KTH Industrial Engineering and Management

Analysis of simultaneous cooling and heating in supermarket refrigeration systems

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Master of Science Thesis

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KTH Industrial Engineering and Management	**Analysis of simultaneous cooling and heating in supermarket refrigeration systems**	
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Abstract

In this master thesis project, conventional supermarket refrigeration systems using R404A are compared with refrigeration system solutions using natural refrigerants such as carbon dioxide and ammonia. This systems analysis considers the behavior of those systems in floating condensing and heat recovery mode. System heating and cooling COP have been calculated by using computer simulation with the calculation software EES (Engineering Equation Solver). The impact of important parameters such as sub-cooling, external superheating and compressor discharge was also determined through the computer models.

The estimation of the system annual energy consumptions shows that systems using natural refrigerant can compete with systems using artificial refrigerant by using heat recovery system such as heat pump cascade, heat pump cascade for sub-cooling, fixed pressure system and de-superheater. If the indirect emission of systems using natural refrigerant and artificial refrigerant is approximately similar, the direct emission for carbon dioxide systems and ammonia systems can be estimated to be 10000 times less important than R404A systems.

Multi-unit refrigeration systems have also been studied in this project; it appears that in theory COP improvement of 10% is possible if the condensing temperature of each unit is controlled adequately.

Acknowlegments

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Finally, I want to thank the entire applied thermodynamics department for my reception in the laboratory.

Nomenclature

COP: Coefficient Of Performance

MT: Medium Temperature

LT: Low Temperature

HRR: Heat Recovery Ratio

DSH: De-SuperHeater

HP: Heat Pump

FHP: Fixed Head Pressure

HPSC: Heat Pump Cascade for Sub-Cooling

R404A conv SW: R404A Swedish conventional system

R404A conv US: R404A U.S conventional system

TR1: Transcritical CO_2 system 1

TR2 transcritilcal CO_2 system 2

GHG: GreenHouse Gas

COP tot: total COP

Sub: sub-cooling

FC: Floating Condensing

T cond: condensing temperature

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1. Introduction

1.1 Background

During the 20th century artificial refrigerants such as HCFC or CFC were commonly used due to their reliability in term of performance and safety, whereas natural refrigerant such as CO_2 tended to be neglected. Nevertheless in 1996, the use of artificial refrigerants became forbidden by the Montreal protocol in developed countries since artificial refrigerants turned out to be hugely harmful for the environment. Actually the GWP (global warming impact) of artificial refrigerants is, for most of them, thousand times higher than that of CO_2.
Therefore natural refrigerants have made a come-back on the refrigeration market. The current technology allows refrigeration systems using natural refrigerant to be much more efficient than in the past.
The topical issue in supermarket refrigeration is thus to ensure an equivalent system reliability by using natural refrigerants instead of HCFC or CFC refrigerants. CO_2 which has been commonly used in supermarket before the appearance of artificial refrigerants and NH_3 which is currently widely used as refrigerant in industry could be good alternatives to artificial refrigerant.

1.2 Refrigeration in supermarket

In supermarket, refrigeration is the major consumers of energy. As figure 1.1 shows in Swedish supermarkets, refrigeration represents 47% of the total energy usage. For this reason improving the energy efficiency of supermarket refrigeration systems would significantly reduce the energy consumption of supermarket.

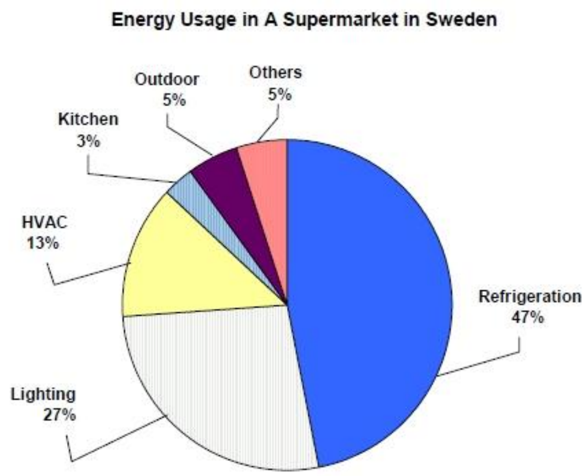

Figure 1.1: breakdown of energy usage in a supermarket in Sweden [1]

In addition, leakage of refrigerant is quite important in supermarket refrigeration: According to figure 1.2, leakage of refrigerant is around 12% of the total refrigerant charge over one year. The direct emission from supermarket refrigeration system is thus not negligible regarding the usual high GWP of artificial refrigerant. In Swedish supermarket, indirect R404A refrigeration systems are traditionally used: In the medium temperature unit, R404A is used for the compression cycle setting in the refrigeration machine room, and brine in the pipes which connect the food storage cabins to the refrigeration machine room.

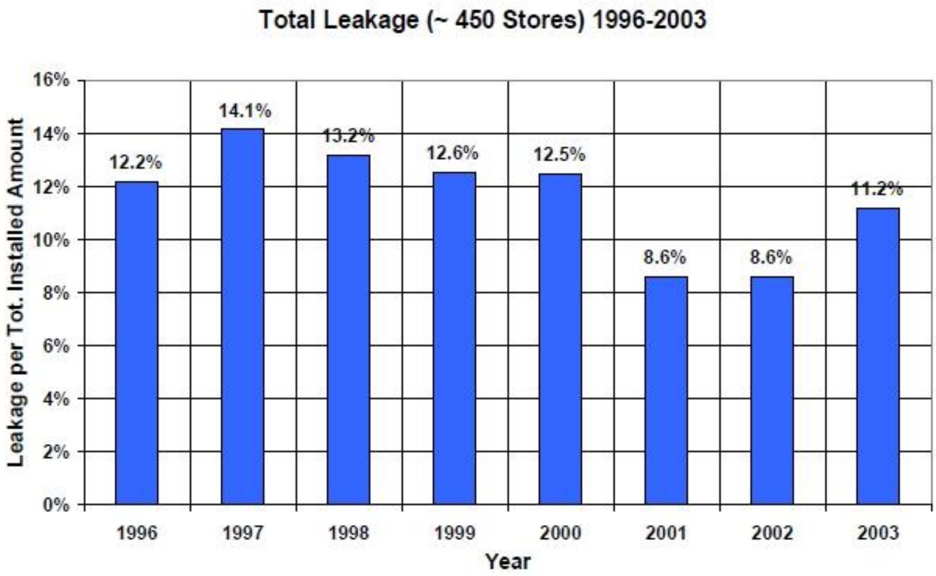

Figure 1.2: Refrigerant leakage in Swedish supermarket [1]

1.3 Project objectives

This project investigates the refrigeration and heat recovery performances of the new system solutions which are mainly based on natural refrigerants in supermarket refrigeration. The new system solutions with natural refrigerants are compared with traditional refrigeration systems using artificial refrigerants. Different heat recovery solutions have also been investigated and compared.

The project system analysis focuses on refrigerant properties, system solution layout, system solution cooling and heating performances, and refrigeration system key parameters such as sub-cooling, external superheat, condensing temperature and discharge pressure.

This project includes also an estimation of the total annual emission and energy consumption of each system solution studied.

The main purpose of this project is to study the behaviour of different refrigeration system solutions in heating recovery mode.

1.4 Methodology

The first step of this master's thesis work was to survey the conventional and the new potential refrigeration and heat recovery systems and investigate their compatibility.

In a second time, from existing EES (engineering equation solver) system models and EES system models coded for this work, the efficiencies and the influence of refrigeration key parameters have been estimated for each system solution and compared with others.

Afterward the annual energy consumption of all systems has been calculated by taking in account their cooling and heating performances and the outdoor conditions in Sweden.

Finally the results obtained by simulations have been compared with experimental data and validated for the supermarket CO2 system with heat pump cascade, TR1 and the CO_2 system with de-superheater, TR2. From these results, correlations describing the system solutions have been established. By integrating these correlations to an optimization algorithm, the influence of the condensing temperature in each system unit on the total system performance have been evaluated.

2. Refrigeration and heat recovery system

Refrigeration systems in supermarket are closed systems generally constituted by chilled and frozen units (figure 2.1). Products in chiller and freezer are respectively stored at +3 and -18°C which approximately correspond to evaporation temperatures at -10°C for the medium temperature unit and -35°C for the low temperature unit.

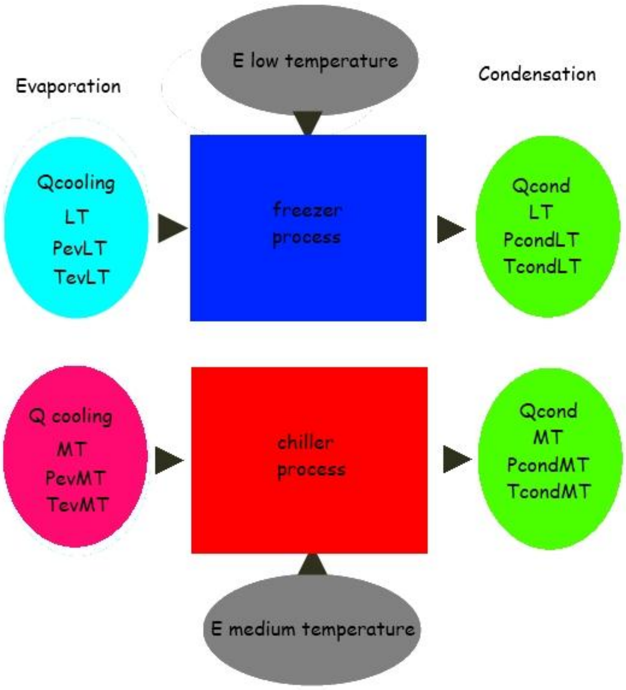

Figure 2.1: schematic of refrigeration system in supermarket

Refrigeration system process can be considered as two similar processes for the chiller and for the freezer. For both processes the heat is taken from the food storage cabin by evaporation, heat is released by the condenser to the ambient. Work is also necessary in order to run the pump, compressors, fans (figure 2.1). Thus the energy balance for both chiller and freezer is:

$$Q_{cond}=Q_{cooling}+E \tag{2-1}$$

Instead of releasing heat to the ambient heat can be recovered from the condensation. When the refrigerant enters into the condenser since the compression, the refrigerant has a high temperature and has to be cooled in order to reach the necessary condensation temperature. This heat with a high energy content can be recovered and provide to the HVAC system. In the same way if the temperature condensation is sufficient, the latent heat from the condensation phase can be released to the HVAC system. In addition, condensers are often design for several refrigerant mass flow and heat content. Consequently the refrigerant is sub-cooled after the condensation is totally achieved. Thus heat can be recovered by adding heat recovery system between the compressor and condenser, directly connected to the condenser or after this one condenser by using a sub-cooler connected to a heat pump.

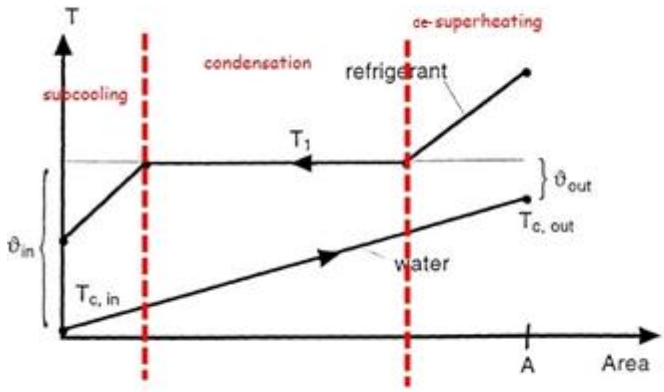

Figure 2.2: temperature profile througout a condenser

Refrigeration systems can be characterized by several indicators. The cooling COP (Coefficient Of Performance) which is the ratio between the cooling capacity and the electrical power used, it values the effectiveness of refrigeration system from an energy point of view. The heating COP which is the ratio between the heat recovered and the energy difference between heat recovery and floating condensing mode; in the case of refrigeration system, the heating COP values the heating performance of the heat recovery system associated. The HRR (heat recovery ratio) defined as the ratio between the quantity of heat recovered and the cooling capacity; it characterizes the system ability to recover heat. Finally the energy consumption monthly or annually links the system to notions of time, cost and indirect gas emission. The combination of these parameters gives a refrigeration system overview which helps in the analysis and choice of refrigeration systems.

2.1 General overview of refrigeration systems

- *Direct system*

Direct systems are traditionally used in supermarket refrigeration. As figure 2.1.1 shows, these systems are simple with just one thermodynamic refrigeration cycle. The heat is extracted from the store food cabins and provides through pipes to the machine room where the refrigerant is compressed to the required condensation temperature. Since the distance between the machine room and the store food cabins, the pipe connecting the both unit are long. Consequently the refrigerant charge is important [1].

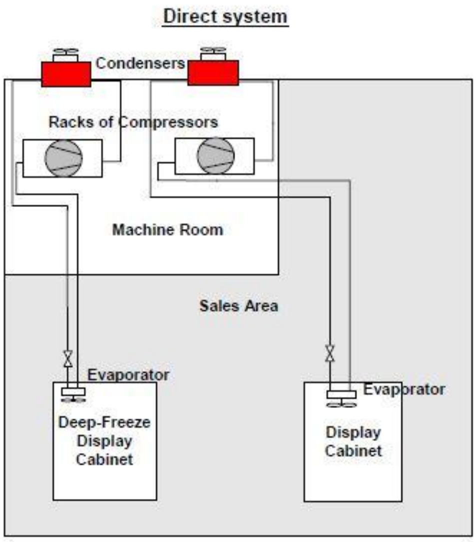

Figure 2.1.1: schematic of a direct refrigeration system [1].

- *Indirect system*

In order to decrease the refrigerant charge, indirect systems are used. As figure 2.1.2 sets, in those systems, there is a secondary loop with a less environment armful refrigerant. This refrigerant is utilized as cooling fluid for the condensation of main loop refrigerant. The coefficient of performance is less than the direct system and extra pumps are needed in the secondary loop, but the charge is widely reduced [1].

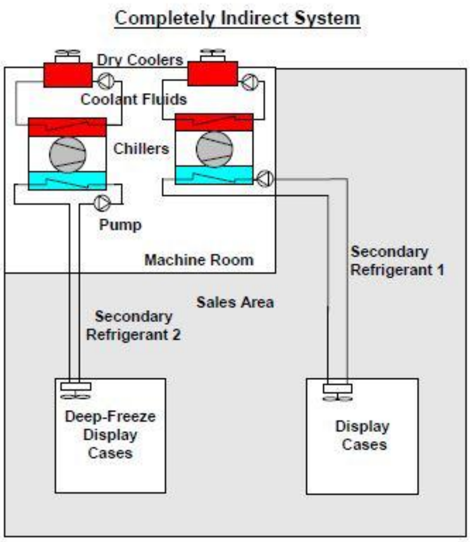

Figure 2.1.2: schematic of an indirect refrigeration system [1]

- *Cascade system*

The cascade refrigeration system is described in figure 2.1.3. In the low temperature unit, the pressure ratio between condensation pressure and evaporation pressure is large. Thus the COP is often low for low temperature unit. In the cascade system, the secondary refrigerant receives enough heat from the main loop condenser to reach the medium temperature. Afterwards the second refrigerant is compressed to the required low temperature evaporation pressure. The energy use in the low temperature is reduced but low and medium temperature units become dependent on each other [1].

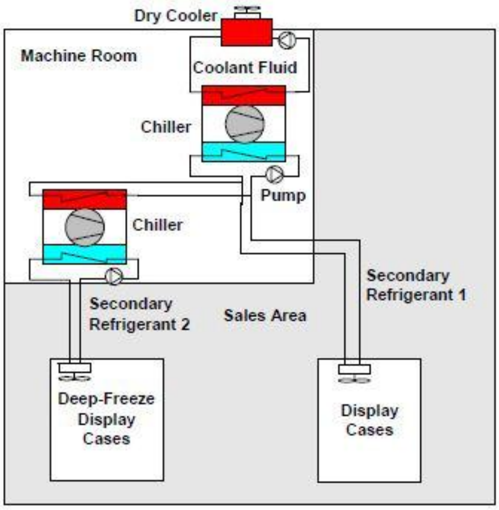

Figure 2.1.3: schematic of a cascade system [1]

2.2 Heat recovery systems in supermarket

Most of supermarkets in Sweden used heat recovery system in order to assist the sales area heating during winter time. Nevertheless to provide heat with sufficient energy content, high condensation temperatures are often required which decreases the system coefficient of performance. ([1], [3]) Recovery systems are added to the heat rejection side, which is basically constituted by compressor, condenser and an expander. Several types of heat recovery systems exist.

- *Fixed head pressure*

In the fixed head pressure heat recovery system (figure 2.2.1), the heat rejected by the condenser is provides to HVAC system through a secondary cooling loop. When heating is not needed, the refrigeration system runs in floating condensing, whereas when cooling is needed, the condensation pressure is increased to a fixed pressure enough high to the required temperature level to the HVAC. For system without heat pump included, HVAC requires an inlet temperature of 45°C [2].

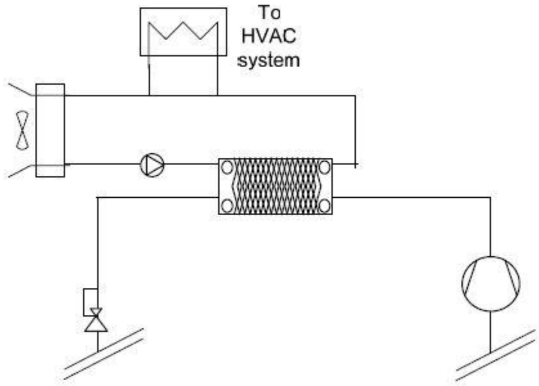

Figure 2.2.1: schematic of a fixed head pressure system [2]

- *De-superheater*

After compression, the refrigerant is superheated. The discharge temperatures are about 50°C, 70°C and 30°C for respectively CO_2, NH_3 and R404A(condensation temperature 25°C, evaporation temperature 5°C). By including a de-superheater before the condenser (or gas cooler) as in figure 2.2.2, the superheat can be recovered and rejected to the HVAC. The condensing pressure is regulated to provide the heat needed for the supermarket. The addition of de-superheater is more relevant for refrigeration system running with a refrigerant which has high discharge temperature such as NH_3 [2].

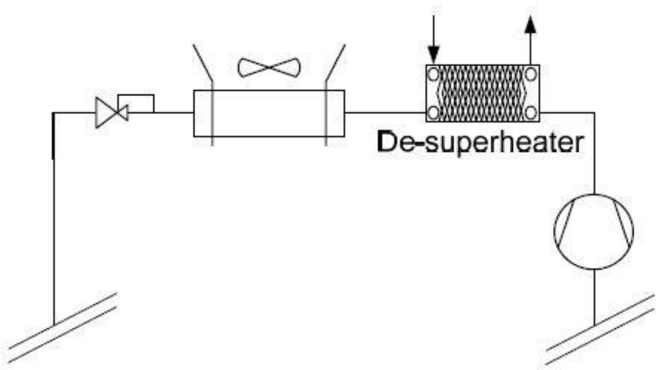

Figure 2.2.2: schematic of a de-superheater system [2]

- *Heat pump cascade*

In this heat recovery system the heat is directly extracted from the condenser and provides to a heat pump connected to the HVAC system. By using a heat pump (figure 2.2.3) the system is able to operate at lower condensing pressure and temperature than a fixed head pressure system. For instance for a refrigeration system the condensing temperature needed for fixed head pressure system is generally about 40°C whereas for heat pump system the condensing temperature is about 20°C. [2]

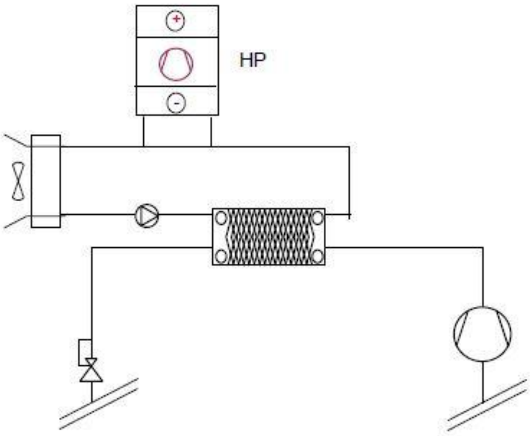

Figure 2.2.3: schematic of a heat pump cascade system [2]

- *Heat pump cascade for sub-cooling*

In this system described in figure 2.2.4, the heat pump is connected to a sub-cooler to recover heat but also to increase the system efficiency. Due to the sub-cooling, the enthalpy difference of vaporization becomes larger which according to the energy balance, decreases the mass flow. Thus compression needs less energy to be performed. [2]

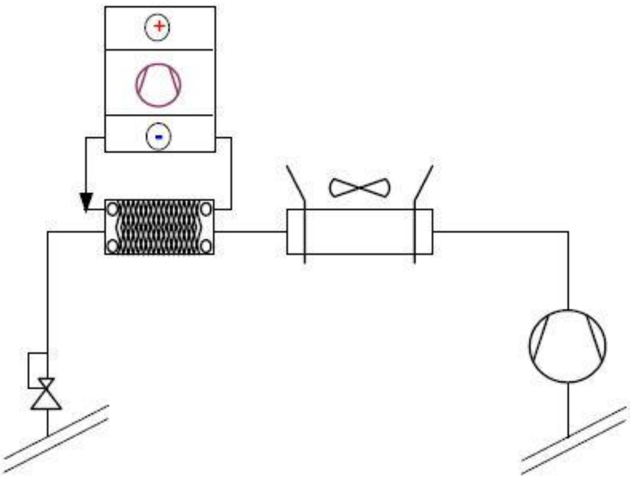

Figure 2.2.4: schematic of a heat pump cascade for sub-cooling system [2]

2.3 Refrigerants used in Supermarket

2.3.1 Refrigerant overview

Artificial refrigerants have been used during the all 20th century, but due to their impact on the environment and their role in the global warming process, the use of artificial refrigerant has to decrease. Nevertheless they are very reliable concerning the energy efficiency and the safety. Since R404 A is the most traditional artificial refrigerant used in supermarket refrigeration, this study mainly focuses on R404A.

CO_2 refrigerant were used during the 19th century, but the fact that high condensation pressure are need for such refrigeration system, favored the use of artificial system. Nowadays they are used again and the technology progress in refrigeration components enables the conception of efficient refrigeration system.

Ammonia is mainly used as refrigerant in industry for cooling process. Since its high latent heat, the refrigerating ability of ammonia is more important than other refrigerants. Ammonia is flammable and toxic but its strong smell allow operators to notice leakage occurrences. Nonetheless this strong smell can be a problem in supermarket refrigeration if the smell is released to the sales area. [4]

2.3.2 Refrigerant properties

The choice of a refrigerant is determined by its physical properties. Ammonia density is lower than R404A and CO_2 which implies an higher compressor swept volume [4], but ammonia has a higher refrigerating effect per mass flow unit since its higher heat latent of vaporization. Nevertheless, ammonia used to have high charge according to the overall space occupied. Ammonia charge is often above the maximum charge authorized by the legislation. Therefore ammonia refrigeration systems are more used in large industry. [4]

In the opposite, CO_2 has a high volumetric refrigeration capacity, thus CO_2 refrigeration systems have smaller compressors than other refrigerants for an identical refrigeration capacity [2]. CO2 is also characterized by a low critical temperature corresponding to a high critical pressure. The low critical temperature allows CO_2 refrigeration systems to operate above the critical point. Consequently, the condenser heat capacity is more important, so more heat can be recovered [2]. Nonetheless CO_2 systems runs at higher pressure, the large difference between evaporation and condensation pressure leads to a much lower COP than system using other refrigerants.

