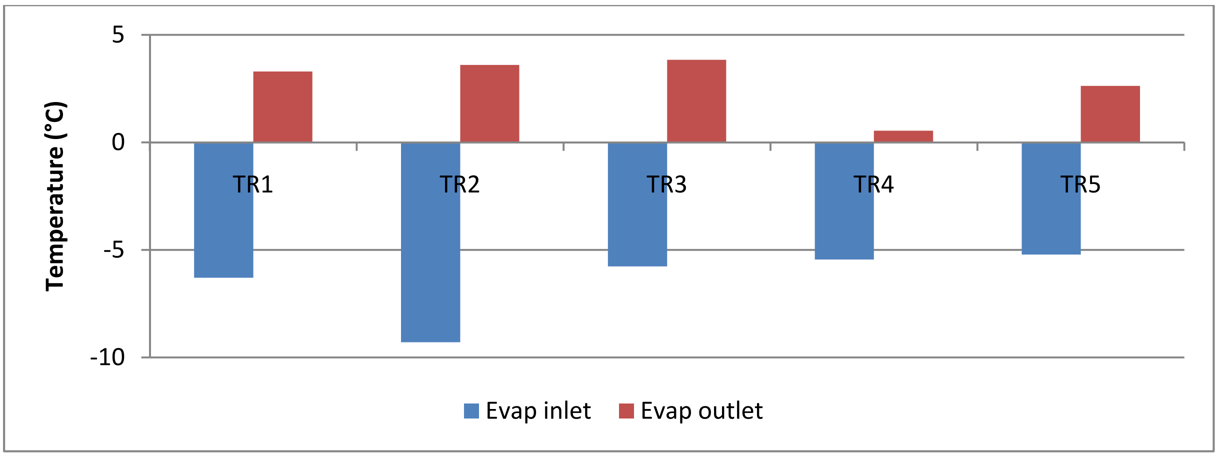

Figure 31: Inlet and outlet temperature on the refrigerant side of the medium temperature evaporator for all the supermarkets

9.5. Logarithmic mean temperature difference

The temperature profile for the air and the refrigerant in an evaporator can be approximated according to Figure 32 if the superheating in the evaporator is neglected.

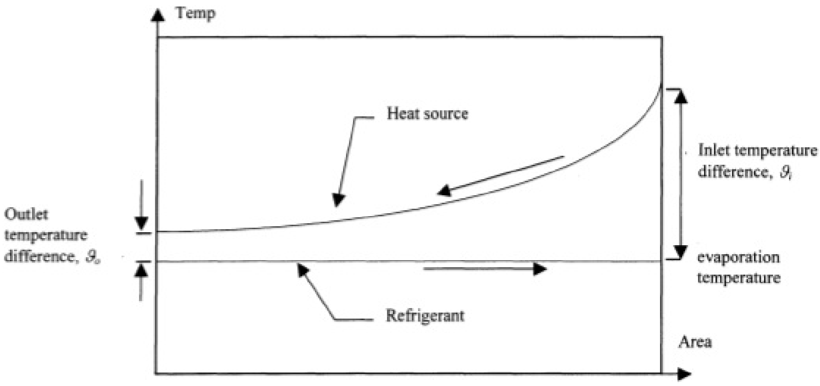

Figure 32: Temperature profile in evaporator

The logarithmic mean temperature difference can be defined as:

$$\text{LMTD} = \frac{(\theta_1 - \theta_0)}{LN\left(\frac{\theta_1}{\theta_0}\right)} \qquad (13)$$

Huge differences of temperatures between the air and the refrigerant side have to be small to lower the energy consumption. Figure 33 and Figure 34 show the LMTD for the medium and low temperature stage. LMTD for TR3, TR4 and TR5 can be averaged around 6 K for both units and around 10 K for TR1 and TR2.

Hence these figures regroup both the level of temperatures on the refrigerant or the air side. The higher values obtained for TR1 and TR2 clearly show that they will consume more energy.

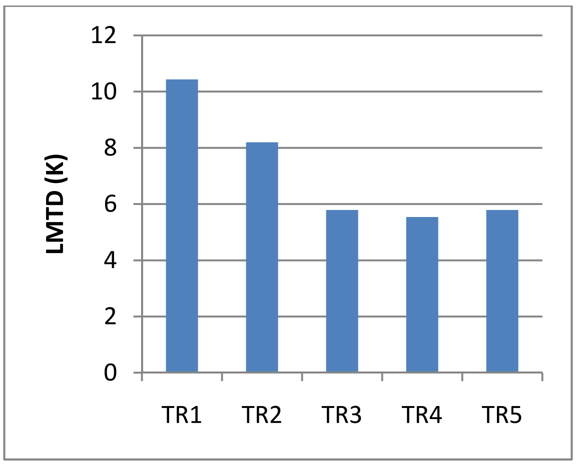

Figure 33: LMTD for the low temperature stage

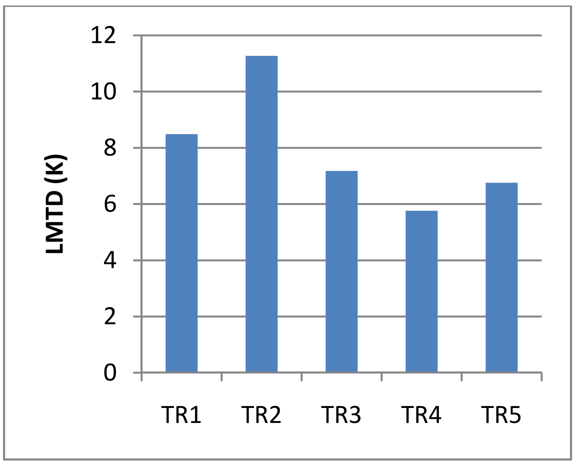

Figure 34: LMTD for the medium temperature stage

9.6. Defrost

Defrost occurs quiet regularly in the units in order to prevent the evaporator from frosting and lowering the area used for the heat transfer between the refrigerant and the air. This effect is taken out in all the previous plots as the defrost frequency is not the same for all the supermarkets.

9.6.1. Low temperature stage

Usually for freezers defrosting is realized with electrical resistances. It can be noticed that the defrost percentage of the time is not the same for all the freezers especially for TR4.

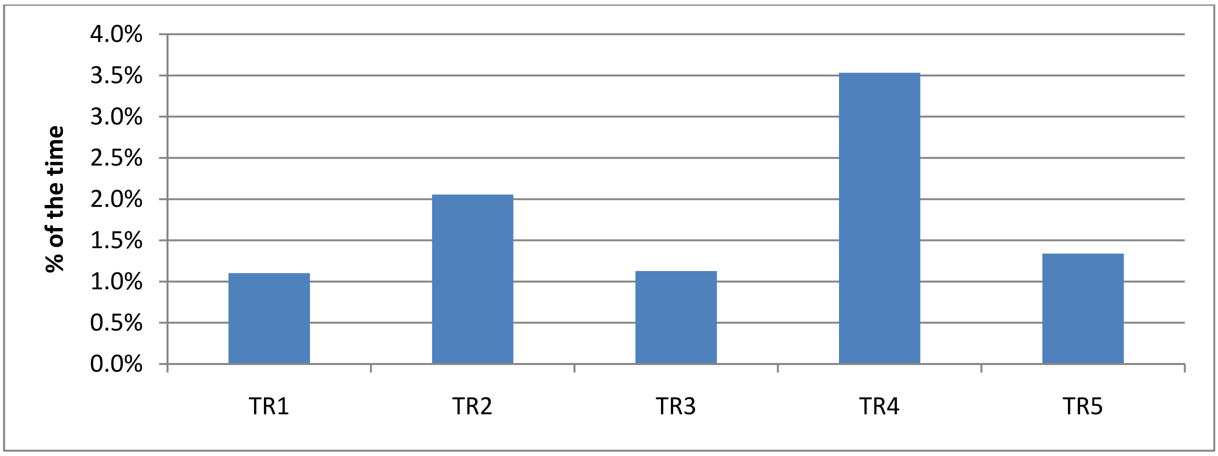

Figure 35: Defrosting time

9.6.2. Medium temperature stage

TR1 and TR2 are in defrost mode 9 % of the time and TR3, TR4 and TR5 are in a defrost mode more often, 13 % of the time.

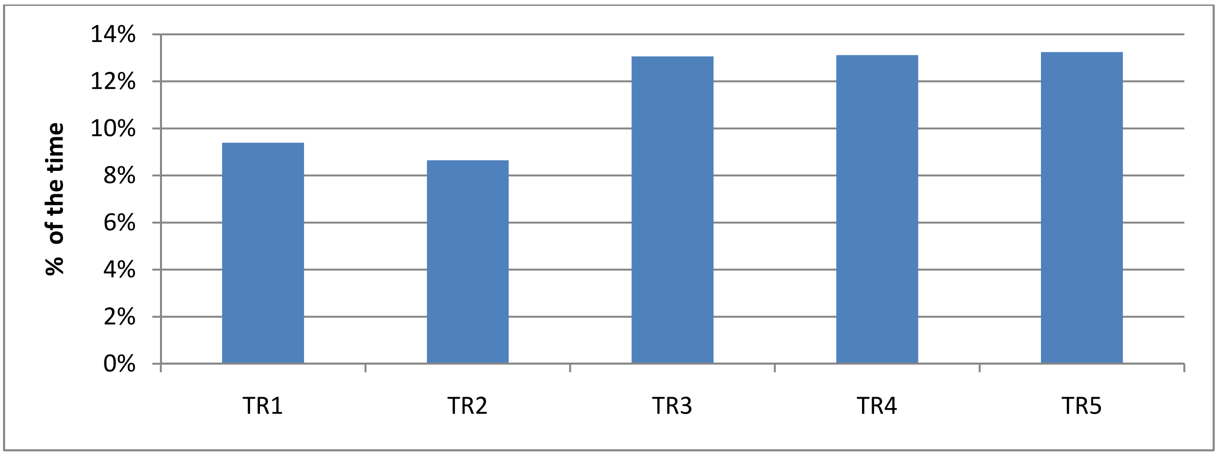

Figure 36: Defrosting time

10. SUPERHEATING STUDY

Two different kind of superheating have to be defined: the internal and the external superheat. The internal superheat occurs in the evaporator and is defined as the temperature difference between the outlet temperature of the evaporator on the refrigerant side and the temperature of the refrigerant at the saturated point with the same pressure. On the other hand, the external superheat is defined as the temperature difference between the inlet temperature of the compressor and the outlet temperature of the evaporator on the refrigerant side.

External superheat has always a negative impact on the system efficiency but it has to be considered that a vapor phase is needed at the inlet of the compressor. A common minimum external superheating value for CO2 systems is assumed to be around 12 K (Frechelox, 2009).

All the values used in this study are available on Appendix F.

10.1. Low temperature stage

Figure 37 below shows the internal and the external superheat for all the supermarkets. It can be clearly seen that the superheating values are far over the minimum value required especially for the first three supermarkets. However TR4 and TR5 have a better total superheat value close to 16 °C. External superheat for TR1 is very extremely high as it is a parallel system and because the condenser for the low temperature stage is set at the same pressure level as for the medium temperature level. Higher values of superheating will have a strong negative impact on the system efficiency.

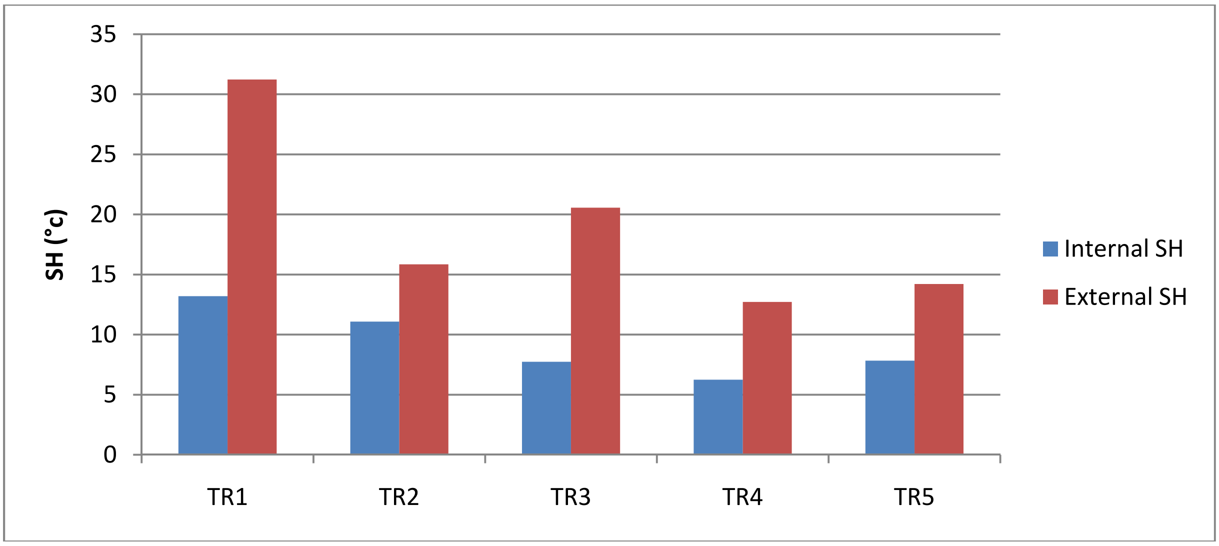

Figure 37: Superheat for the low temperature stage

10.2. Medium temperature stage

Figure 38 below shows the superheating values for the medium temperature. TR1, TR4 and TR5 have reasonable values of total superheat but it is rather high for TR2 and TR3 because the evaporating temperature of TR2 is lower than the other systems and the return air temperature for TR3 is also low compared to the other supermarkets.

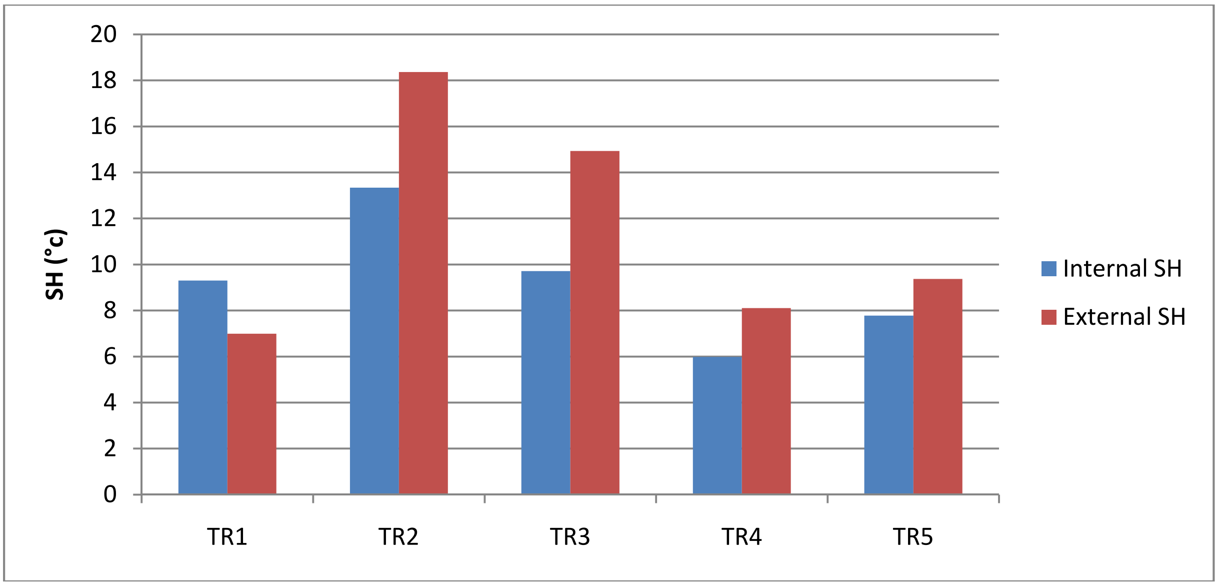

Figure 38: Superheat for the medium temperature stage

All these effects analyzed in details have now to be quantified to point out which one of these parameters is the most important.

11. SIMULATIONS

In order to be able to identify which are the most important parameters influencing the total COP some simulations using excel have been developed.

On Figure 39 below is plotted the total COP against the condensing temperature. The aim of these simulations is to start from a reference model where all the input parameters are the same and to add progressively the real value of each parameter for each supermarket. By doing this the system solution itself can be studied and the order of importance for all the parameters as well. It can be seen as a decomposition of the problem.