In a practical point of view, R404A is reliable refrigerant, installation size and efficiency are both satisfactory since rather good physical properties with no marginal value for none of them. Unfortunately in an environmental point of view R404A is an inadequate refrigerant. Its GWP (Global Warming Potential) is 3800 which means that 1 kg of R404A corresponds to an equivalent of 3800 kg of CO_2.

refrigerant	R404 A	CO_2	Ammonia
Molar weight (g/mol)	97.60	44.01	17.03
Density (kg/m^3)	4.4	2.0	0.76
Volumetric refrigeration capacity (0°C kj/m^3 sam)	5000	22500	5000
Critical temperature(°C)	72.1	31.2	132.4
Critical pressure (bar)	37.4	73.8	118
Specific heat capacity (kJ/kg.K) (1.013 0°C)	0.833	0.828	2.178
Latent heat of vaporization (kj/kg 1.013 bar boiling point)	250	578.08	1371.2
GWP	3800	1	>1
COP	4.21	2.96	4.84

Table 2.3.2: refrigerant physical properties ([4], [5])

2.3.3 Refrigerants and heat recovery systems

- *system description*

Refrigeration systems should have a good COP but also good heat recovery capacity as well. The refrigerant is really determining for both of these aspects. In order to compare the refrigerants NH_3, CO_2 and R404A in a relevant way, a simple system model is used. Thus the system differences depend only on the refrigerants.

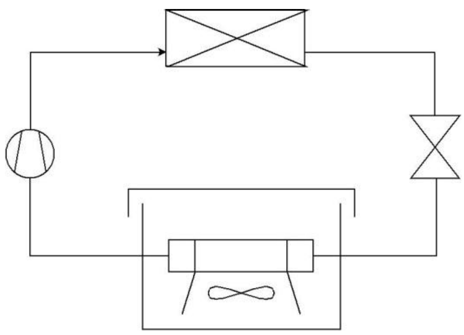

Figure 2.3.3.1: simple cycle of a refrigeration system

The model exposed in figure 2.3.3.1, is constituted by an evaporator, a compressor, an expander and a condenser/gas cooler. Low and medium temperature units are considered and studied separately. The following parameters are used:

	Low temperature unit	Medium temperature unit
Evaporation temperature (C)	-35	-10
Refrigerating heat capacity (kW)	35	100
Internal superheat (C)	5	5
External superheat (C)	15	15

Table 2.3.3.1: assumptions of the refrigerant property analysis

- *system analysis*

In order to satisfy the supermarket heating demand, the heat recovery system has to provide a certain amount of heat with the adequate temperature to the HVAC. Usually a condensing temperature around 30°C is needed for system without heat pump and around 20°C for system with one ([2], [14]). As figure 2.3.3.2 shows that when condensing temperature increases the COP decreases as well and the system using NH_3 has higher COP than the other systems. For a condensing temperature of 30°C and an evaporation temperature of -10°C, the NH3 system COP is about 4,0 , the CO_2 and R404A system COP are respectively about 2,7 and 2,1. When the condensing temperature is lower than 30°C, CO_2 COP is higher than R404A COP; the opposite is observed after a condensing /gas cooler outlet temperature of 30°C. It can be noticed that for CO_2, for temperatures above 28°C in the condenser, the CO_2 state is gas, therefore there is no condensing temperature anymore but the gas cooler exit is used as replacement.

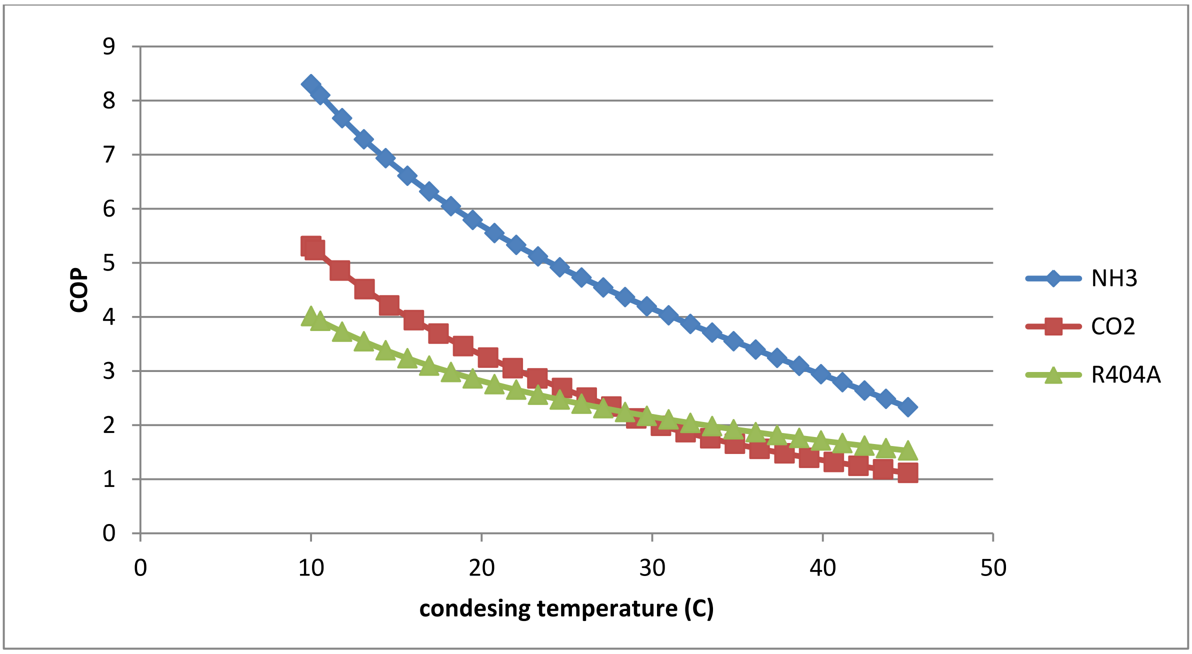

Figure 2.3.3.2: medium temperature COP profiles for CO_2, NH_3 and R404A

The purpose of refrigeration system combined with heat recovery system is to provide the required heating demand to the supermarket and the required cooling capacity in to the food storage cabins. Therefore, the ratio between heat recovered and cooling capacity is used to compare the heat recovery ability of the different systems. Figure 2.3.3.3 represents the ratio between the potential heat which could be extracted by de-superheater and the cooling capacity for the NH_3, CO_2 and R404A as a function of the condensing/gas cooler exit temperature.

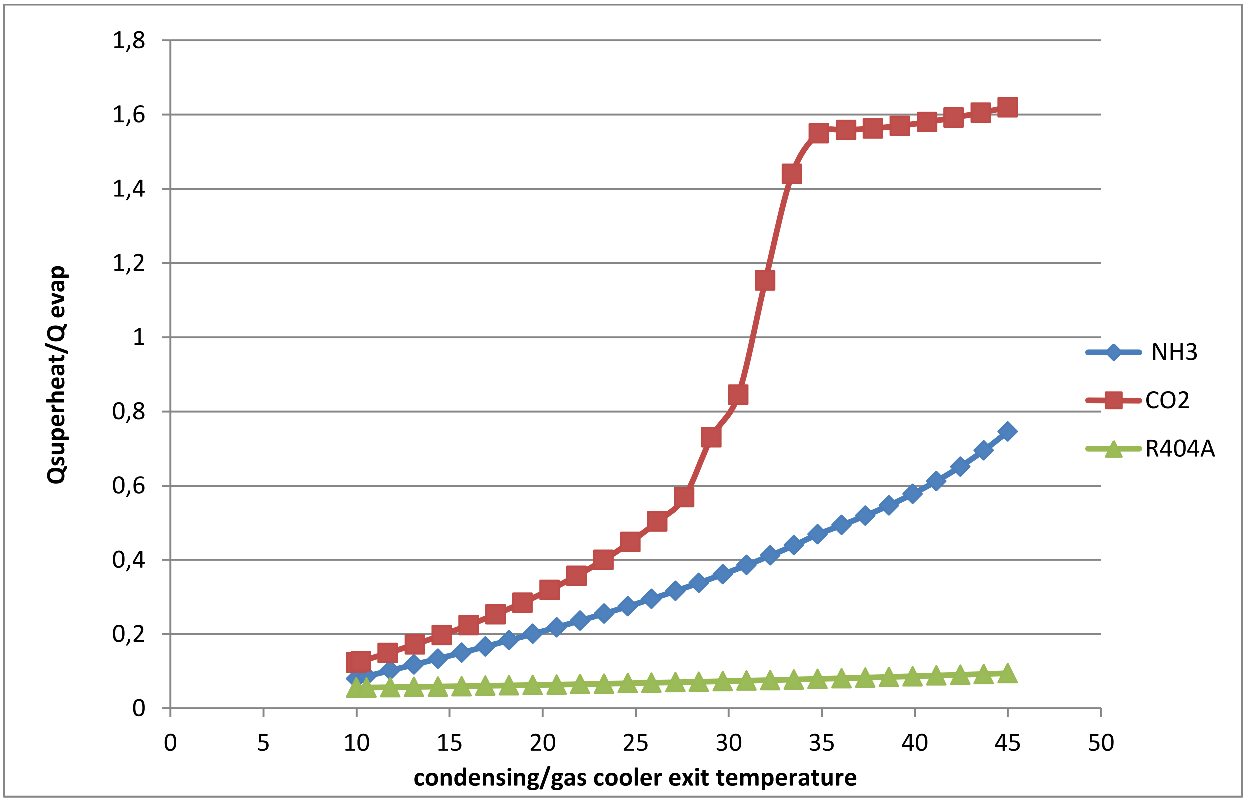

Figure 2.3.3.3: ratio of heat capacity of superheating and cooling capacity profiles for CO_2, NH_3 and R404A

Figure 2.3.3.3 is based on a calculation model which combined the simple refrigeration model described by figure 2.3.3.1 and the heat recovery system implementing a de-superheater set in figure 2.2.2. At medium temperature, the ratio between the heat capacity which potentially could be extracted by a de-superheater and cooling capacity is more important for CO_2 refrigeration system, especially after the discontinuity at 27°C, since the system is running at trans-critical conditions. Despite NH_3 has higher discharge temperature, the superheat capacity is lower than the one of CO2, since the mass flow of ammonia system is ten times lower than CO_2 or R404A systems.
For R404A refrigeration systems, the low discharge temperature makes inappropriate to the use of de-superheater. Due to this fact, although the ratio between the heat capacity extracted by a de-superheater and cooling capacity is increasing for R404, it remains really low in

comparison to other refrigerants.
Nevertheless the system COP has also to be considered. For instance at a condensing temperature of 40°C, the ratio between the heat capacity extracted through a de-superheater and refrigerating capacity is about 1,5 for CO_2 and 0,6 for NH_3 (figure 2.3.3.3), whereas the COP is about 1,7 for CO_2 and 4,7 for NH_3 (figure 2.3.3.2). Thus NH_3 refrigeration systems have better COP, but CO_2 systems have a higher ratio between capacity recovered from de-superheating and refrigerating capacity.

In figure 2.3.3.4, the calculation model which is used, combined the simple refrigeration model described by figure 2.3.3.1 and the heat recovery system with a sub-cooler described in figure 2.2.4. Nevertheless the heat pump is not considered in this calculation. Figure 2.3.3.4 represents the heat capacity which could be potentially obtained from sub-cooling for the three different refrigerant studied.

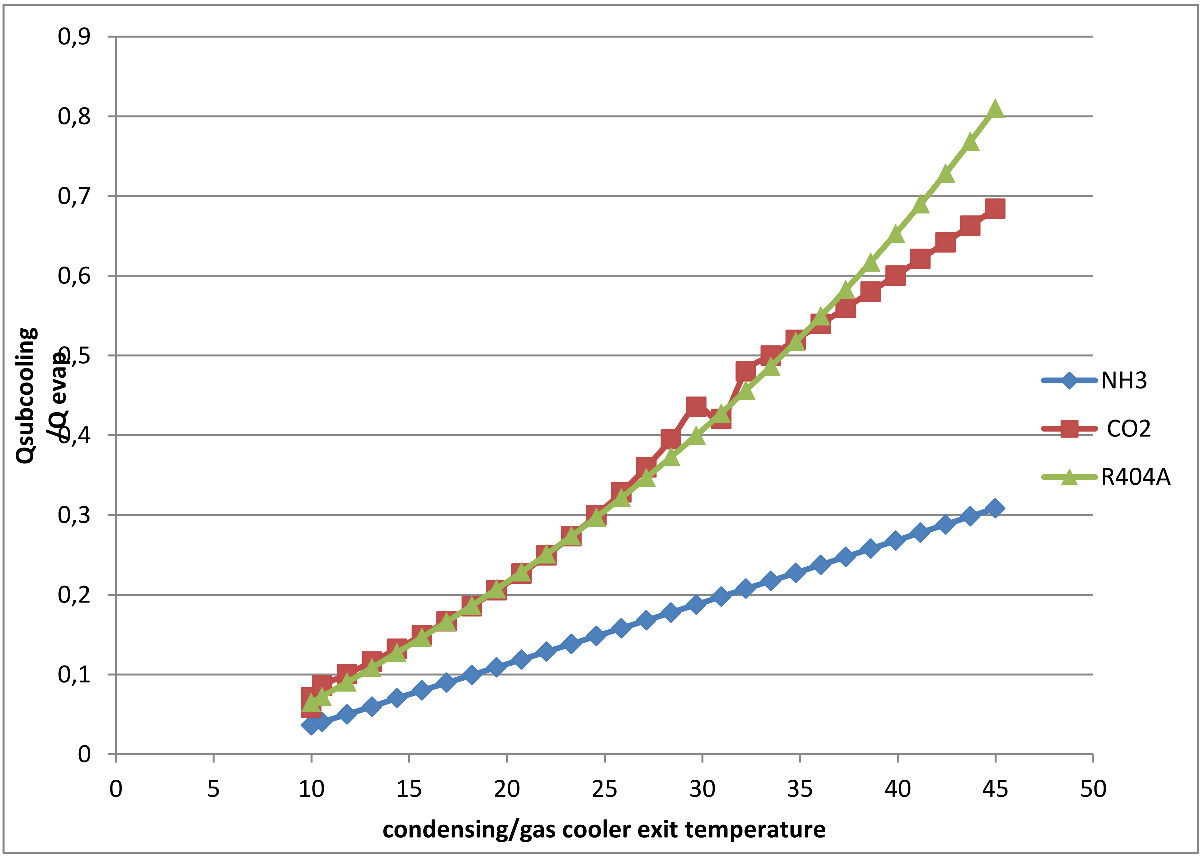

Figure 2.3.3.4: ratio heat capacity of sub-cooling and cooling capacity profiles for CO_2, NH_3 and R404A; sub-cooler outlet temperature 5°C

At medium temperature, the ratio between potential heat capacity recovered from sub-cooling and refrigerating is less for NH_3 and nearly equivalent for CO_2 and R404A. For CO_2 a discontinuity appears around 30°C since the trans-critical conditions.

At low temperature, literature study [4] has shown that due to the large difference between condensing and evaporation pressure, CO_2 system are only efficient if integrated to a cascade system. It has been validated at low temperature that NH_3/CO_2 cascade systems are much more efficient than CO_2 or NH_3 direct systems.[4] Concerning R404A, theoretical study and field data have shown that at low temperature, R404A system presents rather good COP. [8]

	NH_3	CO_2	R404A
strength	-good heat transfer coefficient -high latent heat -high COP -low GWP	-high volumetric refrigerating capacity -smaller component -low GWP -high heat recovery potentiality	-good thermodynamic properties -good COP at low temperature -easy maintenance
weakness	-smell -high practical charge -complex safety management	-low COP at low temperature -high operating pressure	-high GWP

Table 2.3.3.2: Strength and weakness of refrigerants at medium temperature

2.4 Heat recovery and floating condensing system

For supermarket with heating and refrigerating system separated, the refrigerating system operates at floating condensing mode. Therefore the condensing temperature varies according the outside temperature. The heat demand is partially or totally ensured by the refrigerating system in supermarket with heat recovery system included. The drawback of such systems is the high condensing temperature needed. Thus a system combining a heat recovery unit, a floating condensing unit and auxiliary heating system has been described by Jaime Arias [3] as less energy consuming.

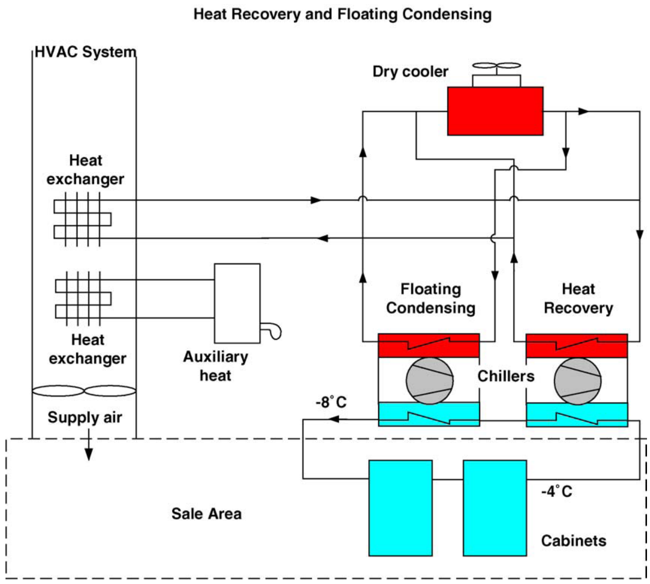

Figure 2.4.1: schematic of heat recovery and floating condensing system

3. Simulation study: refrigeration and heat recovery system analysis

The purpose of system modeling is to point out key parameters of complex system. A model validated by experiments permits to purchase further optimization, a model in contradiction with experimental results reveals interferences of neglected or unknown parameters. For the case of supermarket, deep experimental investigation cannot be realized without disturbing the supermarket activity. Therefore system simulation following by an experimental validation is more appropriated to such system. The models presented in this study have been realized with the software EES (engineering equation solver).

3.1 system modeling

3.1.1 System descriptions

First of all, system's limits, requirements and assumptions have to be well defined. Refrigeration systems are surrounded by the supermarket environment at a temperature of 18°C which is itself surrounded by the outside environment. Thus refrigeration systems interact with two different environments.

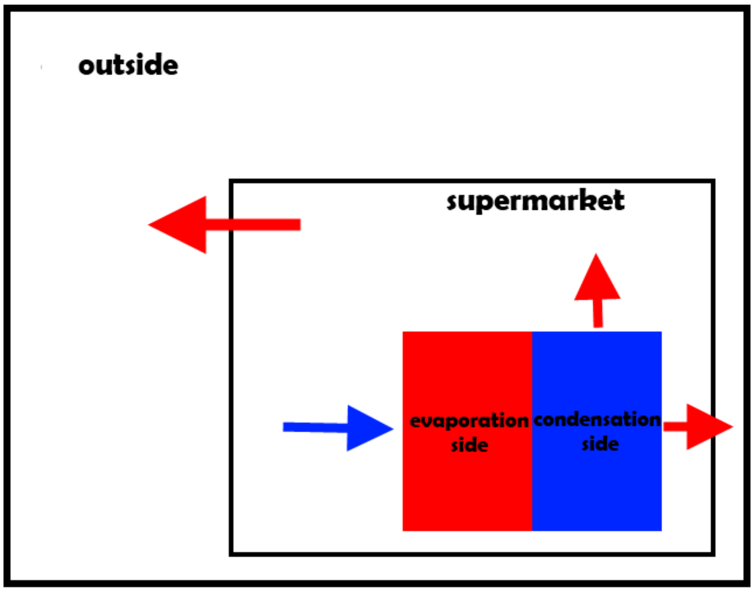

Figure 3.1.1.1: schematic of the system exchange of supermarket in winter time

The following plot was fitted from supermarket refrigeration systems have also to fulfill certain requirements, primarily to ensure a temperature around -18°C and +3°C in the freezer and chiller respectively. If a heat recovery system is included in the system, it has to provide heat to HVAC system. Supermarket heat demand is related to the outside temperature. Heat demand in average size supermarket has been calculated by using the software CYBERMART [1]. The calculated heating demand values for different ambient temperatures; this was done in earlier project [2].

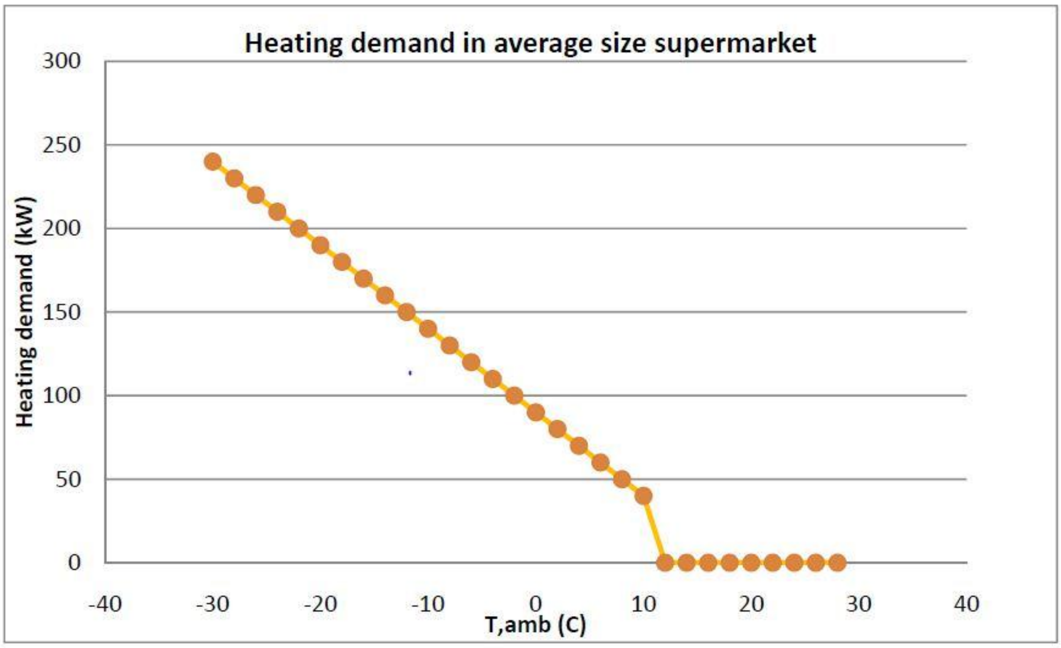

Figure 3.1.1.2: heating demand of an average Swedish supermarket [2]

In order to compare the different models, the similar assumptions have to be implemented. The following table set parameters assumed for all models described in this study:

	Low temperature	Medium temperature
Refrigerating capacity (kW)	35	100
Evaporation temperature (°C)	-35	-10
Minimum condensing temperature (°C)	10	
Internal superheat (°C)	10	
Sub-cooling (°C)	2	
FHP- return temperature (°C)	30	
Heat pump cascade-inlet temperature (°C)	13	
De-superheater return temperature(°C)	20-30	

Table 3.1.1: assumptions for the EES refrigeration model

The return temperature is the temperature of the brine coming from the HVAC system to the heat recovery system as it is set in figure 3.1.1.3. Concerning the de-superheater, this temperature is states to be ideally 20°C, but in reality this this temperature is about 30°C.