The reference model is defined at a condensing temperature of 20 °C. To know the experimental value of the COP (for TR1 and TR4) for this condensing temperature the curve has been extrapolated with a linear regression. Furthermore due to the recent analysis of TR4 and TR5 there are not so many data available to have a wide range of condensing temperature. Hence the experimental values have to be considered carefully.

At the end, with all the specific values added to each supermarket the numeric model should be as close as possible to the experimental value and if not some explications have to be given to explain these differences.

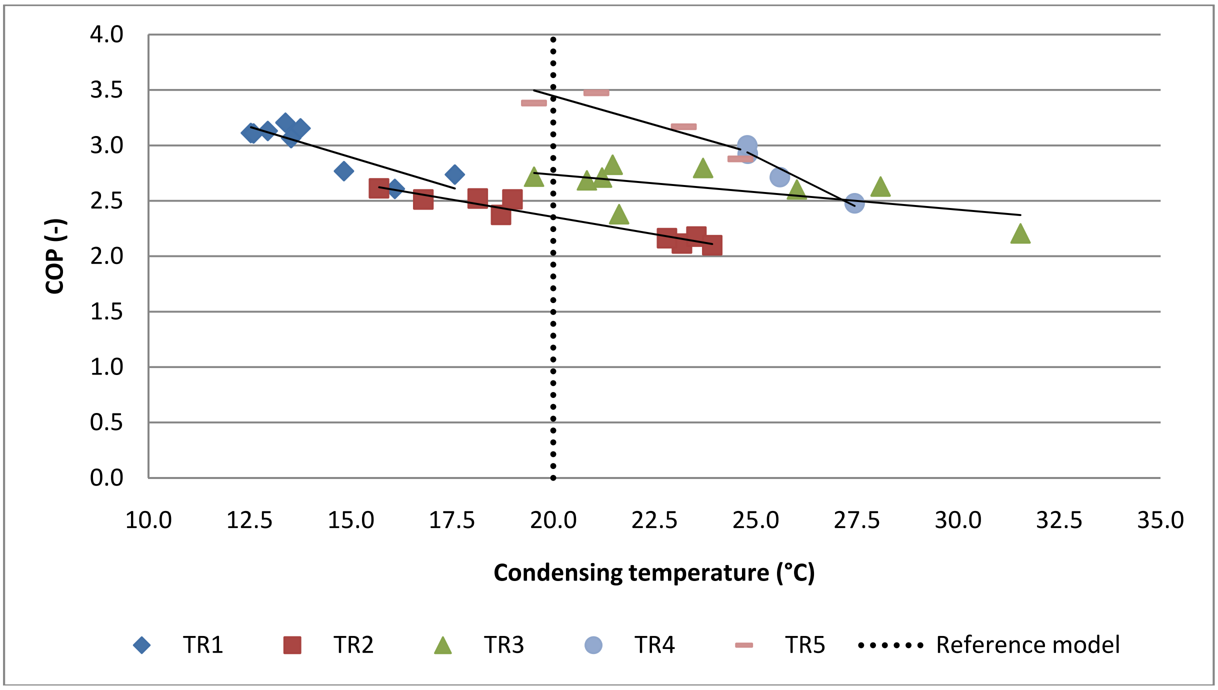

Figure 39: Total COP for all the supermarkets

11.1. Definition of the reference model

The cooling capacities are fixed to 100 kW for the medium temperature stage and 30 kW for the low temperature stage. The total and volumetric efficiencies are also in a way fixed as they are calculated from manufacture data. Mass flows are calculated from the cooling capacity and the enthalpy difference over the evaporators. All the energies are calculated directly from the total and the volumetric efficiencies. The pressure levels are calculated from the evaporating and condensing temperatures.

Table 3 below presents the values of the reference model. These values are referring to the experimental values. All the real values for all these parameters are available on Appendix G.

For TR4 and TR5 the vapor return line at the intermediate stage has to be taken into account. A value of 0.04 and 0.08 kg/s of refrigerant are respectively chosen for TR4 and TR5 according to their experimental value.

Parameter	Value	Parameter	Value
Evaporating temperature (medium)	-10 °C	**Evaporating temperature (low)**	-30 °C
Inlet temperature of the compressor (low)	-5 °C	**Outlet temperature of the compressor (low)**	80 °C
Inlet temperature of the compressor (medium)	20 °C	**Outlet temperature of the compressor (medium)**	90 °C
Outlet temperature of the evaporator (low)	0 °C	**Outlet temperature of the evaporator (medium)**	20 °C
Volumetric efficiency	85 %	**Total efficiency**	60 %
Internal SH (low)	10 °C	**External SH (low)**	10 °C
Internal SH (medium)	10 °C	**External SH (medium)**	10 °C
Subcooling	0 °C		

Table 3: Input parameters of the reference model

The following plots show the influence of each parameter added one after the other. The system solutions themselves are first presented and then the mixture points, the compressor data, the pressure levels, the superheating and the subcooling will be added.

11.2. System solutions

The systems are not presented the same system solution as described on page **Error! Bookmark not defined.**. TR1 is a parallel system, TR2 and TR3 are booster systems and TR4 and TR5 are also booster systems but with a vapor return line at intermediate stage.

It can be seen on Figure 40 that the system solution is playing a major role even if all the supermarkets are sharing the same input parameters. TR1, TR2 and TR3 have exactly the same total COP but TR4 and TR5 have a higher total COP directly due to the return line.

The vapor return line permits to increase the efficiency on the low stage temperature without increasing the charge of refrigerant in the system. As the charge is lower the energy consumption in the compressors is reduced and thus the COP is improved. TR4 has a better COP than TR5 as the mass flow in the return line is lower.

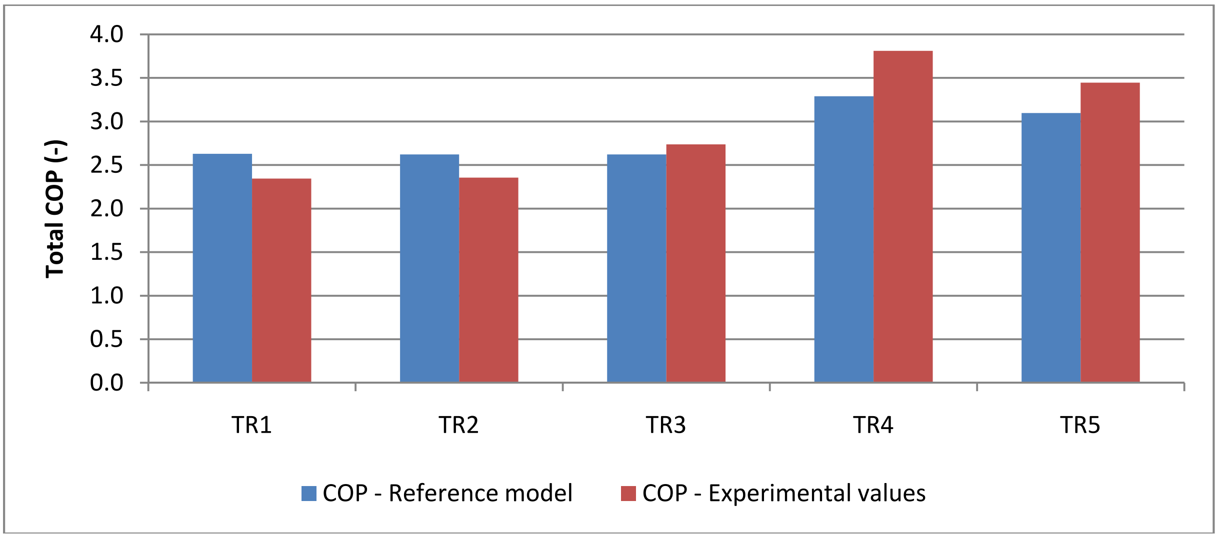

Figure 40: System solutions

11.3. Mixing points

The first plot was not taking into account the mixing point with the outlet temperature of the low stage compressor and the outlet temperature of the medium stage evaporator.

Figure 41 shows the influence of the mixing point on the total COP. It is indeed reduced for all the booster systems. The mixing point is increasing to an average value of 19 °C (instead of 10 °C) but the influence is lower for TR4 and TR5 due to the return line and the lower charge of refrigerant in the system.