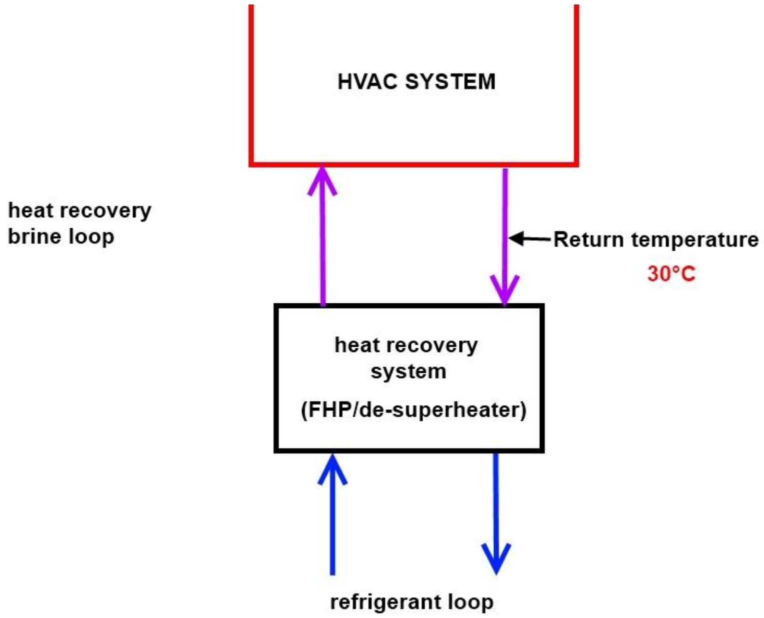

Figure 3.1.1.3: schematic diagram of the return temperature assumption for fixed head pressure and de-superheater system

When a heat pump is used the return temperature from the HVAC system remains 30°C. But a supplementary assumption is necessary concerning the temperature in the brine loop connecting the heat pump to the rest of the refrigeration system. As it is defined in figure 3.1.1.4, the temperature assumption chosen for the calculation is a heat pump cascade inlet temperature of 13°C. ([17],[2])

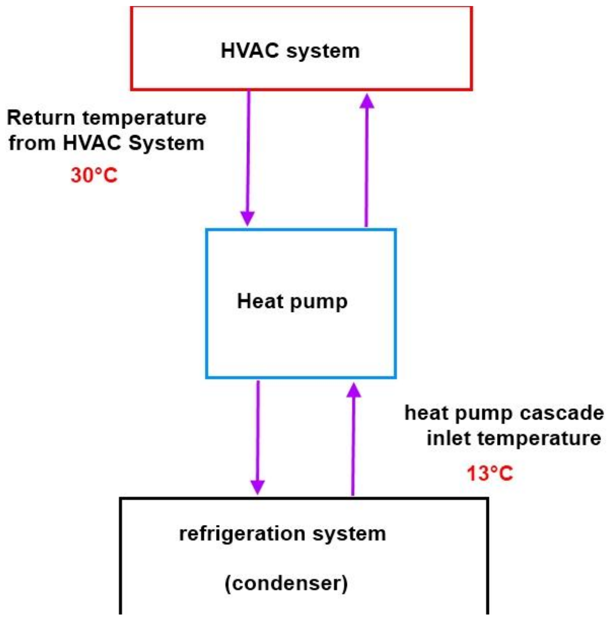

Figure 3.1.1.4: schematic diagram of the return and heat pump cascade outlet temperature assumptions

Specific assumptions have been taken for some aspects of the models; those are based on previous studies from Energy Technology department and literature.

- *pump*

In indirect system, pumps are needed in the secondary loop to drive the fluid from the refrigerating system room to the food storage cabin. The distance is relatively long. The electric power used has been estimated to 6% of the refrigerating capacity ([2], [6]).

- *Brine loss and external superheat*

In direct system, due to the long distance between compressor and evaporator, the superheat gained in the pipes between them is evaluated to be equivalent to an increase of 15°C. In indirect system, compressor and evaporator are situated in the same room, but there is still loss through the pipes. For the models, brine losses have been estimated to 7% of the refrigerating capacity.[2]

- *Heat pump*

In recovery systems, heat pumps can be connected to the condenser or a sub-cooler. For the models, a heat pump with a COP of 3.5 and a maximum heat capacity of 300 KW has been chosen. The inlet temperature minimal has been set to be 13°C. ([2], [17])

- *Compressor efficiency*

Compressor efficiency depends on temperature, pressure and refrigerant. For the models, efficiency correlation from previous studies at the Energy Technology Department have been used and completed by interpolation based on manufacturer data. Compressor efficiency correlations present the following equation form type:

$$\eta = a.(\frac{P_{cond}}{P_{evap}})^2 + b.\frac{P_{cond}}{P_{evap}} + c \qquad (3\text{-}1)$$

The constants a,b and c depends on the compressor type.

- *Refrigerating capacity at medium temperature*

The refrigerating capacity required for the supermarket cabins depends on the outside temperature and humidity. From previous work from The Energy Technology Department, a relation, based on field data, has been set between the refrigerating capacity and the outside temperature.

$$Q = (0.02. T_{outside} + 0.3). Q_{100\%} \qquad (3\text{-}2)$$

$Q_{100\%}$ is the maximal refrigerating capacity designed. Below an outside temperature of 10°C, the refrigerating capacity stays constant when the outside temperature increases (figure 3.1.1.5).

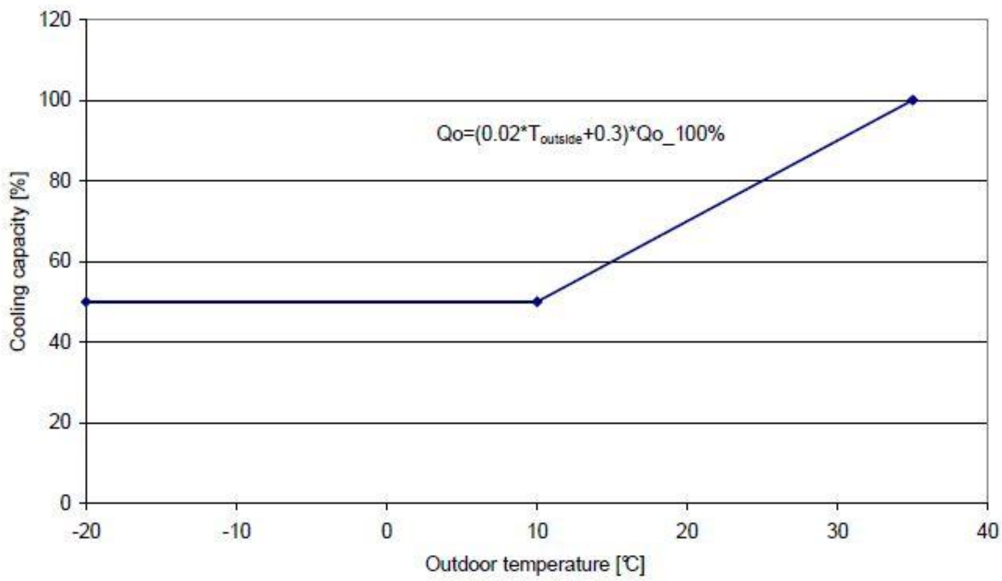

Figure 3.1.1.5: function binding the percentage used of the maximal cooling capacity to the outdoor temperature (Frelechox, 2009)

Actually in winter, humidity is much lower and store temperature is maintained constant, therefore the refrigerating capacity stays constants as well. Nevertheless, this has been observed only for medium temperature unit, low temperature unit refrigerating is not dependent on the outside condition. One reason could be freezers are often closed and cabins are more insulated.

3.1.2 System description

For this study, four different system models have been used, the conventional Swedish R404A supermarket refrigeration system as reference, the conventional US R404A system, the cascade NH_3/CO_2 system developed as prototype by The Energy Technology department [18], TR1 and TR2, two trans-critical CO_2 systems implemented in supermarkets in Sweden.

- *Conventional Swedish R404A system*

In Sweden, supermarket refrigeration systems operate traditionally with R404A as main refrigerant. As shows figure 3.1.2.2, the low temperature unit is direct system connected to the medium temperature unit through a sub-cooler. The medium temperature unit is an indirect system generally brine as refrigerant in the secondary loop in order to reduce the refrigerant charge. A refrigeration system with a CO2 pump circulation at low temperature as well, has been described feasible by [8]. Nevertheless this system is not implemented in Sweden. [2]

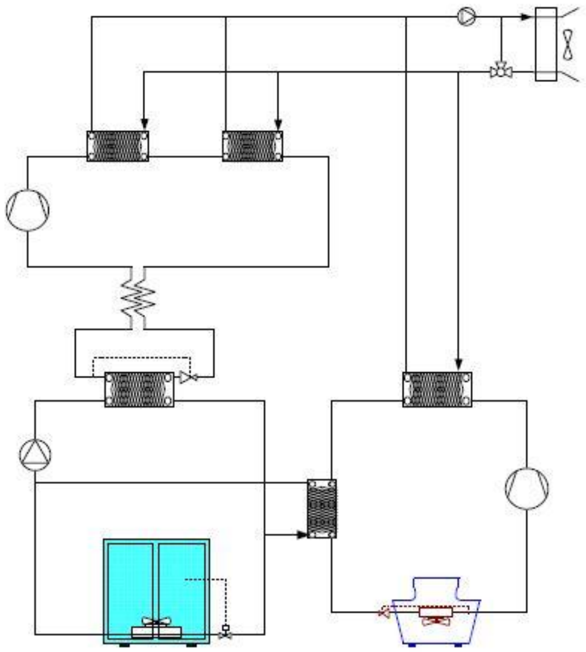

Figure 3.1.2.1: schematic of the conventional Swedish R404A system

- *Conventional US R404A system*

In United States, refrigeration main purpose is rather efficiency than refrigerant charge reduction. Therefore conventional systems in United States are generally R404A direct systems (figure 3.1.2.2) [7]. Heat is often recovered through a fixed head pressure heat recovery system with brine as working fluid in the coolant loop.

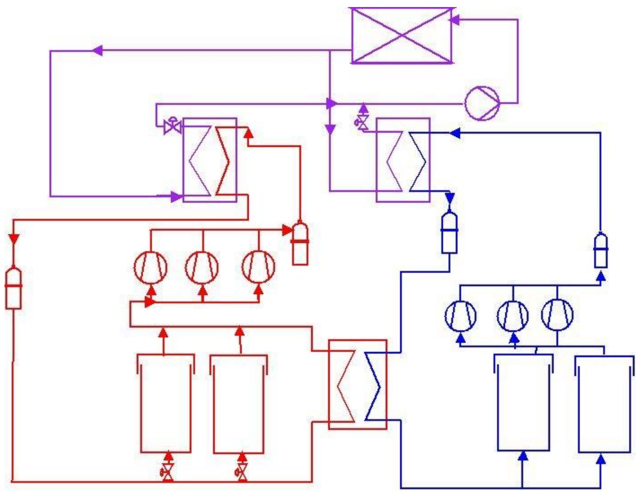

Figure 3.1.2.2: schematic of the conventional U.S.A R404A system

- *Cascade NH_3/CO_2 system*

An alternative to conventional R404A system is to use NH3/CO2 cascade system (figure 3.1.2.3). This system is equivalent to the combination of an indirect system with NH3 as main refrigerant and CO2 in the secondary loop at medium temperature and a CO2 direct which has a condensing temperature about -10°C in the cascade system. Thus the pressure level of the low temperature unit running with CO2 is acceptable. Such systems are not used in Swedish supermarket but it could be relevant to apply them. This study focuses more on this system because previous extensive study has already been carried the Energy Technology Department at KTH [2].

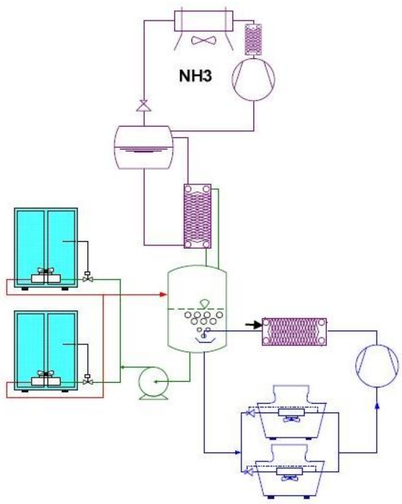

Figure 3.1.2.3: schematic of the cascade NH_3/CO_2 system

- *TR1, trans-critical CO_2 system*

TR1 is a CO2 refrigeration system currently installed in a Swedish supermarket. This a parallel system constituted by two CO2 direct systems. The low temperature level unit has relatively high pressure ratio; therefore a double stage compression with intercooler is used. If heating is needed, heat is rejected from condenser to a coolant loop which connected to heat pump through a heat exchanger.

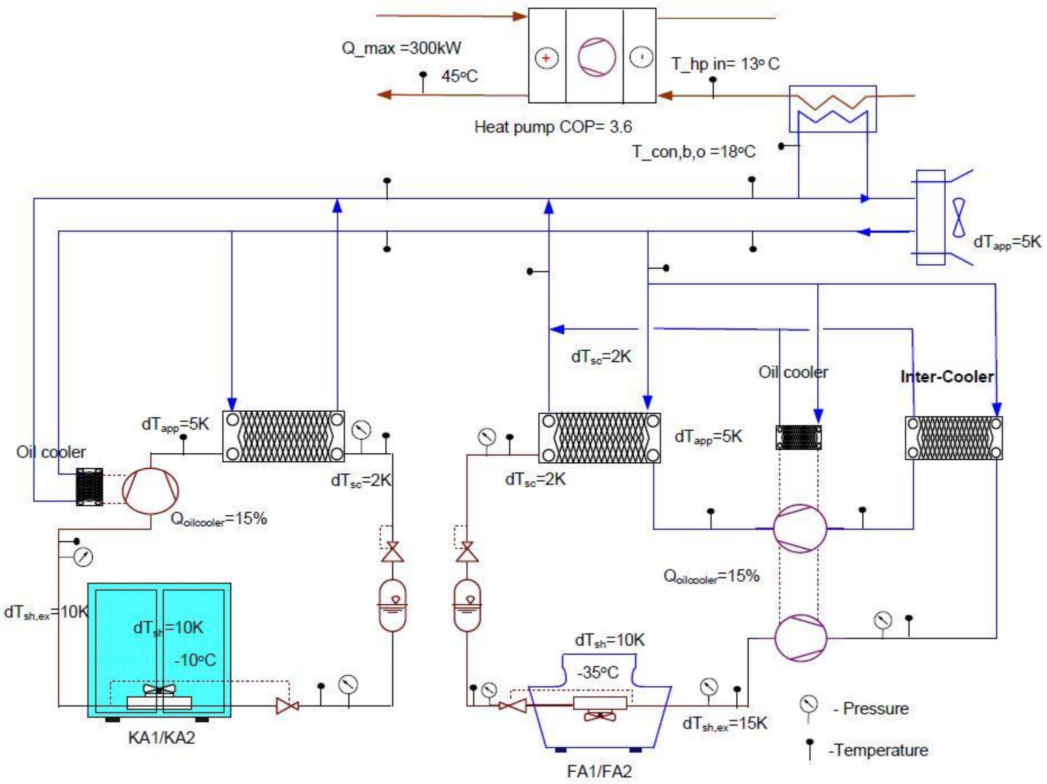

Figure 3.1.2.4: schematic of a CO_2 system: TR1 [2]

- *TR2 ,trans-critical CO_2 system*

As TR1, this system is currently implemented in a supermarket in Sweden. It is characterized by booster compressors in the low temperature unit (figure 3.1.2.5). Heat is rejected to the HVAC system through two de-superheaters located before the gas cooler.

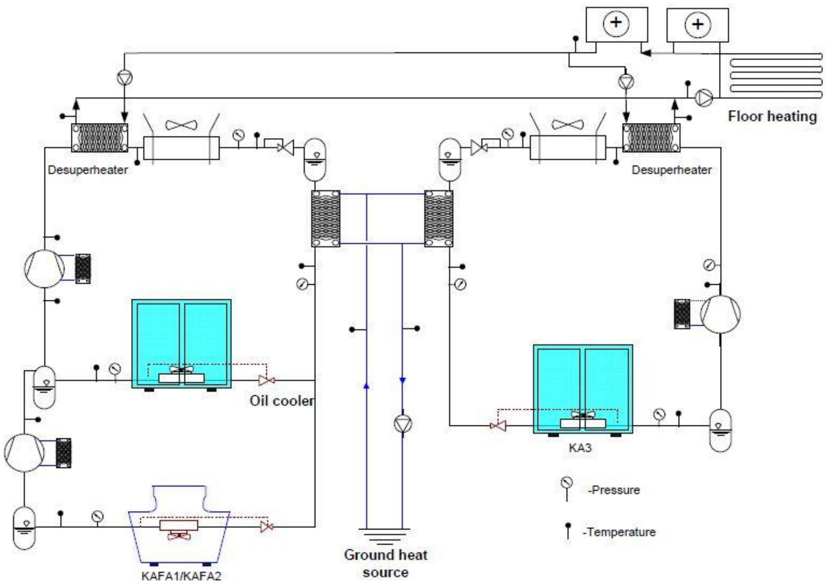

Figure 3.1.2.5: schematic of a CO_2 system: TR2

In the model system analysis, the ground heat source in figure 3.1.2.5 is not considered, and the system is simplified to the left cycle in figure 3.1.2.5, which basically is a booster system.

- *Booster system with heat pump for sub-cooling*

As figure 3.1.2.6 shows, in this system a heat pump is connected to the sub-cooler, in theory, lower condensing pressure than systems using fixed head pressure and heat pump cascade connected to the condenser is required. This system runs in floating condensing mode for temperature higher than 10°C, below 10°C the heat pump extract heat from the sub-cooler for heat recovery and the condenser operated at the minimum condensing pressure. The return temperature from the HVAC is 30°C and the sub-cooling is achieved until 7°C.

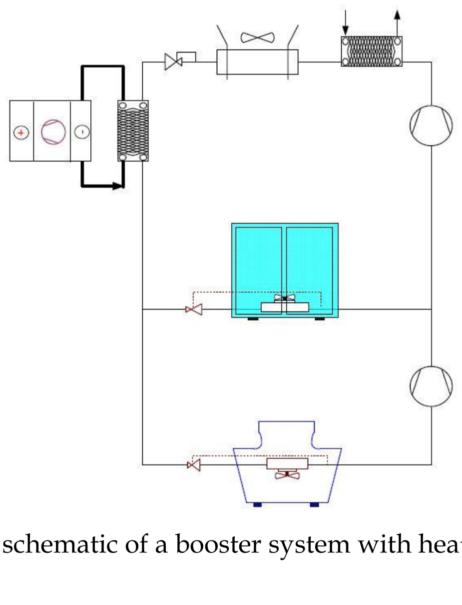

Figure 3.1.2.6: schematic of a booster system with heat pump for sub-cooling.

3.2 Performance comparison

3.2.1 System definitions

In a first time, the performance of the different refrigeration systems studied has to be compare and in a second time, a performance comparison between different combination refrigeration system-heat recovery systems has to be performed since system behavior in heat recovery mode and floating condensing can differ. Nonetheless not all possible combinations, refrigeration-heat recovery, system have to be analyzed; actually some are known to be inefficient.

	Fixed head pressure	de-superheater	Heat pump cascade	Heat pump cascade for sub-cooling
Conventional Swedish R404A	yes	no	no	-
Conventional US R404A	yes	no	no	-
Cascade NH3/CO2	yes	yes	yes	-
TR1	yes	yes	yes	-
TR2 (booster part)	yes	yes	yes	yes

Table 3.2.1: system refrigeration solution studied

Fixed head pressure system is the most conventional heat recovery system, since this system is simple and easy to control.
De-superheater fits to system running with CO_2 and NH_3 since both have a high compressor discharge temperatures, respectively 67°C and 50°C ([2], by assuming isentropic compression between -5°C evaporation and 25°C condensing°); de-superheater can also be used with R404A system but not to supply completely the whole heating demand of supermarket, since discharge compressor temperature is low for R404A system (30°C, [2] assuming isentropic compression between -5°C evaporation and 25°C condensing).
Heat pump cascade connected to the condenser or the sub-cooler allow systems to operate at lower condensing temperature which allows the system to operate at relatively high the COP.
In the following analysis, only the COP of the medium temperature unit of each system is considered. One practical reason is that the previous study performed by the Energy Technology department of KTH [2] focuses on the medium COP. Thus comparisons between the results become possible. Another reason is that using heat recovery for the low temperature unit tends to decrease drastically the system total COP. This is studied more in detail in part 4 of this master’s thesis report.

3.2.2 Floating condensing mode

At floating condensing mode, the condensation temperature follows the outside temperature above 5 °C, below which it stays constant 10°C. Therefore at floating condensing mode, the medium COP is constant until an outside temperature of 5°C as figure 3.2.2.1 shows. For the all systems studied, the coolant loop is not considered in the calculations.

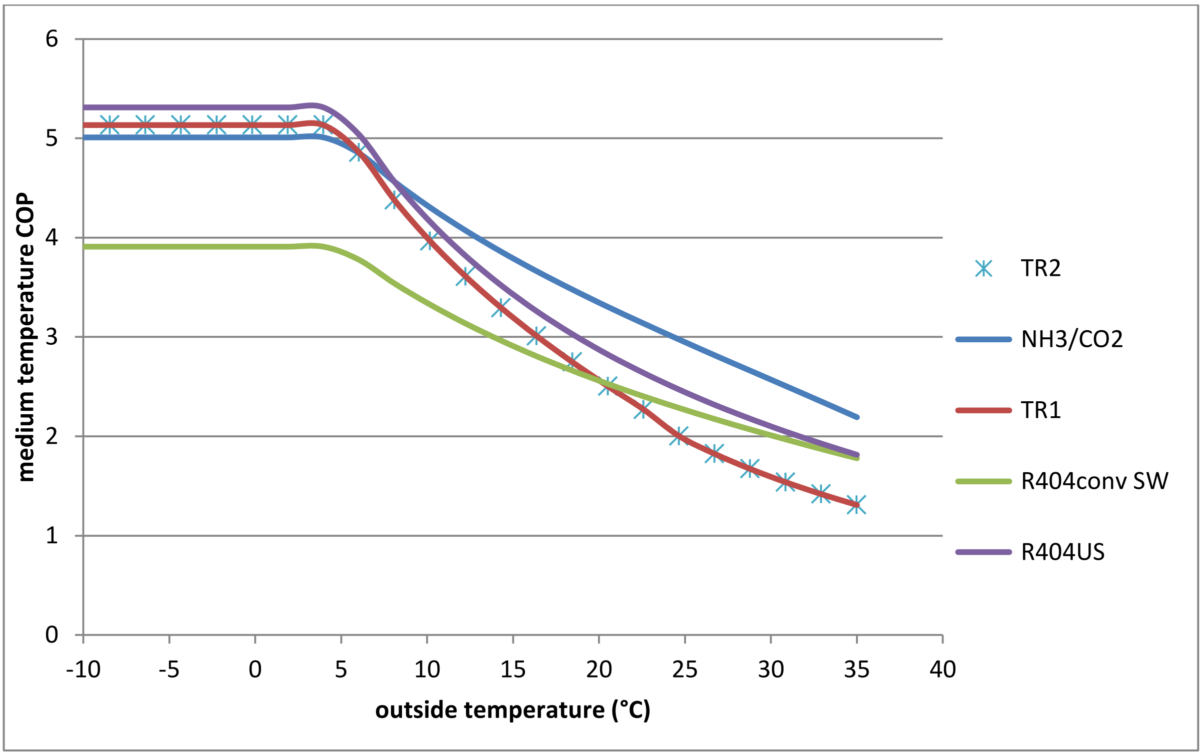

Figure 3.2.2.1: medium temperature COP profiles in floating condensing mode depending on the outdoor temperature

The medium COP of R404A Swedish conventional system is lower than other systems which have globally equivalent COP. This could be explained by the extra electric power used for the pump in the secondary loop and the fact that medium evaporation temperature has to be lower to compensate the brine losses. The medium COP of R404A U.S conventional system is higher than the Swedish one, because this is direct system, thus the losses are less important. It can also be observed that the cascade NH_3/CO_2 system presents better COP especially when the condensing temperature is fluctuating which is explained by the fact that ammonia system tends to have better COP than CO_2 and R404A systems (Table 2.3.2). Nevertheless, the cascade NH_3/CO_2 system presents slightly a lower medium COP in the constant part of the profile. The reason is that not only the ammonia cycle unit is considered in the medium temperature unit but also the tank and the secondary coolant loop (figure 3.2.2.2). It is coherent that TR1 and TR2 have exactly the same medium COP profile, since both cycles are identical concerning the medium temperature unit. After an outside temperature of 20°C, CO_2 systems present the lowest medium COP. This is due to the fact that those systems are operating trans-critically.