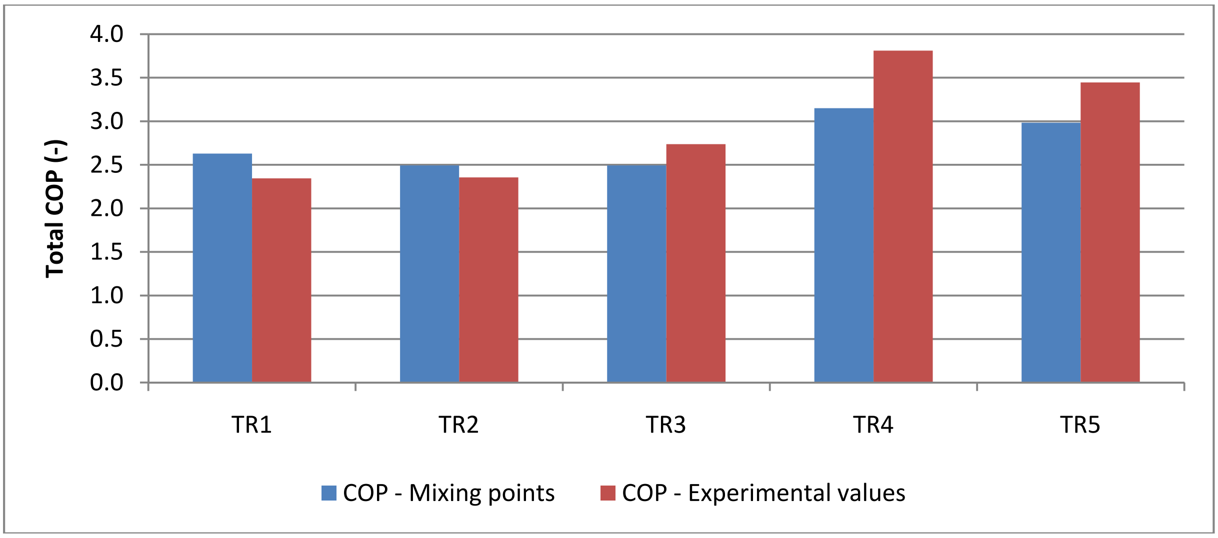

Figure 41: Mixing points influence on the system's total COP

11.4. Compressor efficienies

The real total and volumetric efficiency are now added to the previous result. For TR1, TR2 and TR3 theses efficiencies are better than the fixed values and therefore the total COP is increased. For TR2 and TR3 almost no changes can be observed as the real values are closed to the one of the reference model.

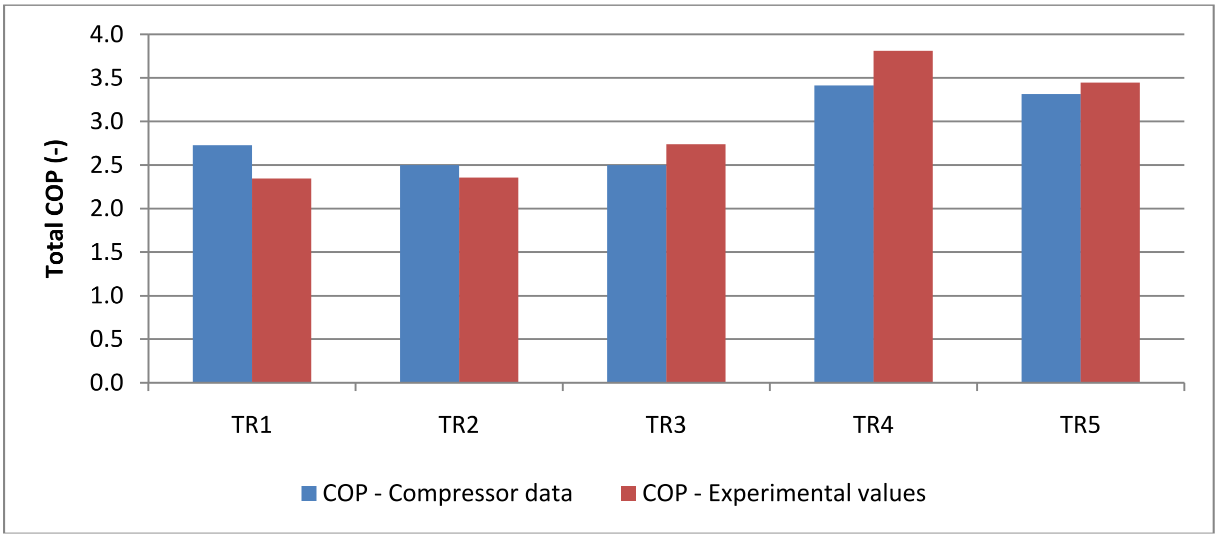

Figure 42: Compressor data

For this case Dorin compressors offers better volumetric efficiencies but on the other hand Bitzer compressors have a better total efficiency. No clear conclusions regarding the chose of the compressor can be established. It directly depends on the system solution. They have to be chosen carefully regarding the study already made on page 19.

11.5. Pressure levels

The pressure levels (high, medium and low) are added to the previous plot on Figure 43. TR4 and TR5 are working on lower pressure levels for both medium and low temperature stage which increase the total COP. TR1 and TR2 have higher pressure levels and it acts as a disadvantage for these systems. TR3 is working around the same pressure levels defined in the reference model.

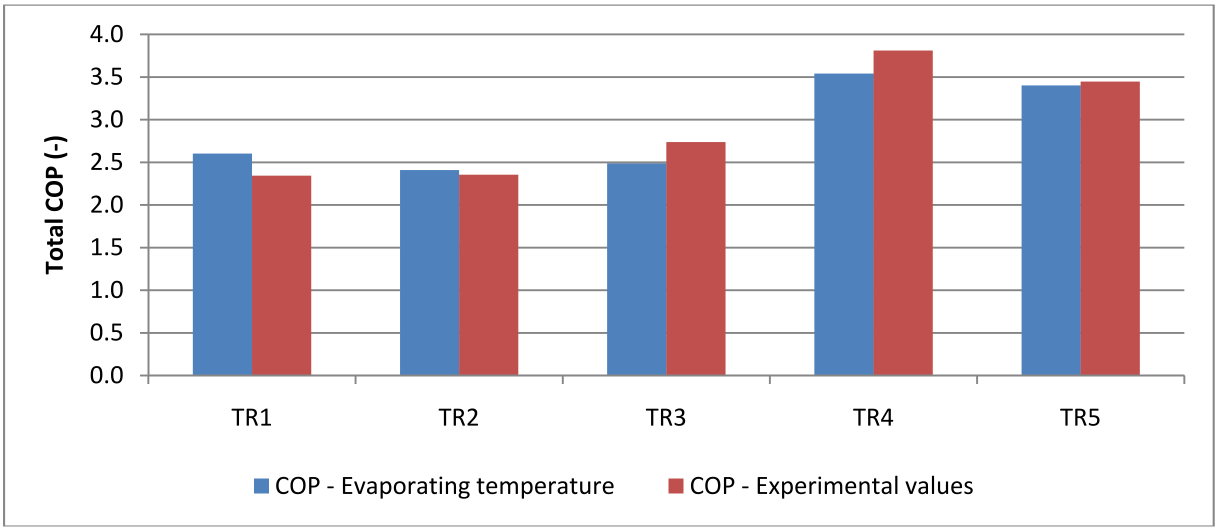

Figure 43: Pressure levels

11.6. Superheating and subcooling

Figure 44 presents the superheating and subcooling influence on the total COP. Only TR3 is sub cooled with a value around 4 °C. For the corresponding values of superheating they have already be described on page 30.

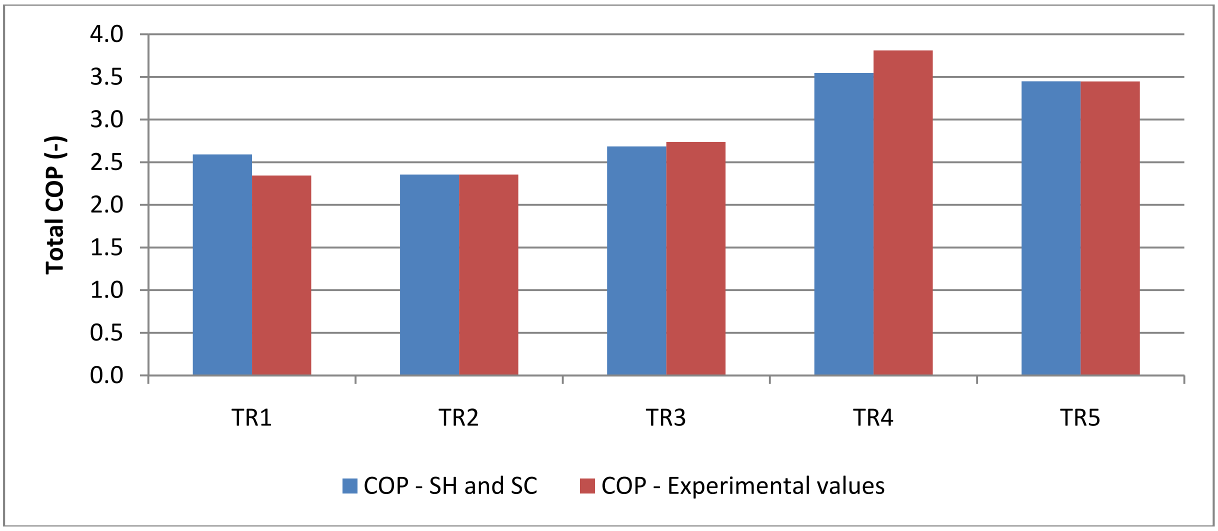

Figure 44: Superheating and subcooling

11.7. Global overview

The following plot presents all the simulations described before in order to see the impact on the total COP. The excel simulations is a good model to predict the behavior of a supermarket as the final result is rather close to the experimental values.

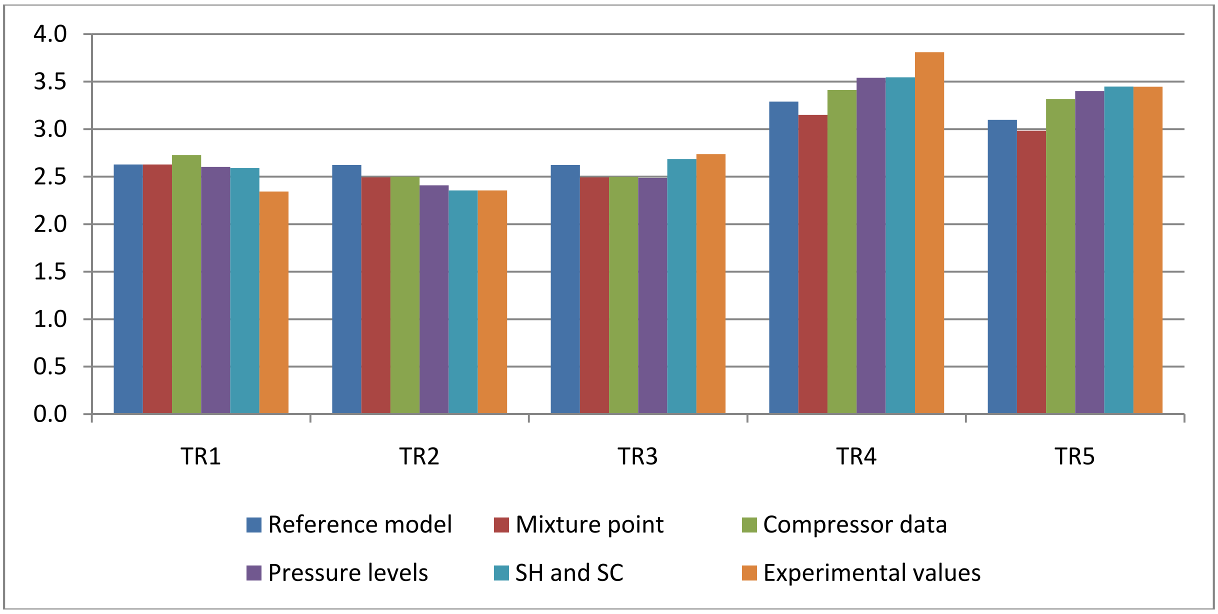

Figure 45: Influence of each parameters

The parameters are not influencing the total COP with the same strength. Therefore they can be classified by order of importance:

Parameter	Order of importance
Cooling system solution	1
Compressor characteristics	2
Pressure levels	3
SH and SC	4

Table 4: Classification of the different parameters

These results will be discussed in the next part.

12. DISCUSSION

There is a growing interest to use CO2 refrigeration systems in supermarkets by the refrigeration industry throughout Europe. Being a natural refrigerant is an advantage to the CO2 campaign as it is environmentally friendly and according to the legislation. However, to compete with traditional HFCs systems, CO2 systems have to be as efficient or even surpass the performances offered by the HFCs. This thesis shows the different performances obtained with CO2 based systems against one standard R-404A system.

Mainly due to its high pressure, CO2 presented a new challenge to the market players in refrigeration and air conditioning. Most of the components must be improved or completely re-developed however this process is being carried out by the main manufacturers. Compressors and most of the valves are now well adjusted to fit the use of CO2 but improvements on evaporators and gas coolers are still needed. One of the main problems of the CO2 technology is the use of evaporators poorly adapted to this fluid. Indeed, tubes diameters and the layout of the evaporator are the same as those used with traditional fluids. Tubes have to be smaller to provide a better utilization of the exchange surface, or even raise the temperature of evaporation, which has a direct effect on the energy efficiency.

This trend has been greatly confirmed in this report. On the main figure showing the total COP with a load ratio of 3 some CO2 based supermarkets are already working at higher efficiencies than a standard R-404A system which means that the manufactures have improved or developed specific components for CO2 systems.

The compressor study led on Dorin and Bitzer compressors shows that in this particularly case Dorin compressor offer better volumetric efficiencies and, on the other hand, Bitzer compressors offer better total efficiency (less losses). But, in general, CO2 compressors are working at higher efficiencies than standard designed R-404A compressors due to the different operating pressure ratio. Hence these results confirm that improvements have been carried out by the manufactures.

Concerning the cabinet study, the field measurements led on the CO2 supermarkets show that the output conditions such as the supply air temperature and the outlet temperature of the evaporator are close to each other for all the supermarkets; result which was expected due to the regulation limits to store food in safe conditions. The TR-x supermarkets can be classified into two groups in term of evaporating temperature, superheating and LMTD for both medium and low temperature stage:

- TR1&2: lower evaporating temperature and indeed a higher superheat value, higher temperature difference between the air and the refrigerant and high LMTD
- TR3&4&5: higher evaporating temperature and lower superheat value, lower temperature difference between the air and the refrigerant and lower LMTD

Hence, the TR1 and TR2 evaporators are poorly designed compared to the other supermarkets and it will reduce the efficiency of these systems. TR1 and TR2 are rather old CO2 systems compared to TR3, TR4 and TR5 however TR2 and TR3 are sharing the same layout (transcritical booster systems). It is then clearly shown that a great effort have been carried out by the manufactures over the years to improve the efficiency of the evaporators. TR4 and TR5 even surpass the other supermarket in term of evaporator design efficiency. They are working closer to the common superheat value of 12 K.

Finally, Excel simulations have been used in order to uniform the working conditions for each supermarket and thus allow fair comparison between all the systems. These simulations have shown the order of influence of each parameter such as the cooling system solution itself, the pressure levels, the degrees of superheat... on the total COP. The simulations were validated with field measurement. However the results for TR4 and TR5 have to be taken carefully as the collected data represent less than a year.

Most of the factors which are influencing the total COP depend directly on technical solutions such as the cooling system solution or the compressor‘s type used. If they are regulated the same way, booster systems coupled with a return line and run by Bitzer compressors will offer the best performances. However there is no clear difference between a standard booster system (TR2 and TR3) and a simpler parallel system (TR1)

Regulation processes such as the pressure levels or the superheat value have a lower impact in this case.

13. CONCLUSION

This thesis evaluated different field installations with different refrigeration system solutions. Excel simulations for each of the systems under investigation have been run. The systems under investigations have been compared based on the results from the field measurements and the computer simulation models. This work compares six different field installations; five CO2 based systems and one standard R-404A system.