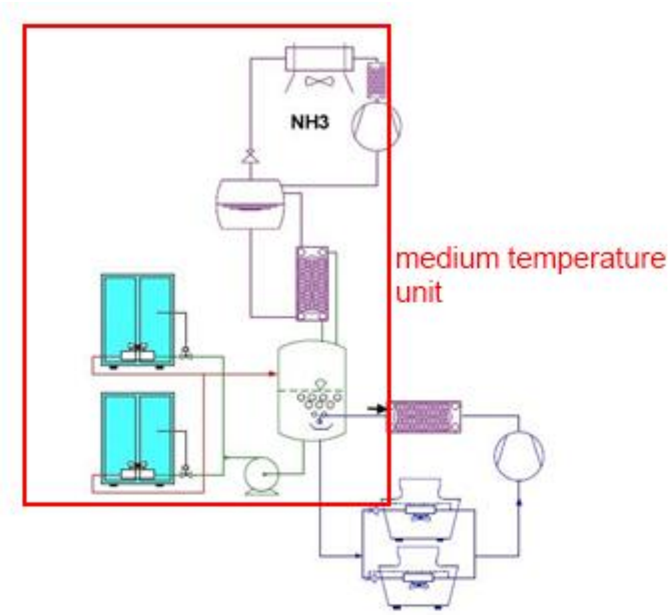

Figure 3.2.2.2: schematic diagram of medium temperature unit of the NH_3/CO_2 cascade system.

3.2.3 Fixed head pressure and heat pump cascade connected to condenser

In heat recovery mode, fixed condensing pressure and temperature are applied in order to fulfill the supermarket heat demand with the condition that the return temperature of the working fluid in the coolant loop from the HVAC system is 30°C. After 10 °C, heating is not needed anymore, and the system is operating at floating condensing conditions. Trans-critical CO_2 system (TR1) has a lower COP than conventional R404A system with a fixed head pressure system, since CO_2 systems have generally relatively low COP at high heat sink temperatures.

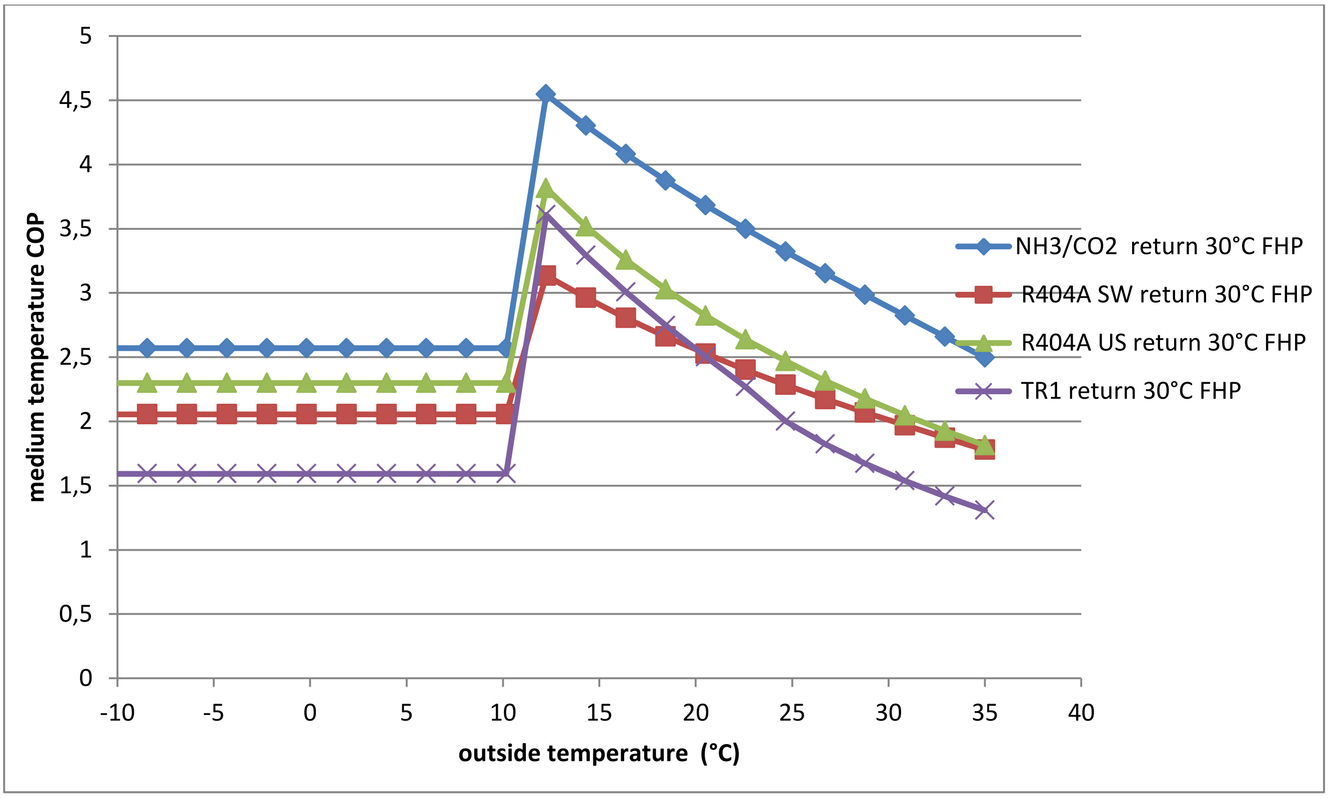

Figure 3.2.3.1: medium temperature COP profiles with fixed pressure depending on the outdoor temperature

Nevertheless by including to the heat recovery system a heat pump cascade as described in figure 2.2.3 and with condition set in table 3.1.1, trans-critical CO2 system becomes more competitive than the Swedish conventional systems in heat recovery mode as figure 3.2.3.2 shows. The heat pump allows the system to have a return temperature to the system condenser at 13°C instead of 30°C: The heat pump is controlled in a way that the return temperature from the HVAC system to the heat remains to be 30°C.

Ammonia system generally have higher COP than other system [4], which appears in this case, NH3/CO2 system with heat pump cascade has higher COP than all other systems and with fixed head pressure, the system has a rather acceptable COP.

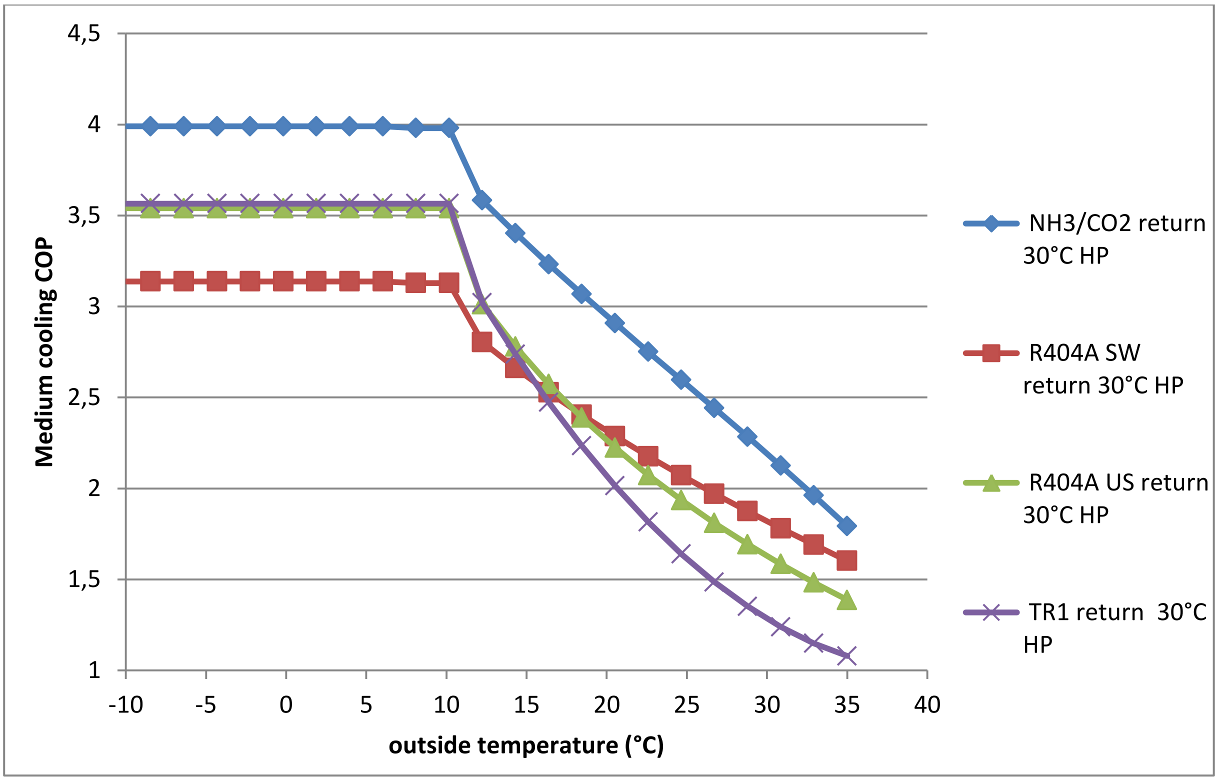

Figure 3.2.3.2: medium temperature COP profiles with heat pump cascade connected to the condenser depending on the outdoor temperature

3.2.4 De-superheater

In comparison to refrigeration NH_3/CO_2 systems using fixed head pressure and heat pump cascade systems, NH_3/CO_2 systems using de-superheater presents a lower cooling COP in the case of 100% of the heat demand, defined in figure 3.1.1.2, is satisfied. Figure 3.2.4.1 compares the medium cooling COP of NH3/CO_2 systems using fixed head pressure and heat pump cascade systems calculated in part 3.2.3 with the cooling medium COP of NH_3/CO_2 systems using de-superheater ensuring 100%, 70% and 40% of the heating supermarket demand (figure 3.1.1.2). However as it is seen in figure 3.2.4.1, NH_3/CO_2 system using de-superheater has rather acceptable COP in comparison to the NH_3/CO_2 systems using fixed head pressure and heat pump cascade if they partially supply the supermarket heat demand. As in previous study of the Energy technology department of KTH [2], the return temperature to the de-supereater (figure 3.1.1.3) is set at 20°C in first time and 30°C in a second time. This way the results obtained in this thesis for the NH_3/CO_2 system using de-superheater can be compared with the results of TR2 with a de-superheater determined by the Energy technology department of KTH [2].

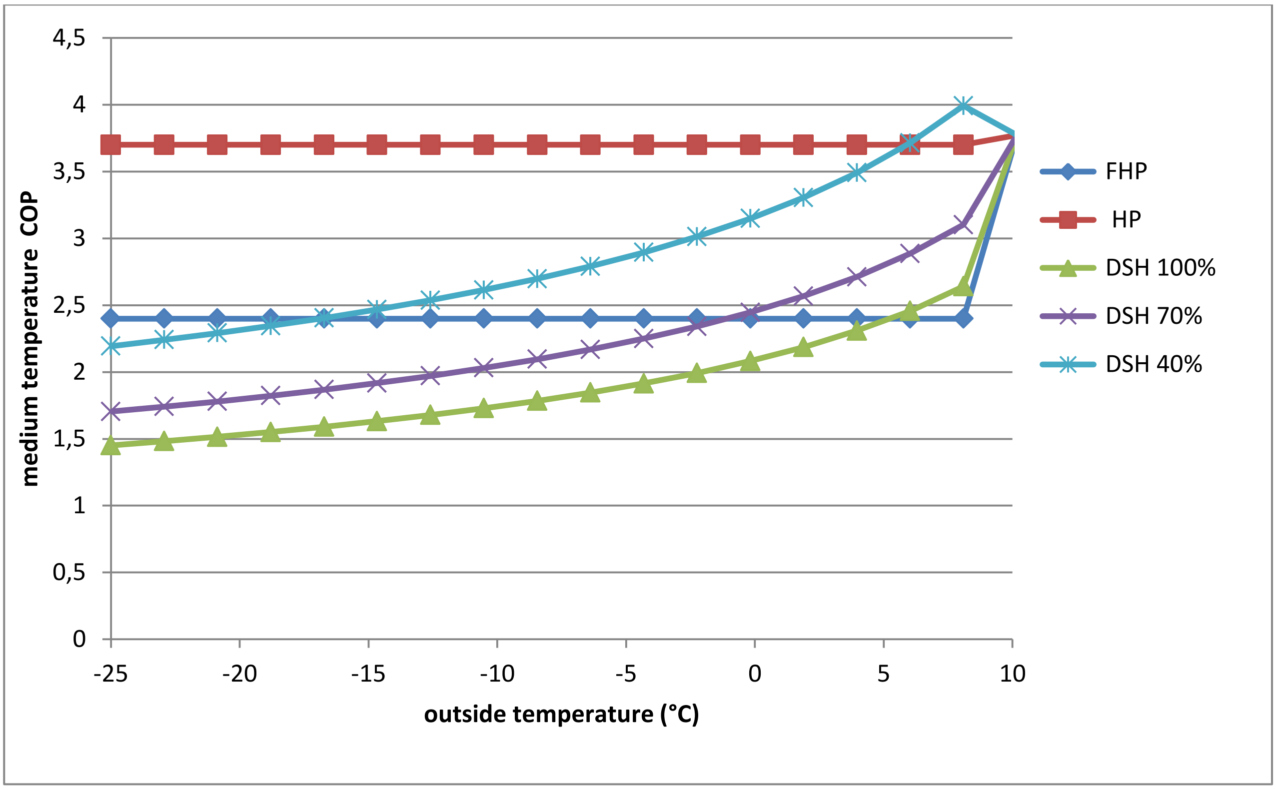

Figure 3.2.4.1: medium temperature COP profiles with fixed head pressure, heat pump cascade and de-superheater for NH_3/CO_2 cascade depending on the outdoor temperature (return temperature 20°C)

Unlike fixed head pressure or heat pump cascade systems, de-superheater allows condensing temperature to vary in heating recovery mode, and then the heat recovered can closely fit to the heating demand. For instance systems with fixed head pressure in recovery mode, the condition of a temperature of 35°C furnished to the HVAC system implies a high condensation temperature, thus as shows figure 3.2.4.2 the amount of heat that can be recovered is much larger than the heating demand. Nevertheless to supply enough heat to the supermarket and a temperature of 35°C to the HVAC system with a de-superheater, high condensation temperatures are required when the outside temperature is low. Therefore combined to an auxiliary heating system, such systems could be reliable since this way the condensation temperature could maybe be reduced.

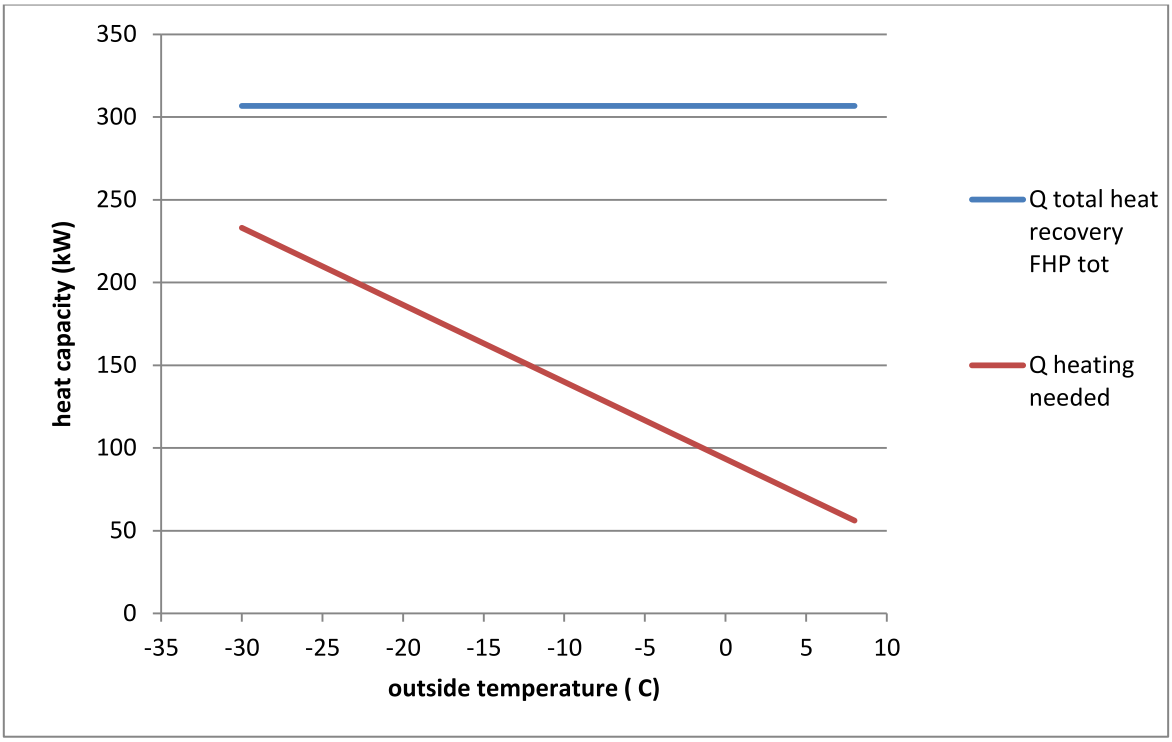

Figure 3.2.4.2: comparison between supermarket heating demand and heat capacity that can be recovered with fixed head pressure system

Figure 3.2.4.3 and 3.2.4.4 represent medium temperature COP for TR2 and NH_3/CO_2 cascade systems using de-superheater with 20°C (figure 3.2.4.3) and 30°C (figure 3.2.4.4) of return temperatures depending on the outdoor temperature. In figure 3.2.4.3 the COP stay constant from an outside temperature of -20°C to an outside temperature of -2°C. This is due to the fact that the condensing temperature is reached inside the de-superheater and the possibility to reject all the refrigeration system heat through de-superheater [2]. In figure 3.2.4.4, there is no flat line, the COP is slightly increasing for outside temperatures below -2°C.The assumed 20°C return temperature from the heating system is rather low, 30°C seems more realistic TR2 and NH_3/CO_2 cascade systems are both using de-superheater for heat recovery. In recovery mode, according to figure 3.2.4.3 and 3.2.4.4, TR2 cooling COP is higher than NH_3/CO_2 cascade in heat recovery mode. Actually since TR2 is at trans-critical state due to CO2 properties, the potential heat which could be recovered from the de-superheater is higher (Figure 2.3.3.3). Therefore for an identical heat recovery ratio, TR2 presents a better COP.

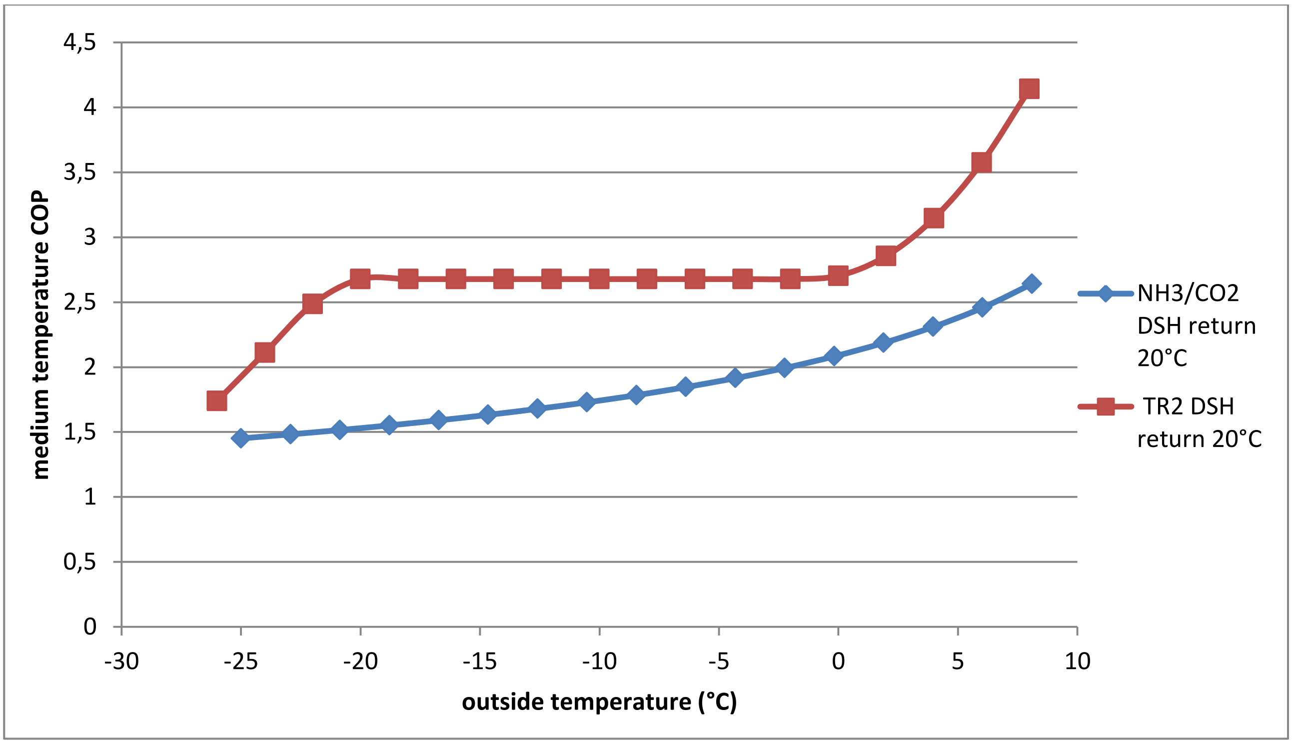

Figure 3.2.4.3: medium temperature COP profiles with de-superheater for TR2 and NH_3/CO_2 cascade with 20°C of return temperature depending on the outdoor temperature

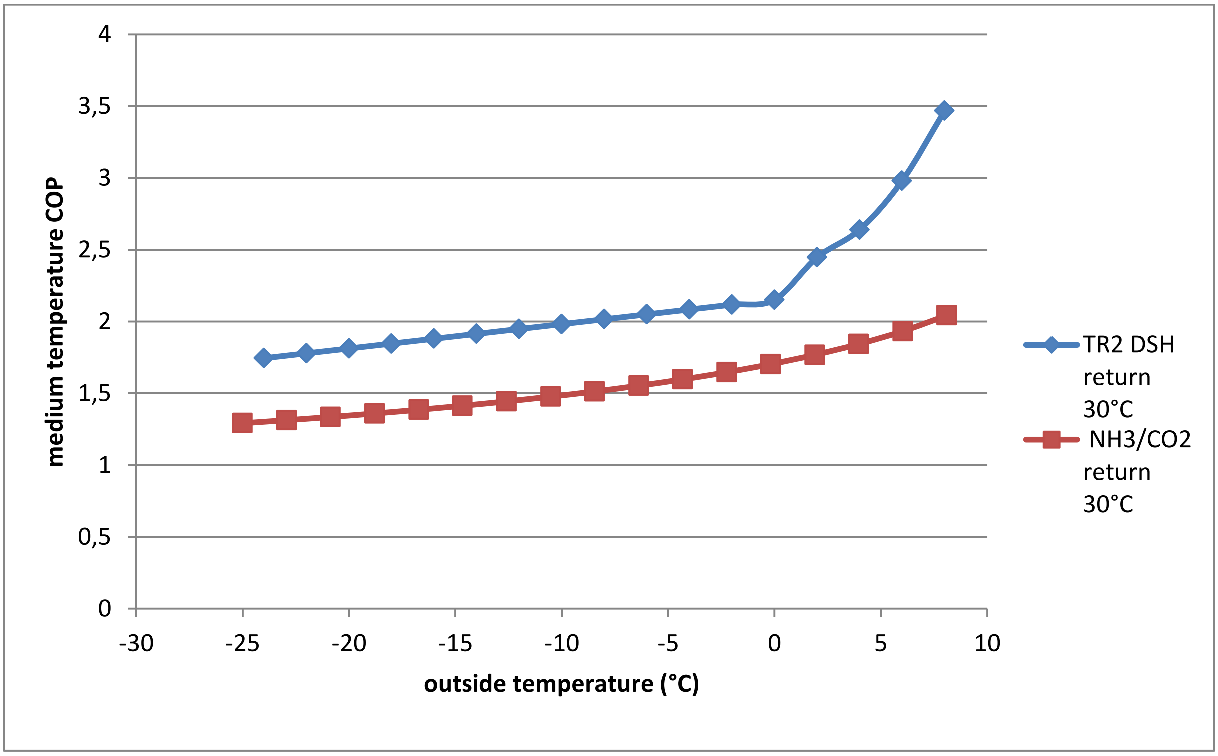

Figure 3.2.4.4: medium temperature COP profiles with de-superheater for TR2 and NH_3/CO_2 cascade with 30°C of return temperature depending on the outdoor temperature

In figure 3.2.4.5, TR2 thermodynamic cycles are represented on a pressure-enthalpy diagram for different condensing temperature. It can be seen that the potential heat which could be recovered by de-superheater increases considerably just above the critical point. In addition, it has been observed that NH_3/CO_2 systems have a mass flow ten times lower in the ammonia cycle than the one of CO_2 systems; thus even if ammonia discharge temperature is higher than the one of carbon dioxide, in heat recovery mode, a larger heat capacity than ammonia system can be provided through de-superheater with carbon dioxide system.

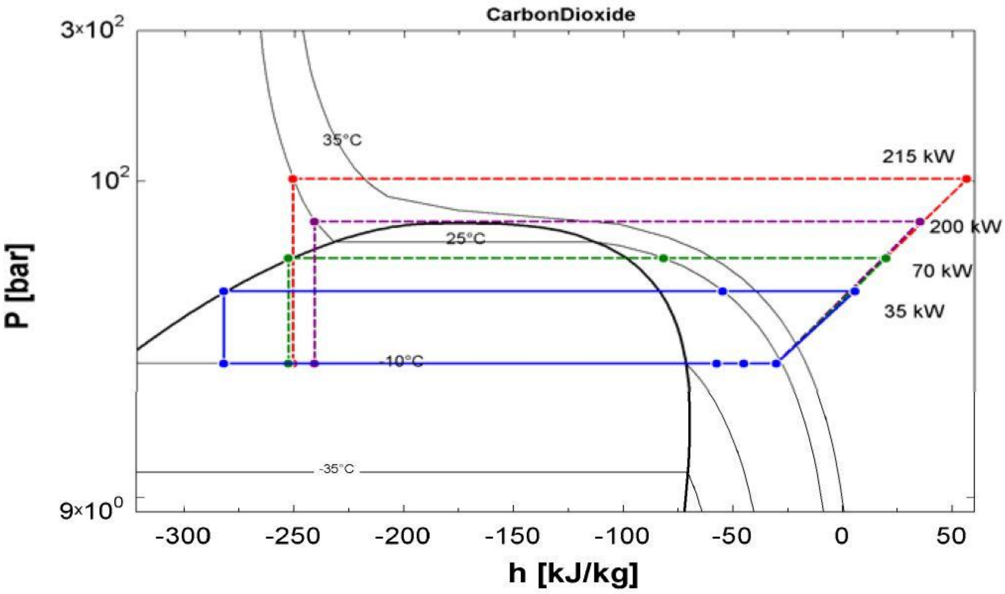

Figure 3.2.4.5: TR2 cycles represented on the pressure-enthalpy diagram of CO_2 [2]

3.2.5 Heat pump cascade for sub-cooling

For heat recovery systems with heat pump connected to the condenser/gas cooler, the heat pump has to be designed and the condensing temperature has to be fixed in a way to ensure an appropriate heating even for very low outside temperature. With the connection of a heat pump to a sub-cooler added after the condenser (figure 2.2.4) , the heat from the condensation can be modulated and adapted to the heating need by switching off or by passing the condenser. In figure, 3.2.5.1 an example of control of the sub-cooler heat capacity is drawn on a pressure-enthalpy diagram. In this example, the sub-cooler heat capacity is adapted to the heat pump demand by reducing the condenser heat capacity. Therefore systems with a heat pump connected to a sub-cooler are able to run at lower condensation pressure. In addition the sub-cooling increases the total COP by reducing the mass flow needed, thus the compressor power as well.

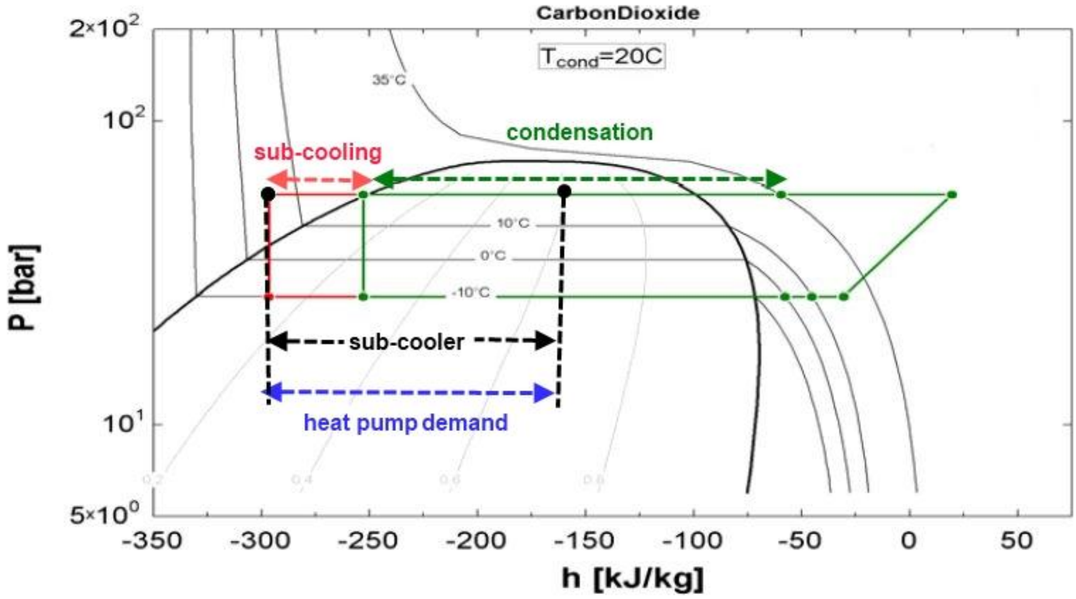

Figure 3.2.5.1: Pressure-enthalpy diagram representing an example of heat pump cascade for sub-cooling operation

In figure 3.2.5.2, the total COP without considering the heat pump consumption are compared for CO_2 systems using heat pump cascade connecting to a sub-cooler. The calculations have been made for sub-cooler exit temperature of 5°C, 2°C, -4°C and -7°C. As shown in figure 3.2.5.2, the total COP increases when the outlet sub-cooler is getting lower.

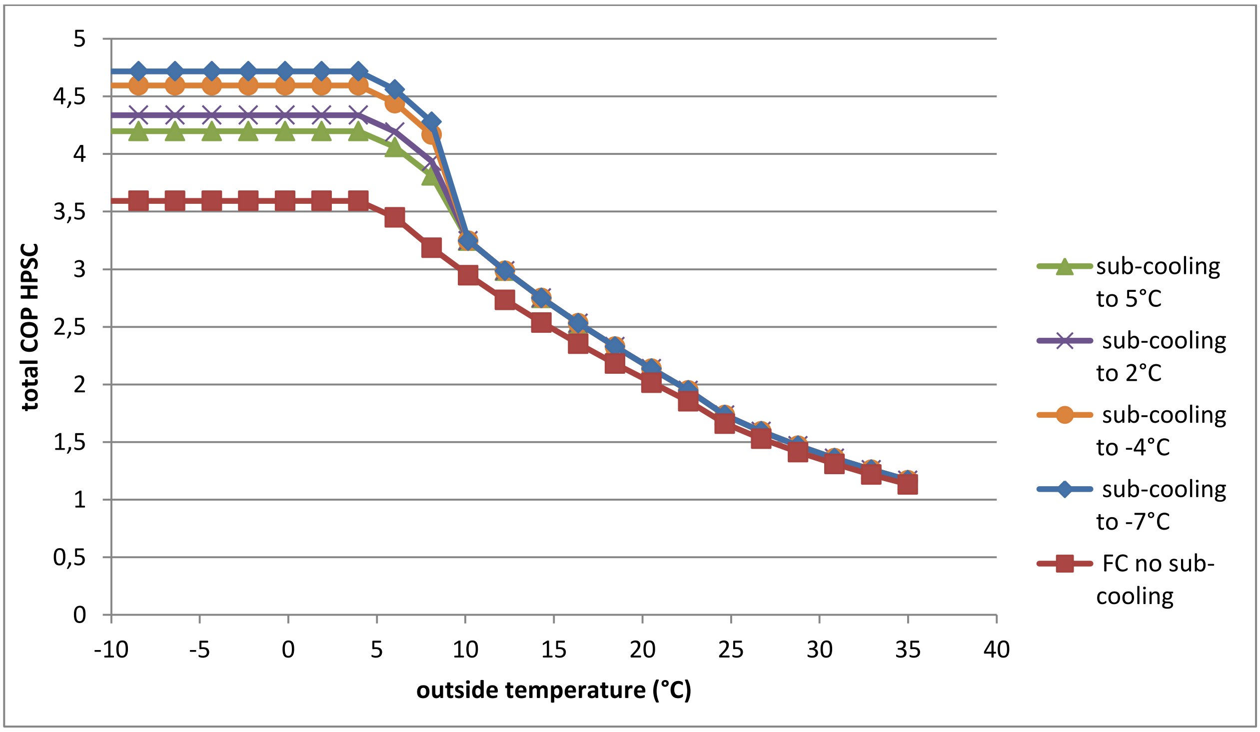

Figure 3.2.5.2: total COP profiles with heat pump for sub-cooling for different outlet sub-cooler temperature depending on the outdoor temperature

According figure 3.1.1.2, the maximum heat capacity needed for an average supermarket is around 250kW, which corresponds to an outside temperature at -30°C. The heat capacity of condensation for system running at a condensing temperature between 5-15°C is, according to the simulation, stated between 280-300kW (for CO_2 system and with the assumptions set in part 4.1.1). Figure 3.2.5.3 represents the total COP profiles with heat pump for sub-cooling for different condensing temperatures 5°C, 10°C and 15°C depending on the outdoor temperature without considering the heat pump power use. If it is assumed that refrigeration systems could be able to operate at a condensing temperature of 5°C, a COP equal to 5,25 could be reached instead of a COP of 4,35 with a condensing temperature of 10°C (figure 3.2.5.3) This is rather a huge enhancement in term of COP.

For most of refrigeration systems, the minimum condensing temperature is chosen in order to ensure a pressure difference across the expansion valve high enough to permit the expander to work correctly and in order to avoid frosting in the condenser. This temperature is often set to be 10°C. However CO_2 systems are known to run at high pressure even at low temperature due to CO_2 proprieties. It should be in theory possible for CO_2 system to operate at a condensing temperature below 10°C. Therefore CO_2 system with heat pump connected to sub-cooler could be a reliable alternative to system running with artificial refrigerant.

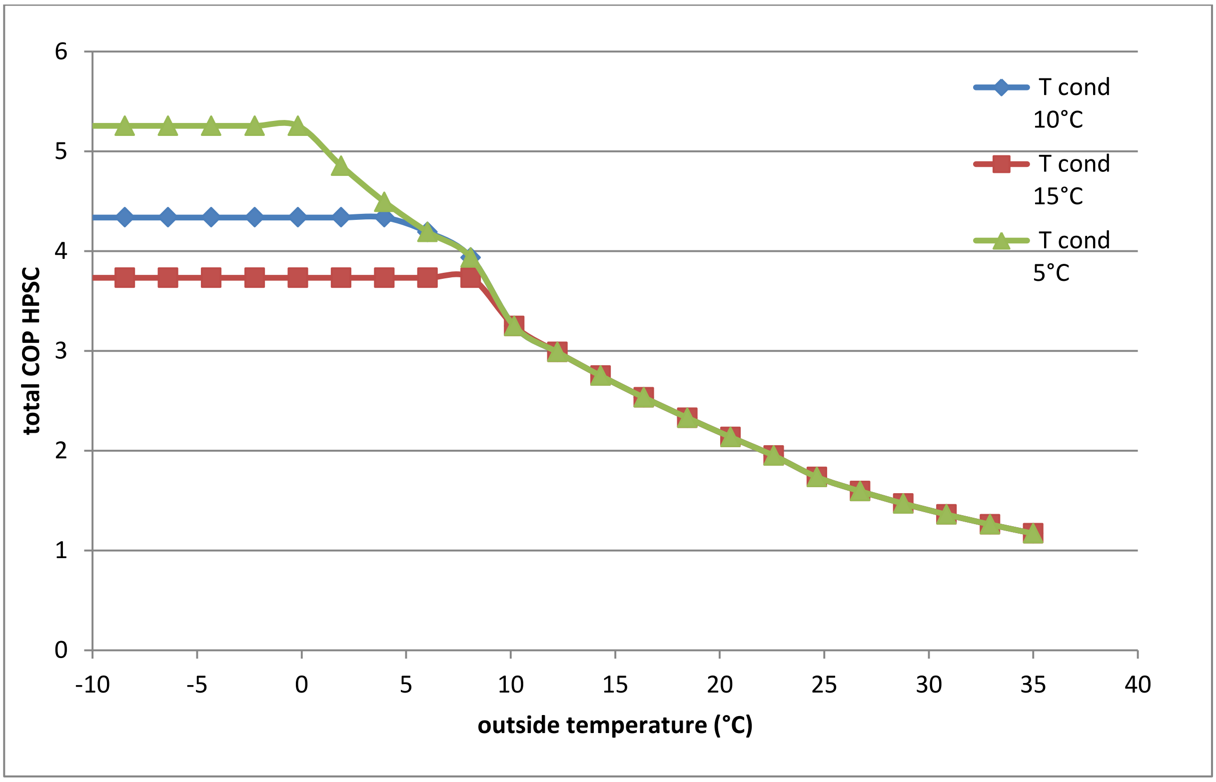

Figure 3.2.5.3: total COP profiles with heat pump for sub-cooling for different condensing temperatures depending on the outdoor temperature without considering the heat pump power use

3.2.6 Heating COP

In heat recovery mode, systems are working as cooling and heating systems at the same time. Thus the system efficiency should be described by a cooling COP and a heating COP. The heating COP is defined as the ratio between the heat recovered and the energy consumption difference between the system running in heat recovery mode and floating condensing mode.

$$COP = \frac{Q_{heat\ recovery}}{E_{heat\ recovery\ mode} - E_{floating\ condensing\ mode}} \qquad (3\text{-}3)$$

In figure 3.2.6.1, the total heating COP are calculated for fixed head pressure systems combined with NH3/CO2, TR1, the Swedish and U.S conventional R404A systems. In figure 3.2.6.2, the total heating COP of the same refrigeration systems are estimated but this times the heat recovery system used is a heat pump cascade connected to the condenser (figure 2.2.3).The heating COP of systems using fixed head pressure or heat pump connected to the condenser are in accordance with their cooling COP. The ammonia system COP is higher and the TR1 COP is the lowest one. The heating COP of TR1 is lower than the other systems because it has low COP at high condensing temperatures. This is due to the fact in heat recovery mode, the condensing temperature (or gas cooler exit temperature above the CO_2 trans-critical point) used to be higher. At the high temperature level CO_2 used to have higher energy consumption than NH_3 and R404A. Therefore the work needed in order to reach this pressure level is more important for CO_2 than the other refrigerants. Nevertheless by using a heat pump cascade connected to the condenser instead of a fixed head pressure system for CO_2 system, CO_2 system can operate at lower condensing temperature so lower pressure level. That is why in figure 3.2.6.2, TR1 heating COP is approximately equivalent to R404A and NH_3 heating COP for outside temperature below -2°C.

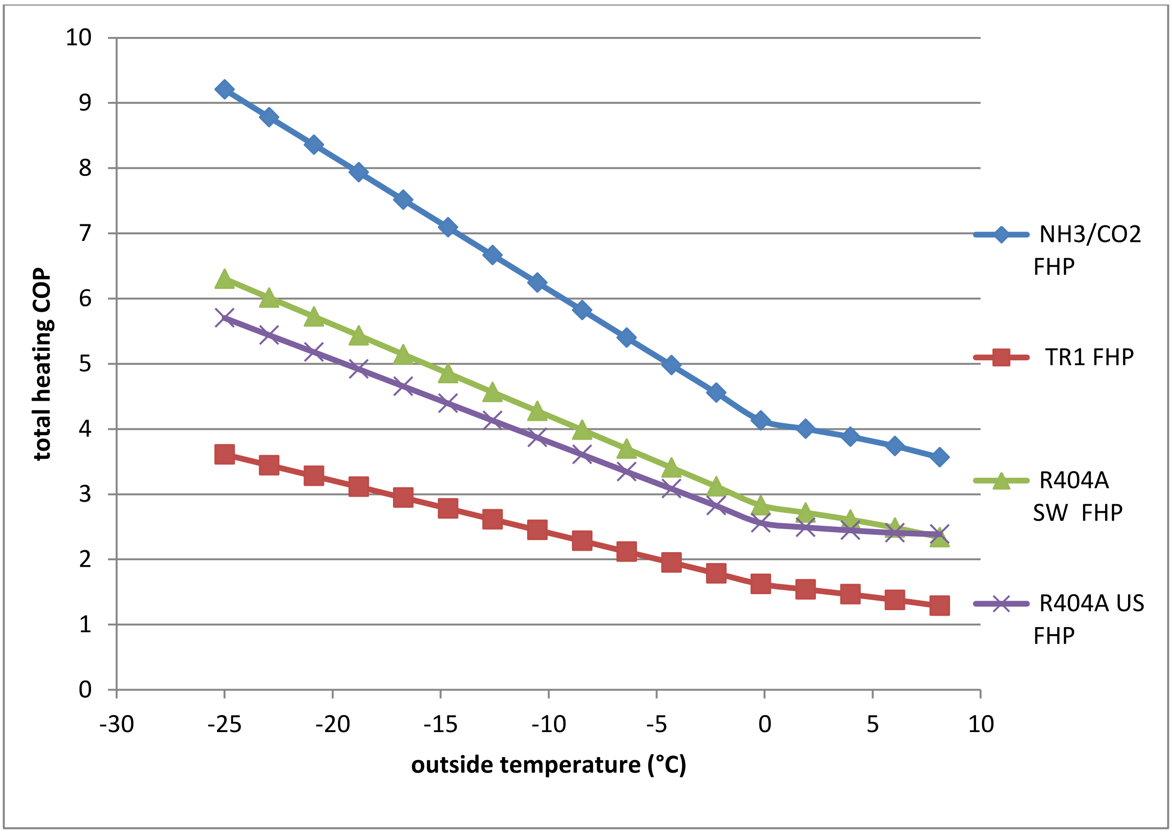

Figure 3.2.6.1: total heating COP profiles with fixed head pressure systems depending on the outdoor temperature

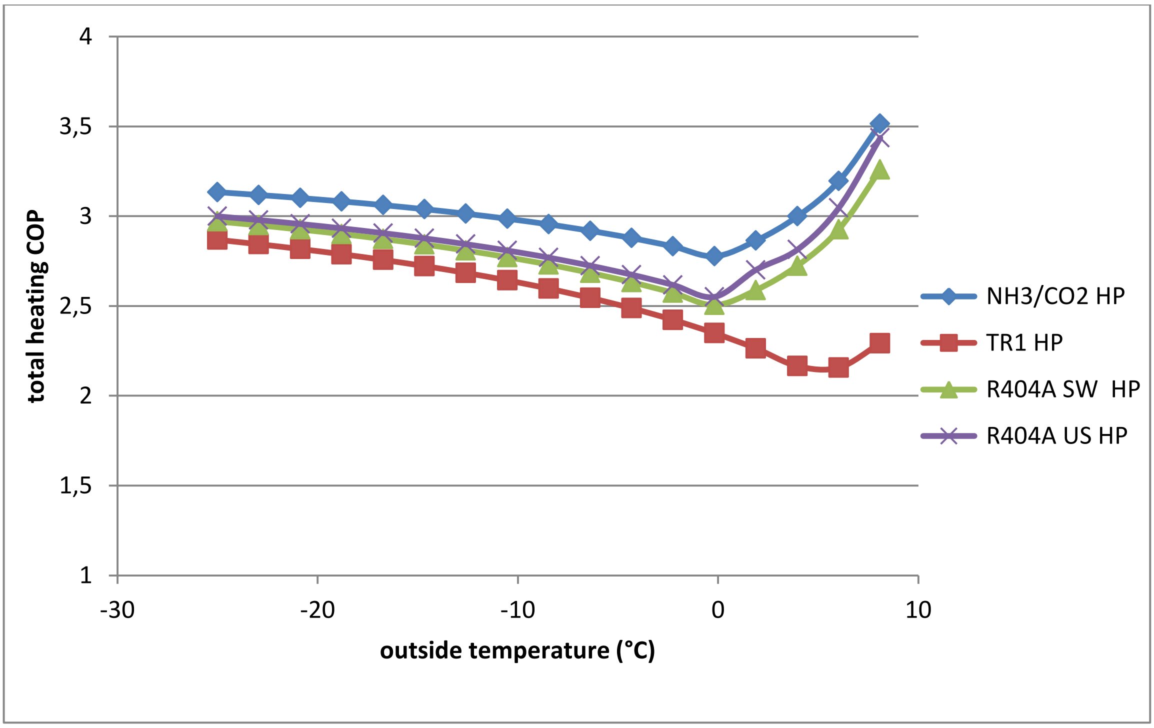

Figure 3.2.6.2: total heating COP profiles with heat pump cascade depending on the outdoor temperature

For systems using de-superheater, such as NH_3/CO_2 cascade and TR2, to recover heat, the return temperature is ideally 20°C [2], but data field measurements show that this is in reality around 30°C. Since the difference of enthalpy is lower with a return temperature of 30°C, the heating COP is as well lower. Figure 3.2.6.1 represents the total heating COP of TR2 and NH3/CO2 systems using a de-superheater with return temperatures of 20°C and 30°C (figure 3.1.1.3) depending on the outdoor temperature. According to figure 3.2.6.1, NH_3/CO_2 cascade heating COP is higher than TR2 heating COP. As it has been set previously, ammonia system potential superheat capacity recovered is lower than the one of CO_2, but the lower energy difference between heat recovery and floating condensing mode of the ammonia system overcomes this and permits NH_3/CO_2 cascade to have a better heating COP than TR2.