A lot of improvements have been conducted throughout the years in CO2 refrigeration systems. The experimental and theoretical studies reported in this thesis clearly show this trend and prove that some new CO2 based system solutions investigated like a transcritical booster system with a return line can be efficient solutions to replace standard R404-A refrigeration systems. The main reasons are that nowadays the manufactures succeeded to improve the efficiency of the evaporator by sizing them correctly. New evaporators offer a higher condensing temperature with a lower superheat and indeed a better efficiency of the system.

A lot a work is still needed to improve the data treatment in term of quality and rapidity. The program used to filter the data for the raw measurement has to be improved by introducing polynomial regressions between two points instead of a fixed value. Furthermore specific correlations in order to calculate the volumetric and total efficiency curves have to be defined for smaller operating pressure range. Finally, the models used for the simulations have to be simplified by removing some unnecessary parameters (low influence) and the use of the average method used has to be confirmed with a deeper analysis.

Cottineau Vincent	Cholet
	2 Septembre 2011

APPENDIX A

CO2 Mollier diagram

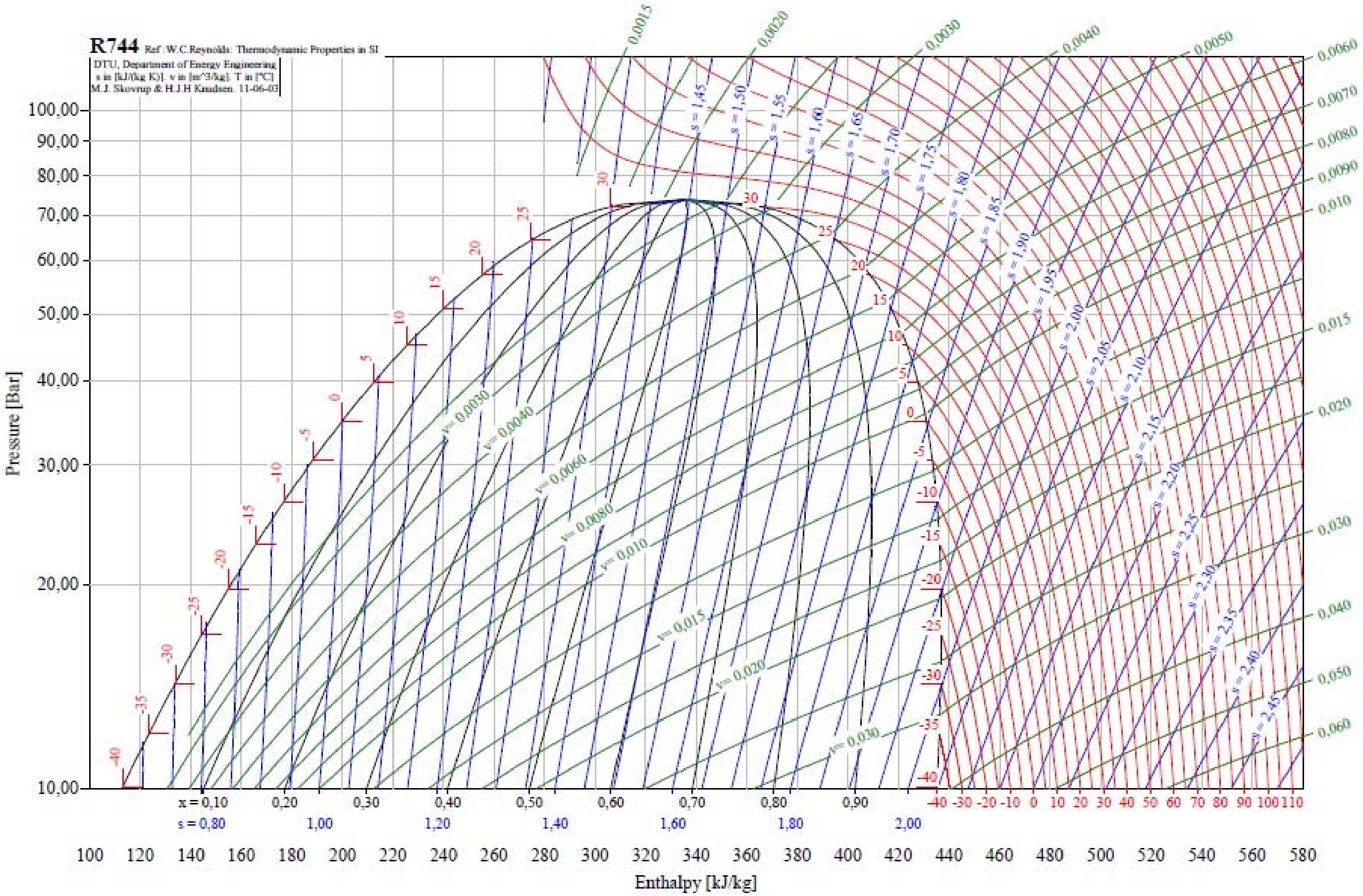

Figure 46: CO2 Mollier diagram (© CoolPack)