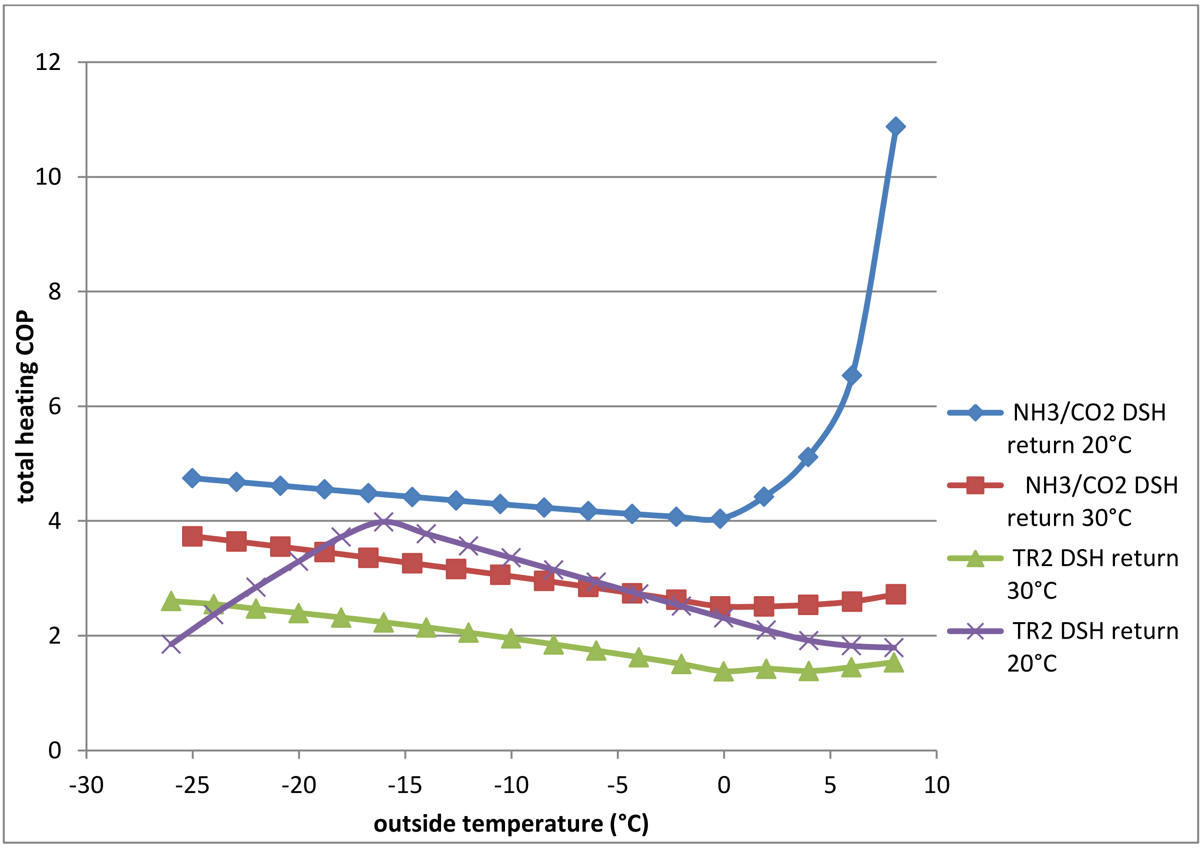

Figure 3.2.6.1: total heating COP profiles with de-superheater depending on the outdoor temperature

3.3 Parameters study

3.3.1 Sub-cooling

Sub-cooling increases usually the system performance. When there is sub-cooling, the condensation and evaporation enthalpy difference are both larger as it is seen in figure 3.3.1.1, which allows the system to operate with a lower mass flow. Due to this lower mass flow, less electrical power is needed to run the compressor. However a lower mass flow leads also to a decrease of heat which can be potentially recovered by a de-superheater.

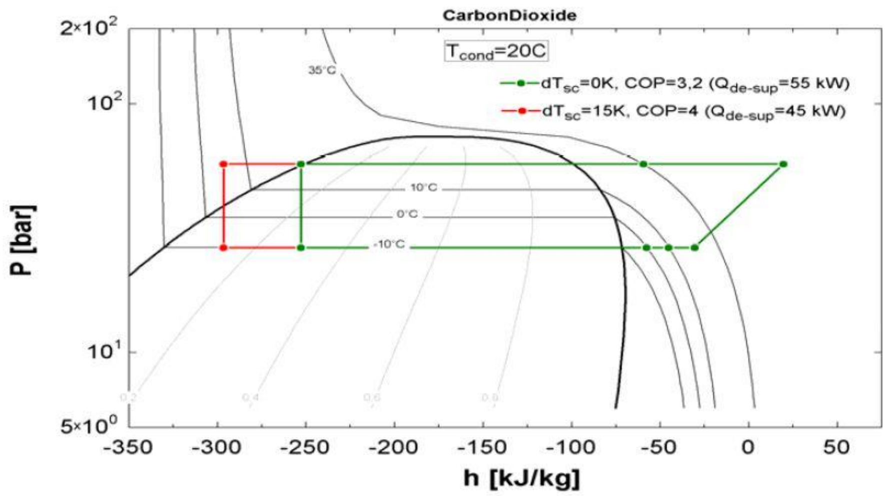

Figure 3.3.1.1: TR2 cycle with sub-cooling represented on a pressure-enthalpy diagram [2]

In figure 3.3.1.2, COP and de-superheater heat capacity deviance for NH3/CO2 cascade system and TR2 is represented as a function of the sub-cooling difference of temperature with a condensing temperature of 20°C. This figure shows the enthalpy difference of condensation increases with sub-cooling but the enthalpy difference of superheating stays identical. Therefore the lower mass flow with sub-cooling makes the superheat capacity lower.

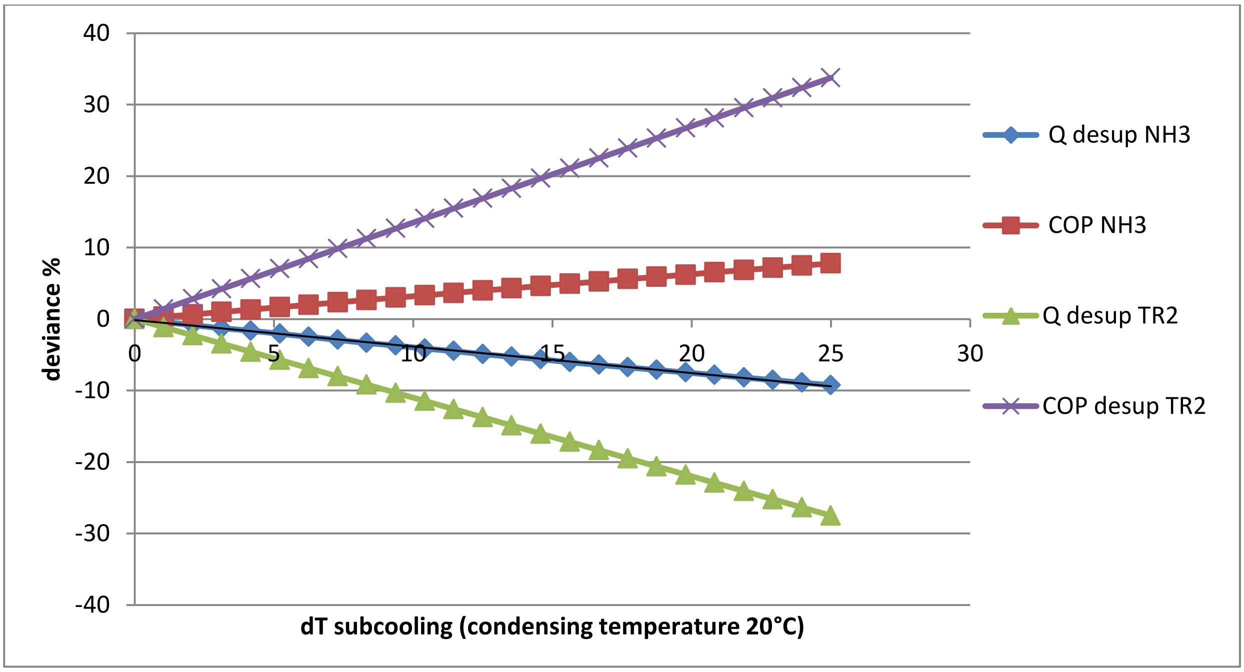

Figure 3.3.1.2: COP and de-superheater heat capacity deviance according to the sub-cooling difference

of temperature with a condensing temperature of 20°C

According to figure 3.3.1.2, for both refrigeration systems, sub-cooling has a positive influence on the system cooling COP, but a negative influence on the potential heat capacity which could be extracted by a de-superheater. Those influences have a bigger impact on CO2 system than on NH_3/CO_2 system. Figure 3.3.1.3 represents the COP of the medium temperature unit for TR2 system with and without sub-cooling as a function of the heat recovery ratio. For CO_2 system such as TR2, the positive influence predominates at a certain heating demand [2]. For instance, with gas cooler exit temperature at 5°C and a return temperature at 35°C for a heat recovery ratio above 115%, the cooling COP becomes lower with sub-cooling than without. [2]

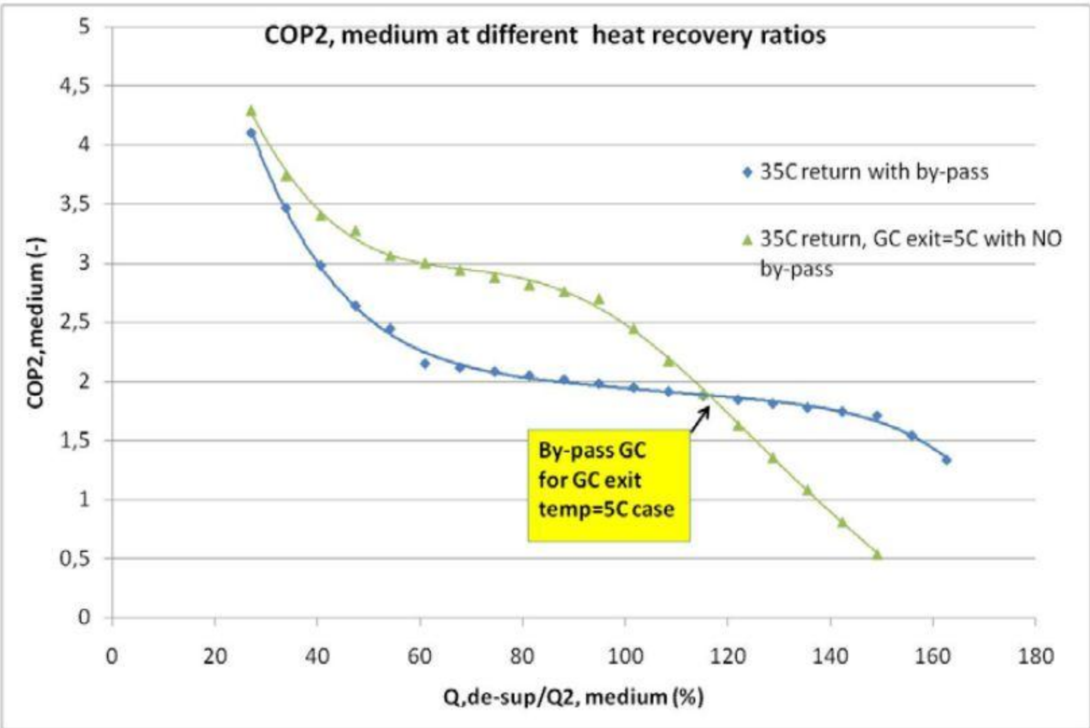

Figure 3.3.1.3: COP of the medium temperature unit for TR2 system as a function of heat recovery ratio [2]

In figure 3.3.1.4, the COP of the medium temperature unit for NH_3/CO_2 cascade system, without sub-cooling and with sub-cooler exit temperature of 5°C as in figure 3.3.1.3 for TR2, is represented as a function of heat recovery ratio As figure 3.3.1.4 shows, concerning NH_3/CO_2 cascade system, the cooling medium COP are slightly and constantly better with sub-cooling. Thus the positive influence of sub-cooling is always prominent but the COP improvement is not really important.

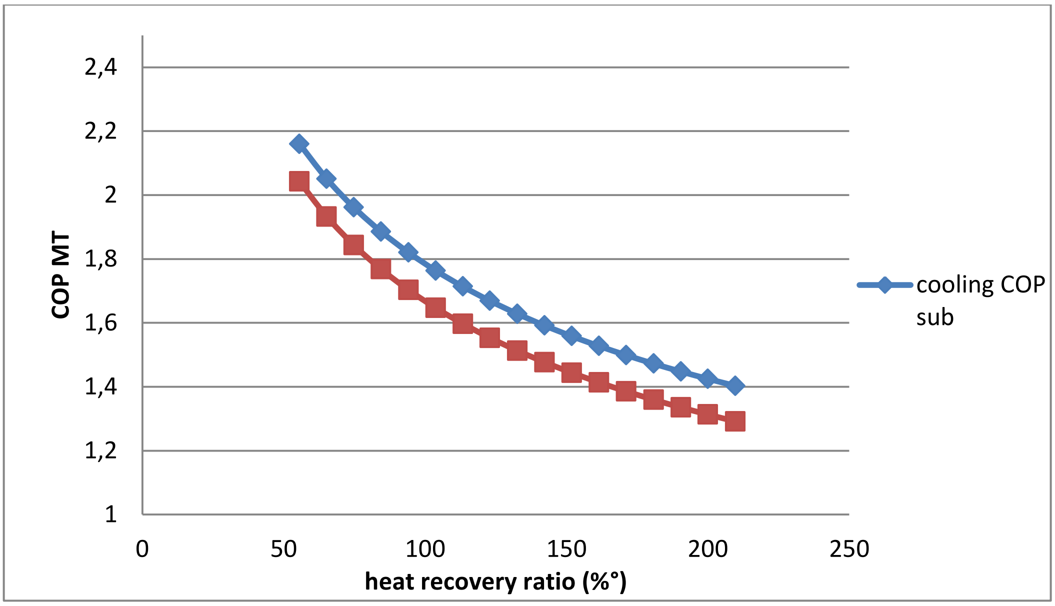

Figure 3.3.1.4: COP of the medium temperature unit for NH_3/CO_2 cascade as a function of heat recovery ratio

3.3.2 External superheating

Due to the distance between the refrigerating system machine room and the food storage cabins, superheating occurs through the pipes connecting the two units. This superheating leads generally to a reduction of COP. Figure 3.3.2.1 shows the low temperature unit COP deviance depending on condensing temperature with 5K, 10K, 15K and 20K of external superheat temperature difference.

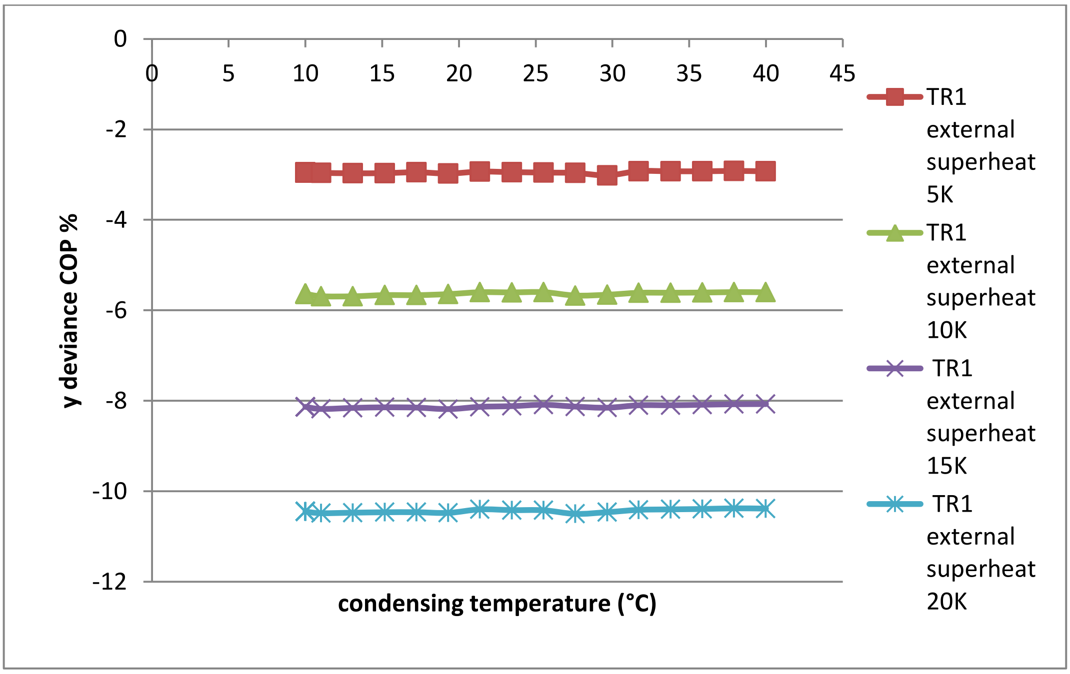

Figure 3.3.2.1: low temperature COP deviance as a function of the condensing temperature

As figure 3.3.2.1 shows, more the temperature difference corresponding to the superheat gained by the system through the pipes is high, more the COP decreases, for a temperature difference of 20K, and the COP is estimated to be about 10% for TR1. The other systems present approximately the same behavior concerning the influence of external superheating (Annexes: Figure 8.1).

External superheating occurs on the evaporating side of refrigeration system thermodynamic cycle, therefore it has an influence on cooling COP, but it should not have an influence on heating COP, since the heat is recovered from the condensing side and since there is no reason that the external superheating which occurs in the pipes connecting the compressor and the freezer is different in floating condensing and heat recovery modes.

3.3.3 Pressure discharge

Compressor efficiency, thus system COP as well, depends on the pressure ratio. Since the evaporation pressure is fixed, it is mostly the discharge pressure which has an influence on the COP. Due to their different properties; refrigerants present different discharge pressures at identical condensing temperature. Therefore it is necessary to analyze the impact of the discharge pressure on systems in the heat recovery mode. In figure 3.3.3.1, the COP of the medium temperature unit and discharge pressure depending on the heat recovery ratio are drawn for NH_3/CO_2 cascade with de-superheater and without subcooling (DSH+no sub).

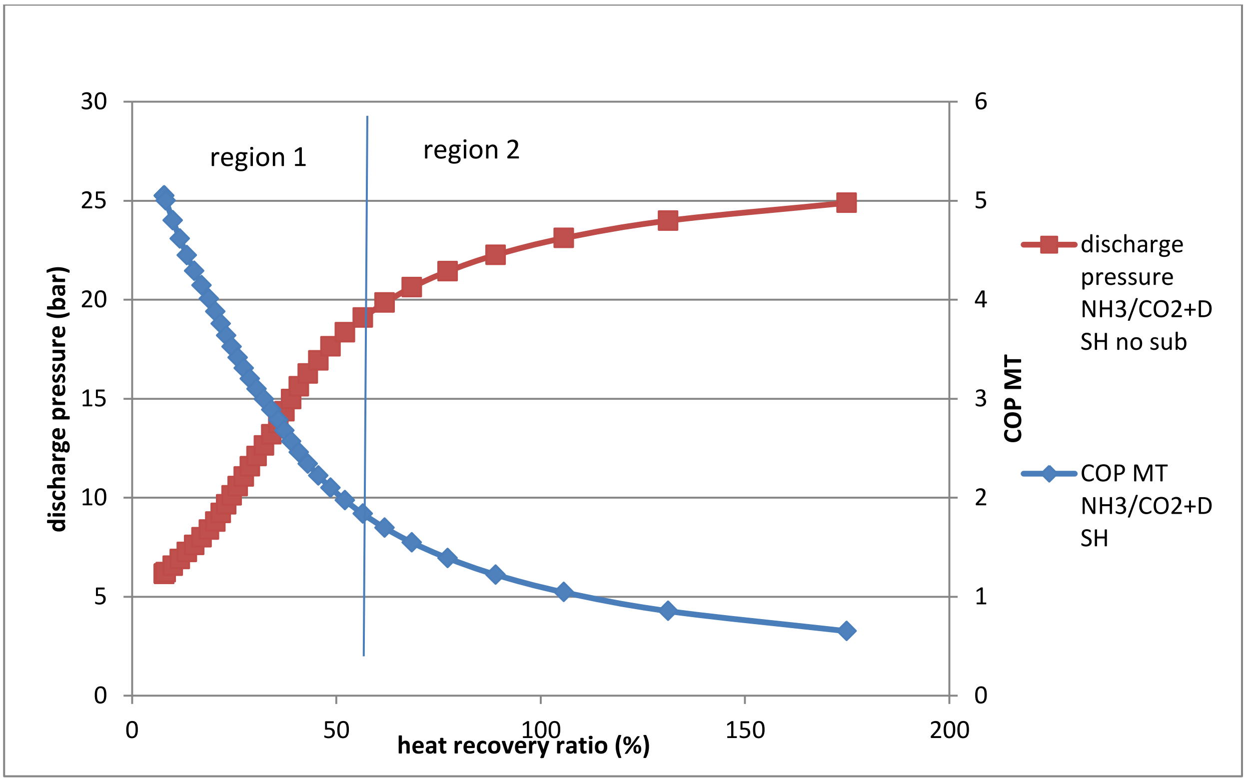

Figure 3.3.3.1: COP of the medium temperature unit and discharge pressure as a function of the heat recovery ratio for NH_3/CO_2 cascade with de-superheater

According to figure 3.3.3.1, for NH_3/CO_2 cascade system, the COP decreases when the heat recovery ratio increases whereas the discharge pressure increases when the heat recovery ratio increases. Two regions can be distinguished. The first is concerning the discharge pressure and medium COP for a heat recovery ratio below 60%. In this region, the discharge pressure has to be extensively increased in order to enhance the heat recovery ratio. In the second region the opposite is observed, a slight increase in discharge pressure leads to a considerable increase in heat recovery ratio. Nevertheless in this region, the COP is much lower.
Figure 3.3.3.2 represents the medium COP and discharge pressure without sub-cooling profiles depending on the heat recovery ratio for TR2.