APPENDIX B

System layout

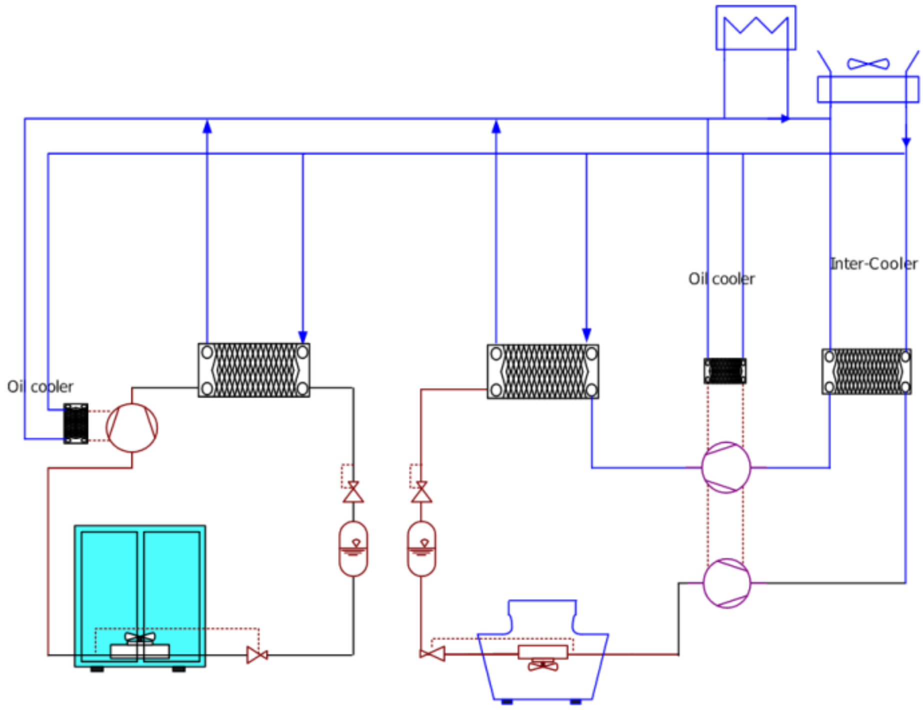

Figure 47: TR1 - System layout

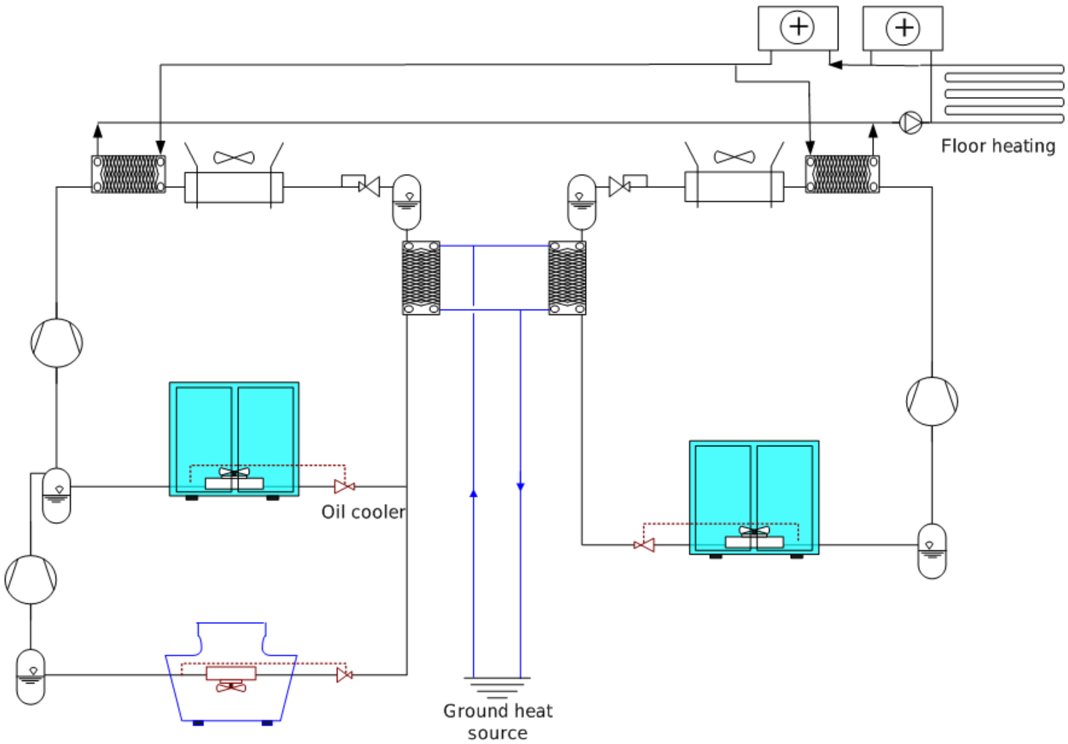

Figure 48: TR2 - System layout

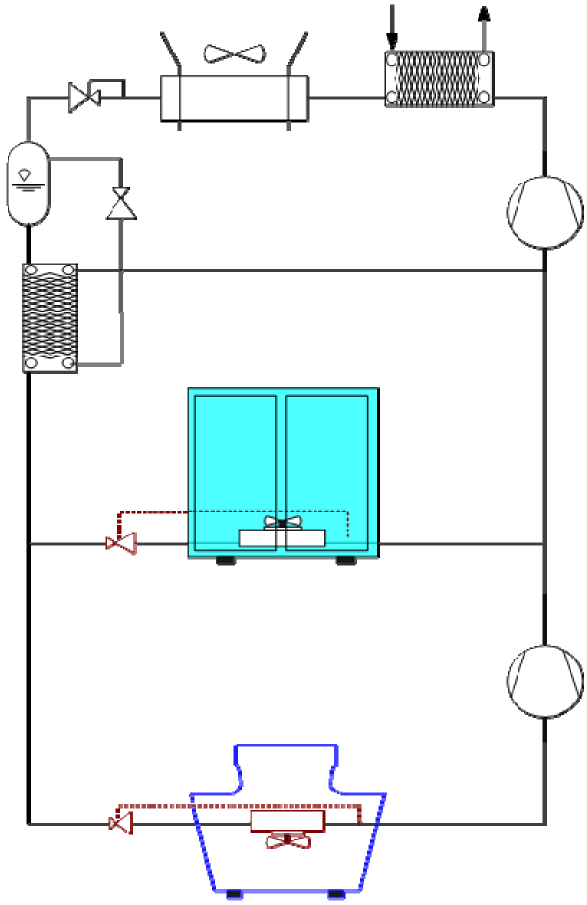

Figure 49: TR3 system layout

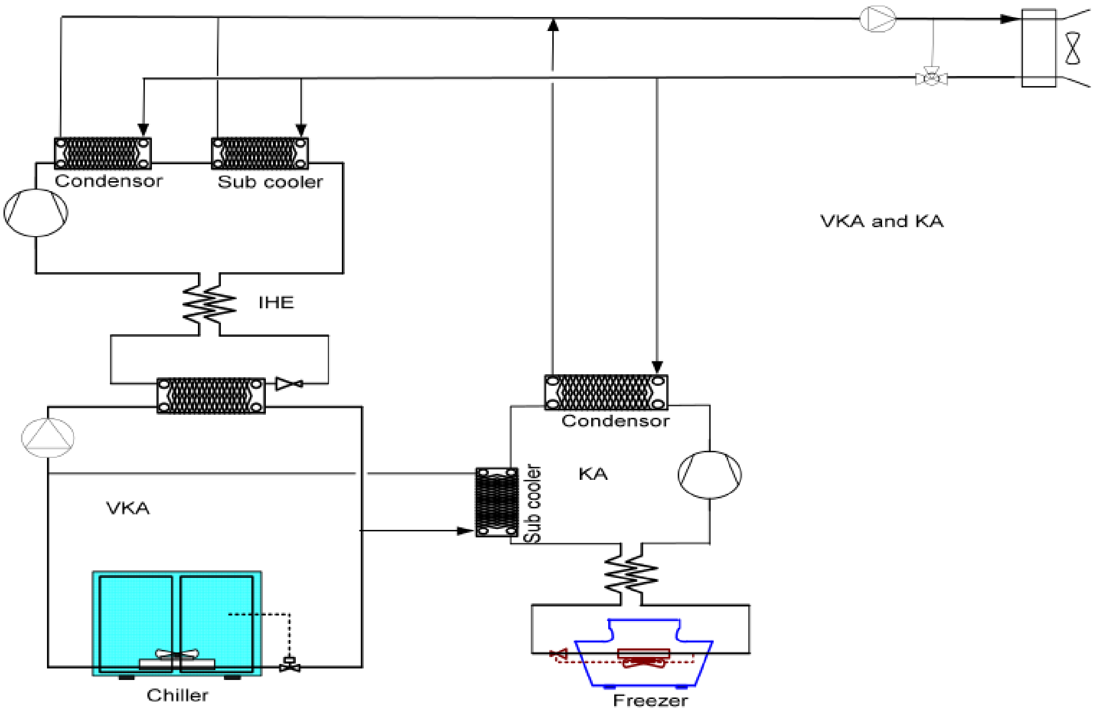

Figure 50: RS1 – System layout

APPENDIX C

The following demonstration illustrates how to calculate a total COP with a fixed load ratio of 3 (Frechelox, 2009).

Calculation

$$COP_{tot} = \frac{\dot{Q}_{o\_fr} + \dot{Q}_{o\_ch}}{\dot{E}_{comp\_fr} + \dot{E}_{comp\_ch}} = \frac{\frac{\dot{Q}_{o\_ch}}{LR} + \dot{Q}_{o\_ch}}{\frac{\dot{Q}_{o\_ch}}{LR \cdot COP_{fr}} + \frac{\dot{Q}_{o\_ch}}{COP_{ch}}} = \frac{\frac{1}{LR} + 1}{\frac{1}{LR \cdot COP_{fr}} + \frac{1}{COP_{ch}}}$$

Demonstration

$LR = LR\ real$

$LR_{corr} = LR\ corrected$

$$COP_{tot\_LR} = \frac{\frac{1}{LR_{corr}} + 1}{\frac{\dot{E}_{comp\_fr}}{LR_{corr} \cdot \dot{Q}_{fr}} + \frac{\dot{E}_{comp\_ch}}{\dot{Q}_{ch}}} = \frac{\frac{1 + LR_{corr}}{LR_{corr}}}{\frac{\dot{E}_{comp\_fr} \cdot \dot{Q}_{ch} + LR \cdot \dot{E}_{comp\_ch} \cdot \dot{Q}_{fr}}{LR_{corr} \cdot \dot{Q}_{fr} \cdot \dot{Q}_{ch}}}$$

$$= \frac{\dot{Q}_{fr} \cdot \dot{Q}_{ch} \cdot (1 + LR_{corr})}{\dot{E}_{comp\_fr} \cdot \dot{Q}_{ch} + LR_{corr} \cdot \dot{E}_{comp\_ch} \cdot \dot{Q}_{fr}} = \frac{\dot{Q}_{fr} \cdot \dot{Q}_{ch} \cdot (1 + LR_{corr})}{\dot{E}_{comp\_fr} \cdot \dot{Q}_{fr} \cdot LR + LR_{corr} \cdot \dot{E}_{comp\_ch} \cdot \dot{Q}_{fr}}$$

$$= \frac{\dot{Q}_{ch} \cdot (1 + LR_{corr})}{\dot{E}_{comp\_fr} \cdot LR + LR_{corr} \cdot \dot{E}_{comp\_ch}} = \frac{\dot{Q}_{ch} \cdot \left(\frac{1 + LR_{corr}}{LR_{corr}} \right)}{\frac{1}{LR_{corr}} \dot{E}_{comp\_fr} \cdot \frac{\dot{Q}_{o\_ch}}{\dot{Q}_{o\_fr}} + \cdot \dot{E}_{comp\_ch}}$$