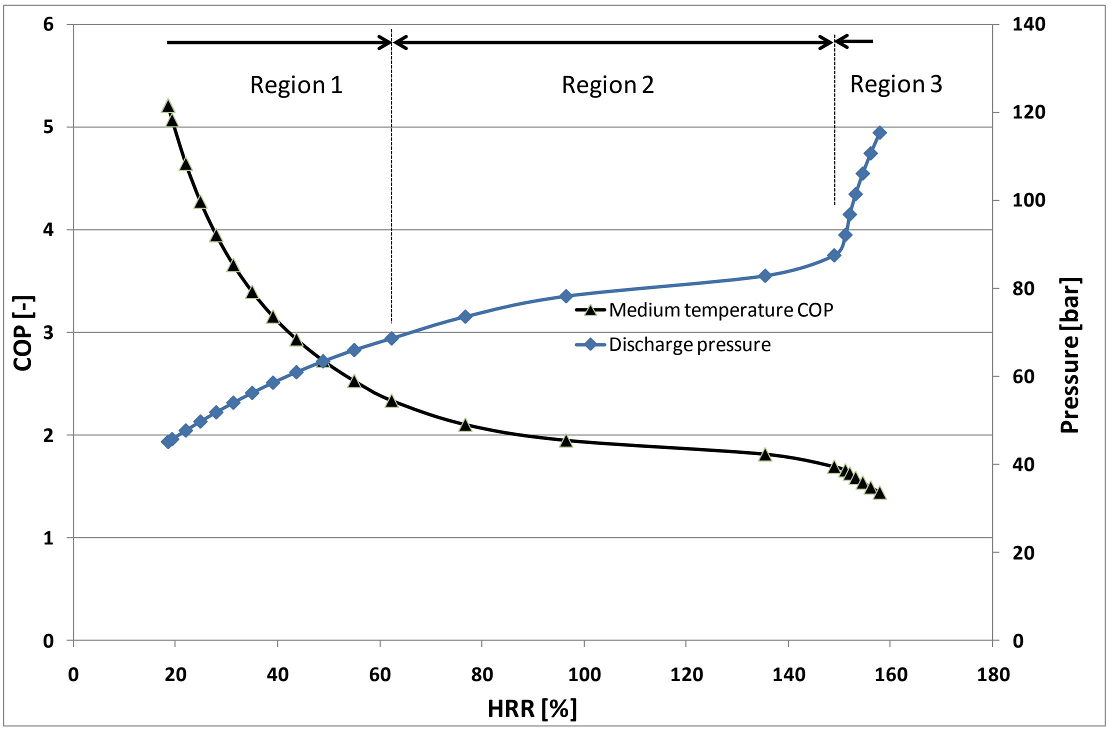

Figure 3.3.3.2: COP of the medium temperature unit and discharge pressure without sub-cooling as a function of the heat recovery ratio for TR2 [2]

According to figure 3.3.3.2, three regions can be observed for TR2. The discharge pressure evolution for the two first regions is approximately similar to the one of ammonia. The difference is that in the first region the system is running at sub-critical condition whereas in the second region the system is running at trans-critical condition. In the third region, the discharge pressure rises considerably with the heat recovery ratio. This sharp increases could be explained the fact that the CO_2 isotherm temperature at the exit of the de-superheater starts to be steep [9].

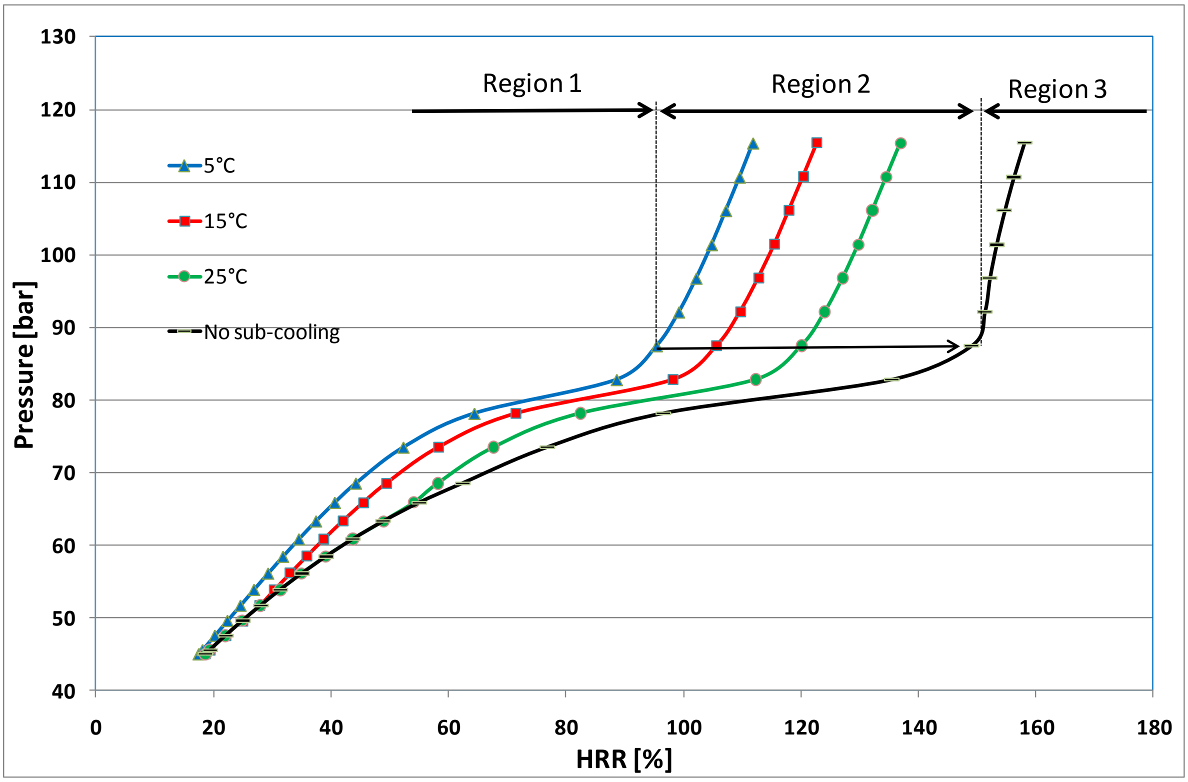

Figure 3.3.3.3: discharge pressure as a function of the heat recovery ratio for TR2 and for different sub-cooling [2]

To conclude whatever the refrigerant, an enhancement of the discharge pressure has a positive impact on the heat recovery ratio but a negative impact on the cooling COP. For CO_2 system, the negative effect of a discharge pressure increase can be compensated by sub-cooling the condensing outlet since as it has been state previously sub-cooling tends to improve the system performance. According to figure 3.3.3.3, a relevant way to process could be to apply sub-cooling when a heat recovery ratio below 90% is needed, and for discharge pressures beyond 88 bar the operation should follow the arrow which concretely means that from the beginning of the arrow the fan speed of the gas cooler should be reduced to allow more heat to be available for recovery through the de-superheater. At the end of the arrow, at about 150% of heat recovery ratio, the gas cooler has to be switched off or by passed and all heat rejected through the de-superheater [9]. However beyond this point, the COP drop has also to be considered in order to check the feasibility of a further discharge pressure increase.

In figure 3.3.3.4, the discharge pressure NH_3/CO_2 cascade and for different sub-cooler exit temperatures 5°C, 15°C and 25 °C is calculated depending on the heat recovery ratio. Concerning NH_3/CO_2 cascade system, as it has been set before; sub-cooling has a neglected impact on the COP and according to figure 3.3.3.4 its impact on the discharge pressure profile can be neglected as well. In addition for an identical heat recovery ratio, Ammonia discharge pressure is globally much lower than the one of CO_2.

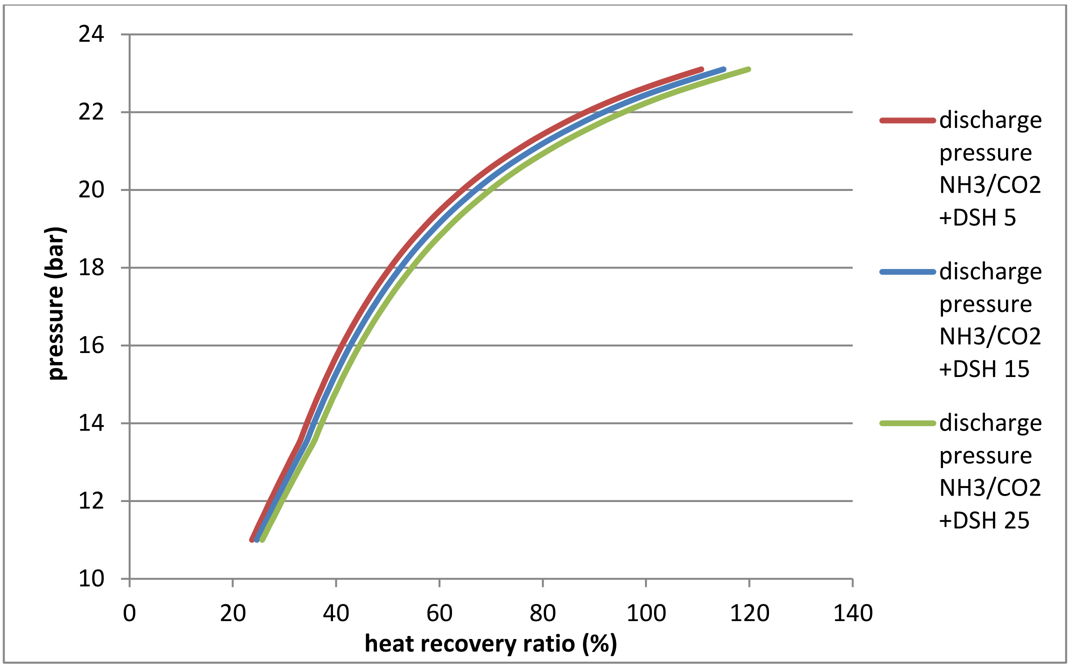

Figure 3.3.3.4: discharge pressure as a function of the heat recovery ratio for NH_3/CO_2 cascade and for different sub-cooling

3.4 annual electrical power use

By using simulation, refrigeration system annual energy consumption can be performed. For the calculations, an average size supermarket in Sweden in the climate of Stockholm is considered. In this part the systems considered are the systems described in part 3.1.2 with the assumptions set in part 3.1.1. In floating condensing mode, the heating is ensured by an external heating system with a COP assumed to be 1,8 , this value is based on the ratio of the prices of electricity and heating [2]. In figure 3.4.1, annual power consumption of TR1, conventional U.S R404A system, conventional Swedish R404A system and NH_3/CO_2 cascade system are compared in floating condensing mode with an external heating system and in heat recovery mode with fixed head pressure system and heat pump cascade connected to the condenser.

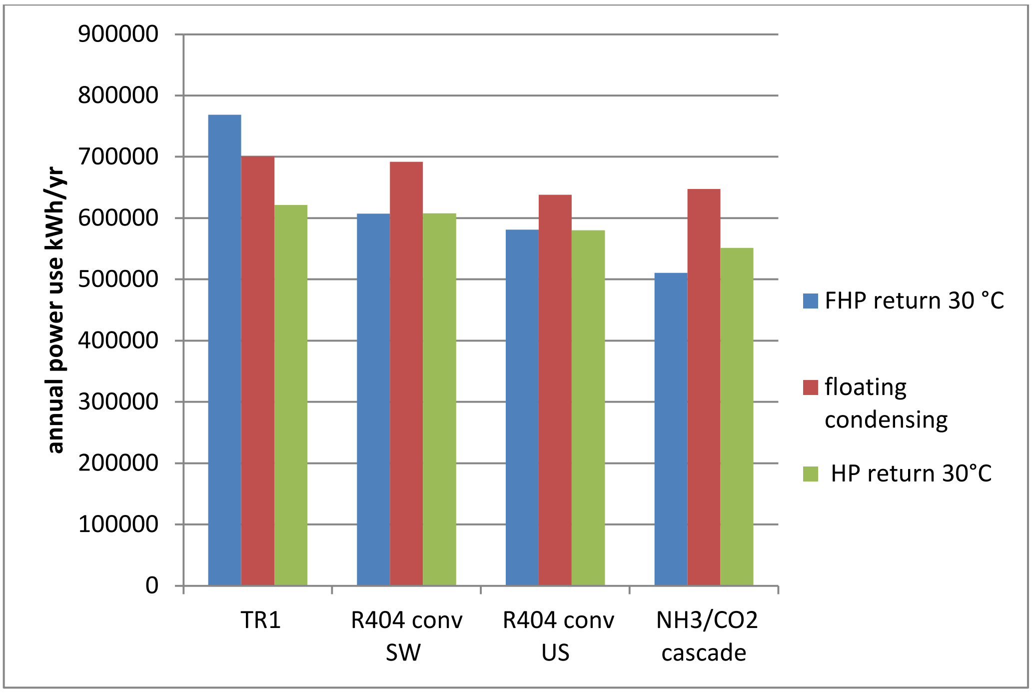

Figure 3.4.1: comparison of annual power uses between systems in floating condensing mode with external heating system, systems using fixed head pressure and systems using heat pump cascade connected to the condenser

According to figure 3.4.1, using fixed head pressure as heat recovery system for TR1 leads to a higher annual energy consumption than using an external heating system and operating in floating condensing mode. This is due to the fact that CO_2 system used to have a low COP at high condensing temperature/gas cooler exit temperature. By using a heat pump cascade connected to the condenser, TR1 operates at lower condensing temperature and presents an annual energy consumption lower than in floating condensing mode with an external heating system.

Concerning R404A refrigeration systems the addition of a heat pump cascade connected to the condenser does not lead to any decrease of annual power use (figure 3.4.1). Actually the required power input of the heat pump for those systems compensates exactly the system power reduction implied by a lower condensing system temperature. Thus a heat pump with a higher COP than 3,5 should be considered to entail improvement for such systems.

In the literature, it has been shown [4] that ammonia systems tend to have a rather good COP, as shows figure 3.4.1; FHP NH_3/CO_2 system presents the lowest annual energy consumption. According to fig below the simulation sets that the annual energy consumption is higher with the addition of a heat pump. This is actually due to the fact that FHP NH_3/CO_2 COP is about 4 in heat recovery mode, which is higher than the {heat pump+ NH_3/CO_2 system} COP.

In figure 3.4.2 the annual power consumptions between systems with fixed head pressure and de-superheater are compared for TR2 and NH_3/CO_2 cascade. As figure this shows, TR2 system with de-superheater has lower annual energy consumption than with fixed head pressure since the system is running at lower condensing/gas cooling pressure with a de-superheater as heat recovery system. The opposite it is observed for NH_3/CO_2 system, but as this is stated in part 4.2, de-superheater system fits more for partial heating supply assisted by an external heating system; in figure 3.4.2 the de-superheater provides 100% of heating required by an average supermarket. It can also be noticed that NH_3/CO_2 system annual energy consumption is lower than TR2 with de-superheater and fixed head pressure as heat recovery systems.

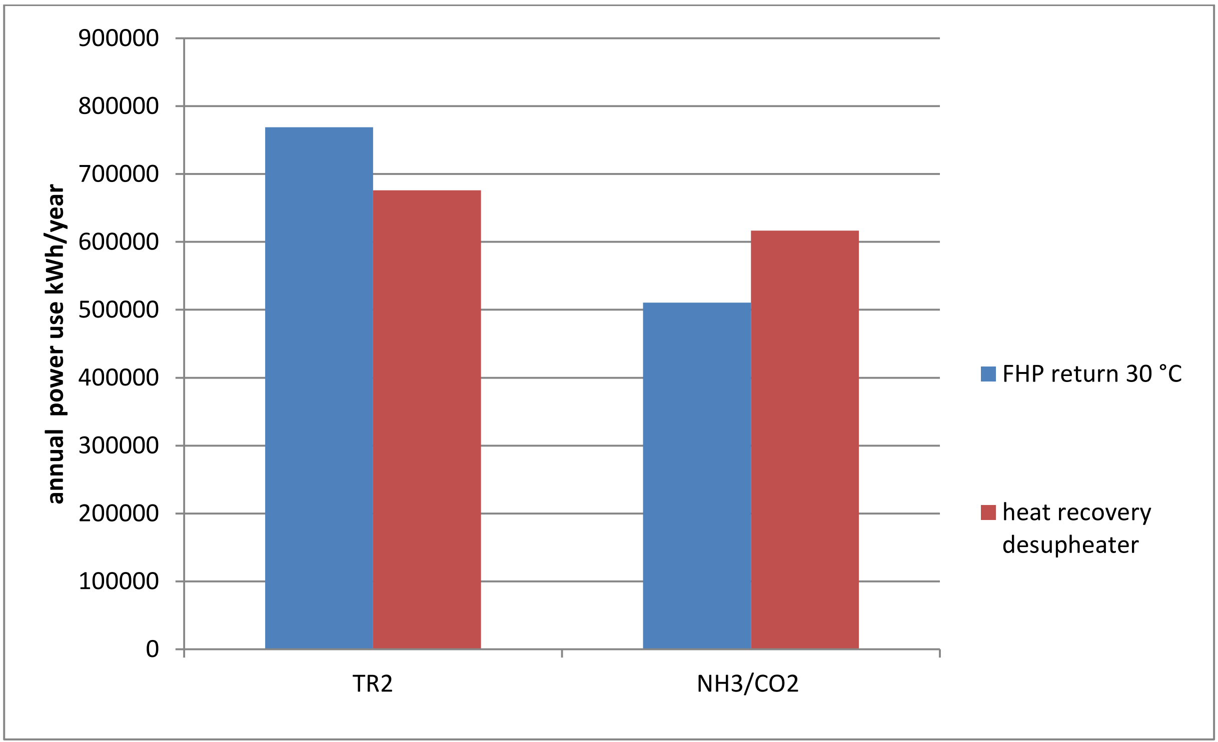

Figure 3.4.2: comparison of annual power uses between systems with fixed head pressure and de-superheater

Figure 3.4.3 represents the annual power use if TR2 with de-superheater systems without sub-cooling and with a sub-cooler exit temperature of 5°C. As stated in part 3.3.1, sub-cooling has a positive impact on system COP for CO_2 systems. Therefore the annual power becomes lower when sub-cooling is performed until 5°C.

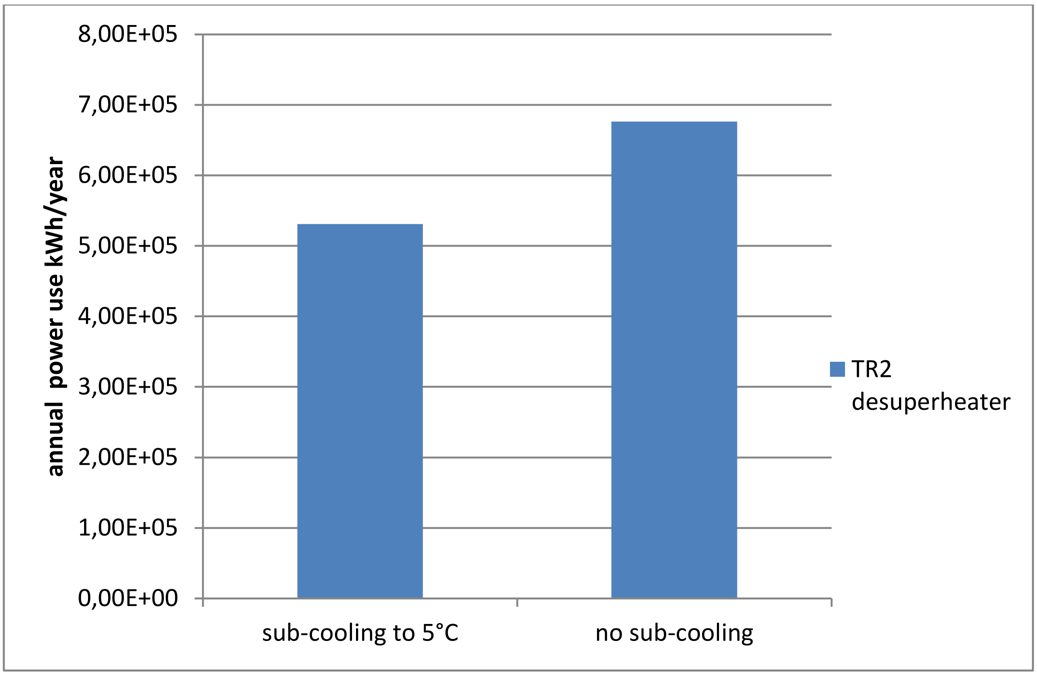

Figure 3.4.3: comparison of annual power uses between systems with fixed head pressure and de-superheater

Figure 3.4.4 compares the annual energy consumption of each different recovery systems combined with a CO_2 system. Thus CO_2 system with heat pump for sub-cooling system is the most energy efficient system, in second position comes CO_2 system with heat pump connected to the condenser. Those two systems have actually the lowest operating condensing temperatures and pressures. Nevertheless CO_2 system with de-superheater could eventually have lower annual energy consumption if combined to an external heating system as it has been set previously for NH_3/CO_2 system. It can be noticed that the difference between the annual energy consumption of the CO_2 system with heat pump connected to the condenser and the one of the CO_2 system with heat pump connected to the sub-cooler is slight. This could be explained by the fact that for TR2 HPSC the heat pump required more input electrical power due to the fact that the system is running at the minimum condensing temperature pressure. The condensation/evaporation pressure gap into the heat pump connected the sub-cooler is higher since the heat pump evaporation occurs at lower temperature than the one of TR1 HP.

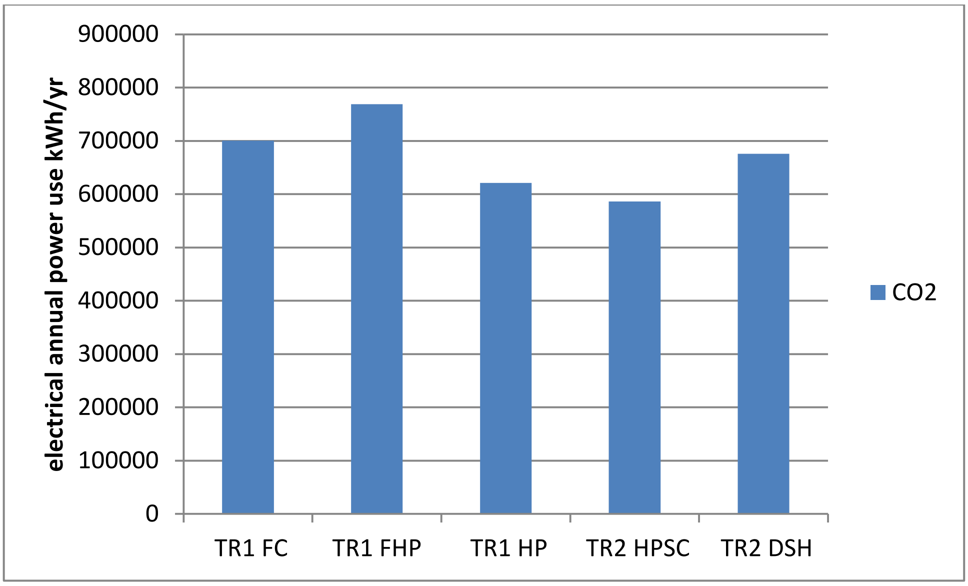

Figure 3.4.4: comparison of annual power uses between CO_2 systems

3.5 annual greenhouse gas emission

For refrigeration systems, two different types of emission have to be considered: the direct emission which is directly leaked by the system to the environment and the indirect emission which is linked to the power consumption of the system.

From the annual power consumption, the annual indirect emission can be evaluated by using the following formula [10]:

$$GHG_{\,indirect\ emission} = F \times E_{annual} \quad (3\text{-}4)$$

The GHG emission is calculated in equivalent of CO_2 mass unit and equal to the multiplication of the annual power consumption and F the emission factor of the country which depends on its energy production facilities (nuclear plant, coal combustion, waste gasification etc). For example the emission factor of Sweden is 0,023 whereas the one of US is 0, 22, this difference is due to the fact that Sweden used much more renewable energy source than US [10].