APPENDIX D

Compressor characteristics

Name	Compressor	Compressor type	Swept volume (m^3/h)	Volumetric efficiency (%)	Total efficiency (%)
TR1	4 Dorin – CO2	KA1&2: TCS373-D	12,6	$\eta_v = -9{,}45\ PR + 107{,}1$	$\eta_v = -2{,}79\ PR^2 + 19{,}51\ PR + 33{,}0$
	2 Dorin – CO2	FA1&2: TCDH372 B-D	12,6	$\eta_v = -1{,}15\ PR + 93{,}9$	$\eta_v = 0{,}08\ PR^2 - 1{,}14\ PR + 65{,}7$
TR2	2 Dorin – CO2	KA1&2: TCS373-D	12,6	$\eta_v = -9{,}45\ PR + 107{,}1$	$\eta_v = -2{,}79\ PR^2 + 19{,}51\ PR + 33{,}0$
	3 Dorin – CO2	FA1&2: SCS362-D	10,7	$\eta_v = -4{,}84\ PR + 95{,}1$	$\eta_v = 0{,}22\ PR^2 - 1{,}02\ PR + 54{,}3$
	4 Dorin – CO2	KA3: TCS373-D	12,6	$\eta_v = -9{,}45\ PR + 107{,}1$	$\eta_v = -2{,}79\ PR^2 + 19{,}51\ PR + 33{,}0$
TR3	4 Dorin – CO2	KA1&2: TCS373-D	12,6	$\eta_v = -9{,}45\ PR + 107{,}1$	$\eta_v = -2{,}79\ PR^2 + 19{,}51\ PR + 33{,}0$
	1 Dorin – CO2	FA1: SCS340-D	7,0	$\eta_v = -4{,}86\ PR + 95{,}2$	$\eta_v = 0{,}21\ PR^2 - 0{,}95\ PR + 54{,}2$
	1 Dorin – CO2	FA1: SCS362-D	10,7	$\eta_v = -4{,}84\ PR + 95{,}1$	$\eta_v = 0{,}22\ PR^2 - 1{,}02\ PR + 54{,}3$
	2 Dorin – CO2	FA2 : SCS340-D	7,0	$\eta_v = -4{,}86\ PR + 95{,}2$	$\eta_v = 0{,}21\ PR^2 - 0{,}95\ PR + 54{,}2$
TR4	2 Bitzer – CO2	KA1&2: 4MTC-10K-l	6,5	$\eta_v = -9{,}53\ PR + 102{,}3$	$\eta_v = -0{,}86\ PR^2 + 1{,}23\ PR + 68{,}4$
	2 Bitzer – CO2	FA1&2: 2MHC-05K-B	1,6	$\eta_v = -14{,}56\ PR + 112{,}8$	$\eta_v = -7{,}29\ PR^2 + 29{,}37\ PR + 27{,}0$
TR5	5 Bitzer – CO2	KA1&2: 4HTC-20KI	12,0	$\eta_v = -9{,}56\ PR + 103{,}6$	$\eta_v = -0{,}65\ PR^2 - 0{,}31\ PR + 72{,}7$
	4 Bitzer – CO2	FA1&2: 2HHC-2K	4,34	$\eta_v = -14{,}52\ PR + 112{,}7$	$\eta_v = -6{,}95\ PR^2 + 27{,}78\ PR + 28{,}8$
RS1	2 Bitzer – R404A	VKA1: 4J-22.2Y	63,5	$\eta_v = 0{,}21\ PR^2 - 7{,}00\ PR + 103{,}1$	$\eta_v = -0{,}02\ PR^2 + 0{,}61\ PR + 62{,}7$
	2 Bitzer – R404A	KA1: 4VCS-6.2	34,7	$\eta_v = 0{,}12\ PR^2 - 5{,}36\ PR + 96{,}4$	$\eta_v = -0{,}04\ PR^2 + 0{,}10\ PR + 60{,}2$

APPENDIX E

Medium stage temperature

MT-stage								
Supermarket	**To**	**Return air**	**Supply air**	**Evap inlet**	**Evap outlet**	**Super h**	**Defrost**	**LMDT**
TR1	-6,3	4,5	0,2	-6,3	3,3	9,6	0,09	8,5
TR2	-9,3	3,7	0,4	-9,3	3,6	12,9	0,09	11,3
TR3	-5,8	1,8	1,1	-5,8	3,8	9,6	0,13	7,2
TR4	-5,4	2,4	-1,3	-2,9	0,5	6,0	0,13	5,8
TR5	-5,2	1,9	1,2	-3,5	2,6	7,8	0,13	6,8
Average	-6,4	2,8	0,3	-5,5	2,8	9,2	0,11	7,9

Low stage temperature

LT-stage								
Supermarket	**To**	**Return air**	**Supply air**	**Evap inlet**	**Evap outlet**	**Super h**	**Defrost**	**LMDT**
TR1	-33,7	-19,2	-26,6	-33,7	-20,6	13,2	0,01	10,4
TR2	-33,1	-22,4	-26,9	-33,1	-22,0	11,1	0,02	8,2
TR3	-29,6	-20,7	-26,1	-29,6	-21,9	7,8	0,01	5,8
TR4	-30,0	-20,7	-27,1	-28,2	-23,9	6,3	0,03	5,5
TR5	-29,4	-19,9	-26,2	-28,2	-21,6	7,8	0,01	5,8
Average	-31,2	-20,6	-26,6	-30,6	-22,0	9,2	0,02	7,1

APPENDIX F

Values of superheating

TR1			
	Internal superheating	External superheating	Total
KA1	8,7	8,5	17,2
KA2	10,0	5,5	15,4
FA1	13,2	31,2	44,4
FA2	13,2	31,2	44,4

TR3			
	Internal superheating	External superheating	Total
KA1	9,0	15,0	24,0
KA2	10,4	14,9	25,2
FA1	7,7	20,6	28,3
FA2	7,7	20,6	28,3

TR2			
	Internal superheating	External superheating	Total
KA1	13,3	18,4	31,7
KA2	13,3	18,4	31,7
KA3	12,8	-2,7	10,1
FA1	11,1	15,8	26,9
FA2	11,1	15,8	26,9

TR4			
	Internal superheating	External superheating	Total
KA1&2	6,0	8,1	14,1
FA1&2	6,3	12,7	19,0

TR5			
	Internal superheating	External superheating	Total
KA1&2	7,8	9,4	17,2
FA1&2	7,8	14,2	22,0

APPENDIX G

Unit	T evap KA	T evap FA	T inlet comp FA	T outlet comp FA	T inlet comp KAFA	T outlet comp KAFA
TR1 KA	-10,4				11,1	89,5
TR1 FA		-34,0	3,7	89,6		
TR2 KAFA	-9,5	-33,1	-2,9	80,7	31,1	112,4
TR3 KAFA	-7,3	-33,2	-6,7	83,6	23,7	96,5
TR4 KAFA	-6,9	-31,2	-11,3	66,2	19,7	85,8
TR5 KAFA	-7,1	-32,1	-12,7	67,5	17,6	82,6

Unit	T outlet evap KA	T outlet evap FA	Total Efficency KAFA	Total Efficency FA	Volumetric Efficiency KAFA	Volumetric Efficiency FA
TR1 KA	3,6		62,3		86,4	
TR1 FA		-19,3		62,1		91,5
TR2 KAFA	3,9	-19,9	61,9	53,1	87,0	85,0
TR3 KAFA	-0,5	-19,9	60,9	53,1	88,2	84,3
TR4 KAFA	-0,6	-24,2	67,5	56,5	83,4	82,1
TR5 KAFA	-0,4	-22,4	69,5	56,4	84,5	81,3

Unit	Internal SH KA	Internal SH FA	External SH KA	External SH FA	Subcooling
TR1 KA	14,0		7,5		0,0
TR1 FA		14,7		23,0	0,0
TR2 KAFA	13,4	13,2	12,1	17,0	0,0
TR3 KAFA	6,8	13,3	6,8	13,3	4,0
TR4 KAFA	6,3	7,0	8,4	12,9	0,0
TR5 KAFA	6,7	9,7	6,7	9,7	0,0

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