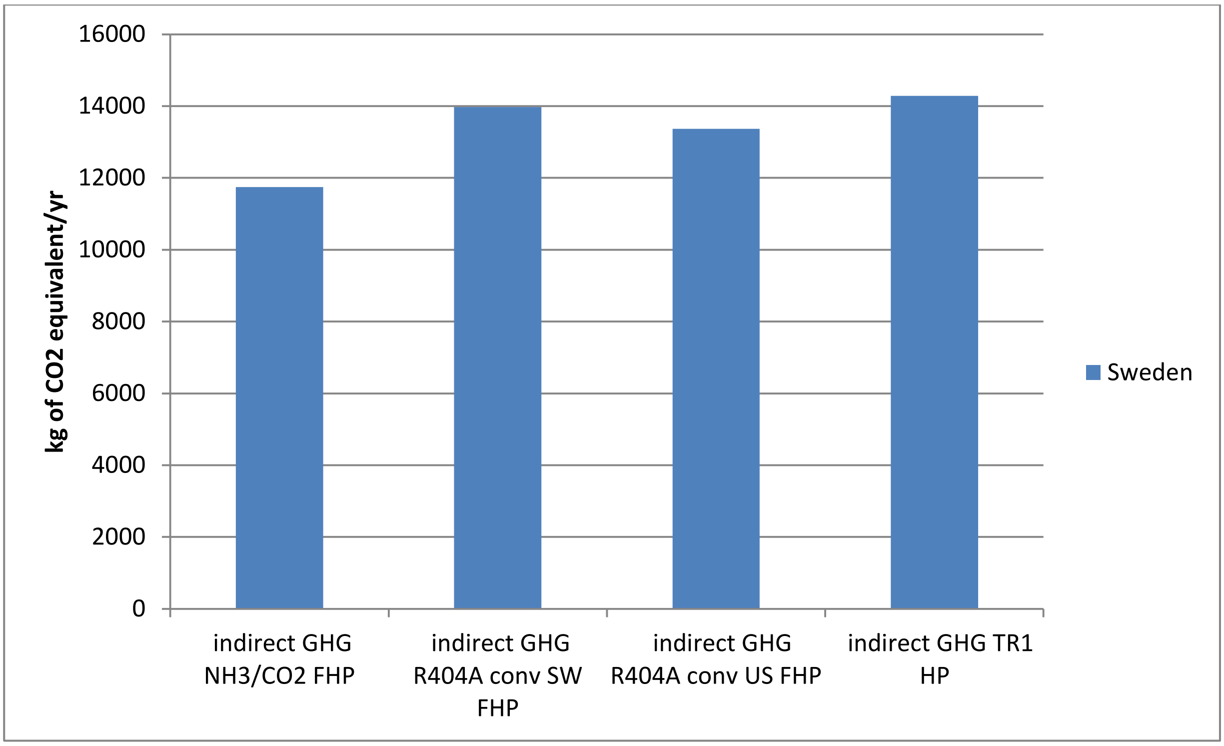

Figure 3.5.1: kg of CO2 equivalent of indirect emissions of systems with fixed head pressure and TR1 with heat pump cascade

In figure 3.5.1, the indirect GHG of systems with fixed head pressure and TR1 with heat pump cascade are compared. The indirect greenhouse gas emissions are proportional to the annual power consumption; the difference between systems is relatively slight. In average indirect emission of refrigeration systems is 14 tons per year.

Nevertheless, direct emissions are quite different from a system to another, they do depend on system refrigerant since the GWP of R404A is 3800 and the one of CO_2 is 1.

The direct emissions are estimated with the following formula [10]:

$$GHG_{direct\ emission} = R.L.GWP \quad (3\text{-}5)$$

R is the system annual recharge in kg, L is the annual leakage rate, the product R.L is estimated to be 15kg/yr for average size supermarket in Sweden with refrigeration system using artificial refrigerant [1]. The value of R.L is unknown for natural refrigerant but since the annual recharge is usually about 120kg for average size Swedish supermarket. Thus it can be assumed that the direct emission of refrigeration system using natural refrigerant is less than 120kg.

	TR1	R404A US	R404A SW	NH3/CO2
direct GHG kg-e CO2/yr	**>120**	**~100000**	**~50000**	**>>120**

Table 3.5: direct emission in kg of equivalent CO2 of the different system solutions

The amount of direct GHG emission is definitely more important for artificial refrigerant systems than natural refrigerant systems (table 3.5). Thus the regulation about artificial refrigerant restriction appears justified.

3.6 Summary

In Sweden, refrigerant charge is reduced by using R404A indirect systems but those systems are rather less efficient than other systems, not only direct artificial refrigerant systems but also systems using natural refrigerants. Trans-critical CO_2 systems, such as TR1, have more competitive efficiency in heat recovery mode with addition of heat pump cascade connected to the condenser or a sub-cooler. NH3/CO2 cascade systems have generally rather good cooling and heating COP with all type of heat recovery systems.

Several parameters have an influence on system COP, sub-cooling tends to increase the COP of CO_2 and R404A systems, but its influence can be neglected for NH_3/CO_2 cascade systems. The control of discharge pressure is primary for trans-critical system since around the critical point, a slight change of pressure can leads to a steep change of performance and amount of available heat recovered. External superheating usually decreases the COP. From the simulation, it appears that external superheating has a constant negative impact not depending on the condensing temperature (Annexes: figure 8.1).

If the power consumption looks approximately in the same range concerning the systems studied, the GHG emission, especially direct, seems sharply higher for R404A refrigeration systems (table 3.5). According to the current global issues, the reduction of artificial refrigerant appears to be more than suitable.

4. System analysis

4.1 Model validation

The conclusions state in the previous part are based on computer models, therefore it is necessary to compare the results obtained by field measurements in order to validate models and conclusions.

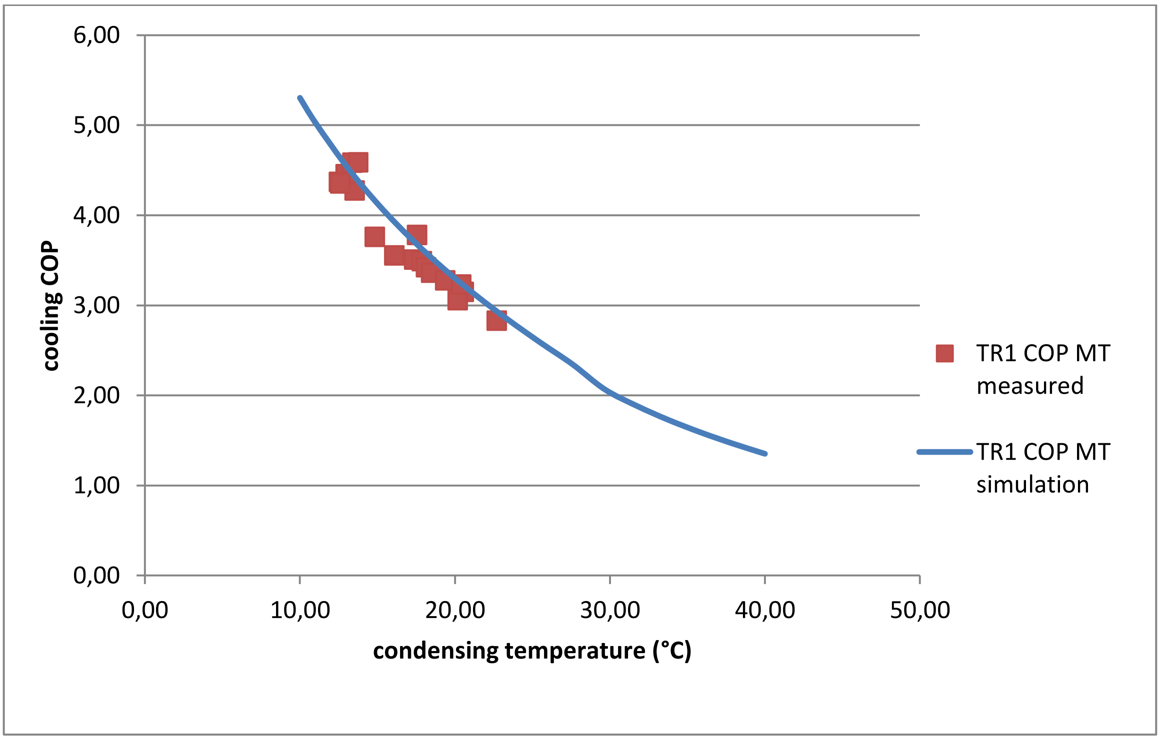

Figure 4.1.1: comparison between medium temperature COP based on experimental data and calculated by simulation for TR1

In figure 4.1.1, the experimental evaluation of TR1 COP coincides with the COP calculated with simulation. Nevertheless it is not possible to check the model for a condensing temperature above 20°C due to the supermarket activity. This is not really a problem since it seems that TR1 is running at condensing temperatures which give optimal COP. Thus the model can be validated for condensing temperatures between 12-20°C, the extrapolation of the model could be eventually assumed according to the continuous evolution of the curve path.

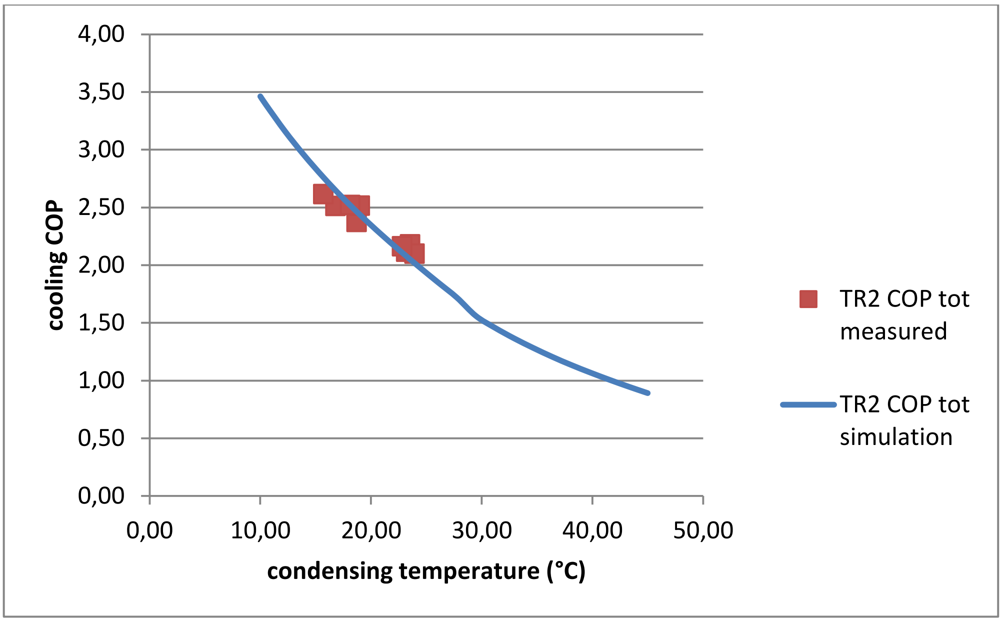

Figure 4.1.2: comparison between total COP based on experimental data and calculated by simulation for TR2

The same observations can be also stipulated for TR2, the COP measured fits to the model simulated. However according to figure 4.1.3, the COP of the conventional Swedish refrigeration system obtained by measurement differ from the one calculated from the model. It can be noticed that different experimental series give different COP values and the path of curves from simulation and experiments follow the same evolution. The model might not be wrong, other parameters not well known as sub-cooling, external superheat and brine loss could have interfered. The accuracy of the measurement device could also explain this difference between the experimental results.

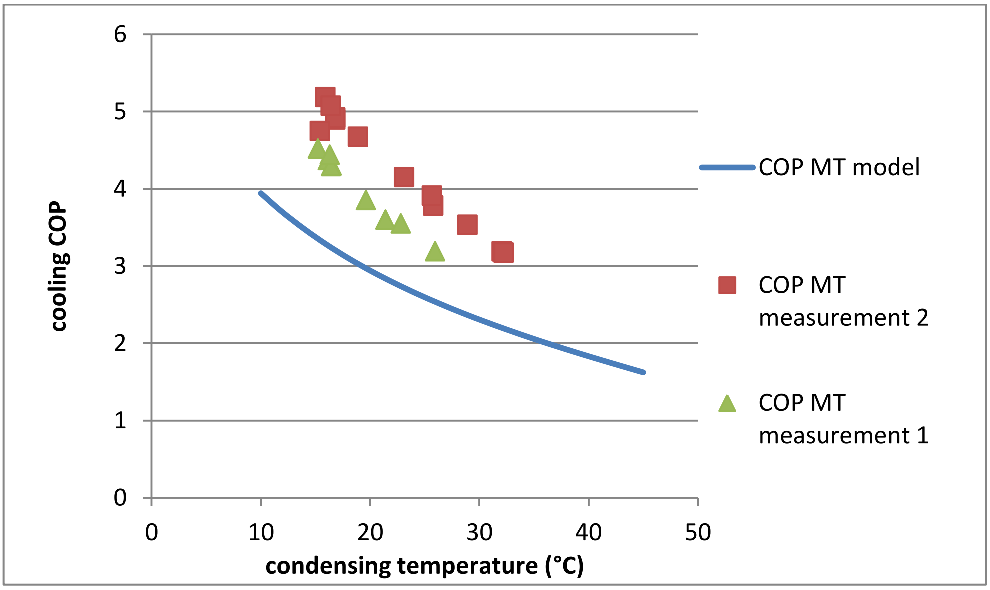

Figure 4.1.3: comparison between medium temperature COP based on experimental data and calculated by simulation for R404A conventional system

By decreasing the brine losses and electrical pump power to 4% of the medium cooling capacity and adding 8°C of sub-cooling from medium temperature unit to the low temperature unit, the model curve in figure 4.1.4 is obtained. With those parameters the model results fit more to the field measurements.

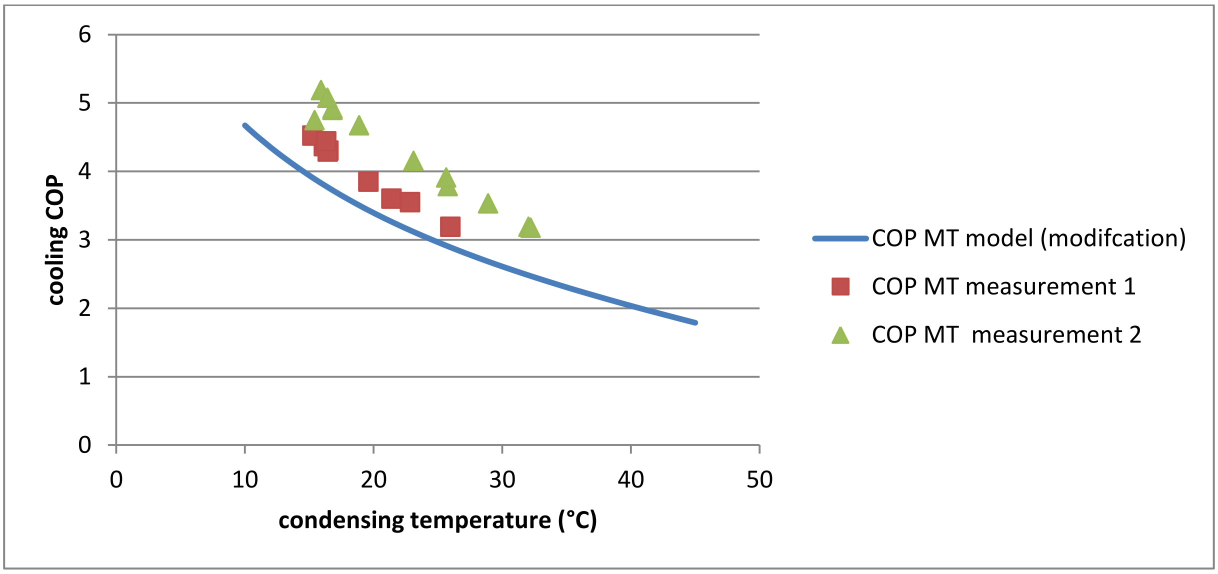

Figure 4.1.4: comparison after adjustments between medium temperature COP based on experimental data and calculated by simulation for R404A conventional system

Measurements cannot be performed for NH_3/CO_2 cascade system and the US. Since the models were build the same way as TR1, TR2 and the conventional Swedish systems, it can be assumes than those models provide coherent results as well. In addition by evaluating the annual TR1 energy consumption with the monthly real total COP based on field data and by using the refrigeration system simulation software Pack II, almost similar annual energy consumptions are obtained as figure 4.1.5 shows. It can be noticed that the slight difference observed for Pack II estimation may be due to the fact that the simulation is running with different data concerning the outside temperature. Therefore, it can be assumed from figure 4.1.5 that one simulation code is better than the other one but it can be assumed that all models provide realistic results.

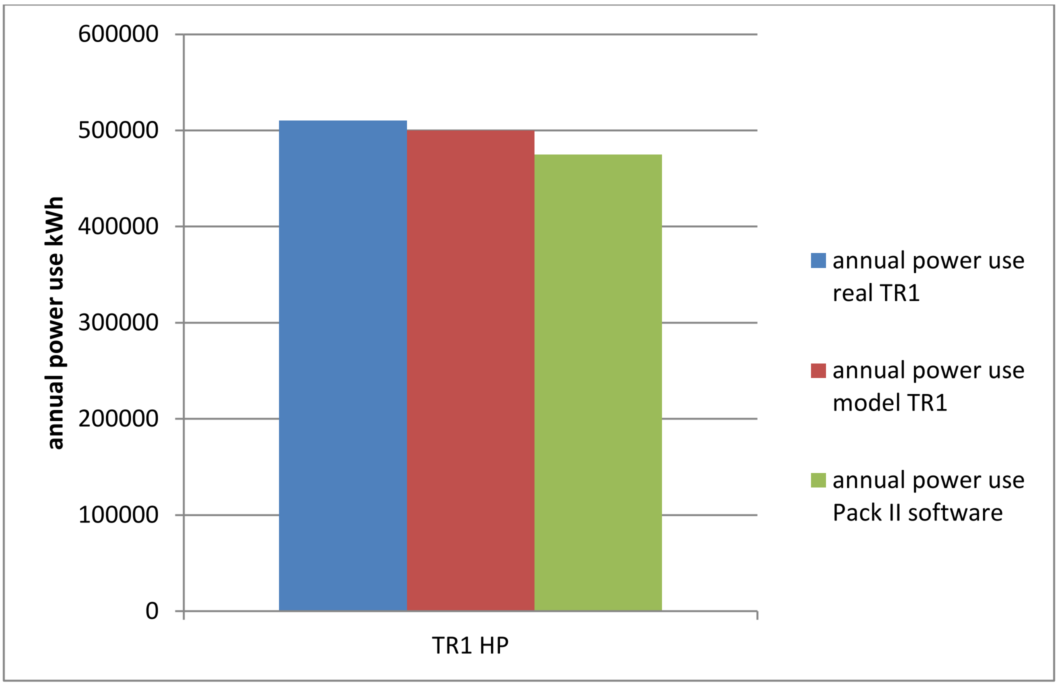

Figure 4.1.5: comparison between TR1 annual power used based on experimental data and calculated by EES and Pack II simulation

4.2 System optimization

4.2.1 Problem

In all models established previously, the condensing temperatures of chiller and freezer are increased at the same temperature level in order to fulfill the necessary heating demand. However this might not be the lowest energy consuming way to process. A further increase of the condensing temperature in one unit combined to a decrease of the condensing in the other unit could eventually give better COP. Since the multiple different factors are getting involved in refrigeration process, the optimization of parameters could not be performed easily by simple optimization methods. The use of a genetic algorithm is thus relevant. Genetic algorithm permits to solve problems with a large amount of parameters and constraints.

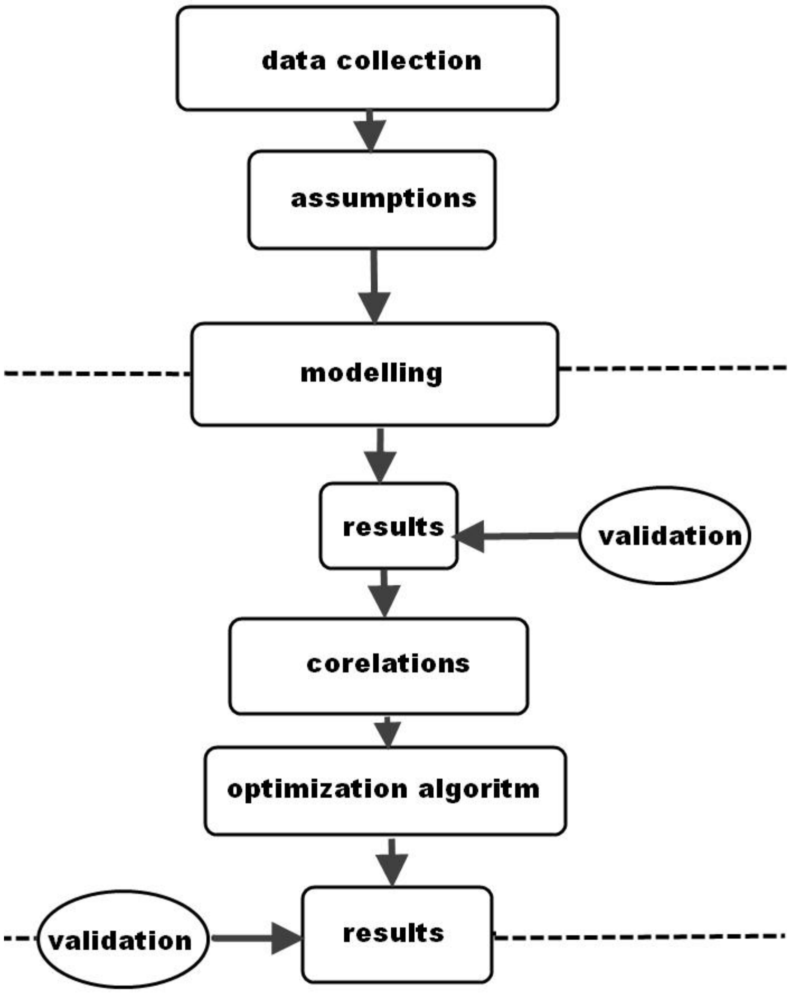

Figure 4.2.1.1: study methodology diagram

In previous studies performed by KTH-Energy Technology department, data collection, assumptions and models have been carried out. By adjusting models, results have been obtained and validated by experimental results during this study. From these results, correlations depending on condensing temperature which describes supermarket refrigeration system behavior have been established: Therefore refrigeration systems can be pictured in simpler way by an equation system. An algorithm such genetic algorithm can be used to solve this equation system in order to give an optimized solution. For this problem, the parameter which has to be optimized is the condensing temperature.(figure 4.2.1.1)

4.2.2 Algorithm description

Genetic algorithm is an optimization method based on the natural evolution principles, such as reproduction, mutation, crossover and selection. This method is used for problem with a large amount of parameters involved [11]. A genetic algorithm is initialized by the creation of an ensemble of series of parameters which potentially are the solutions of the problem studied. Each parameter series is assimilated to a chromosome and each parameter to a gene (figure 4.2.2.1).

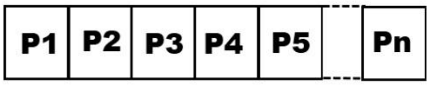

Figure 4.2.2.1: representation of parameter series (chromosome)

In a first time the solution from each chromosome is evaluated. This evaluation permits to rank the chromosome according to their pertinence. The chromosomes which correspond to the best solutions are conserved and the other one are eliminated. From the best chromosome a new generation of chromosome is bred. During the new generation production the phenomena of gene cross over and gene mutation interfere which allow the algorithm to tend to a global optimum instead of a local one.

4.2.3 Model description

A refrigeration system constituted by a number of k units, as exposed in figure 4.2.3.1, is considered, those units are operating at low or medium condensing temperature. A pipe network connects the units to the outside and another one to HVAC system. The external input of the system is therefore the outside temperature and the system target is to provide the adequate inlet temperature and heat to the HVAC system in heat recovery mode. When heating is not needed, the air coolant is rejected to the outside by the pump system.

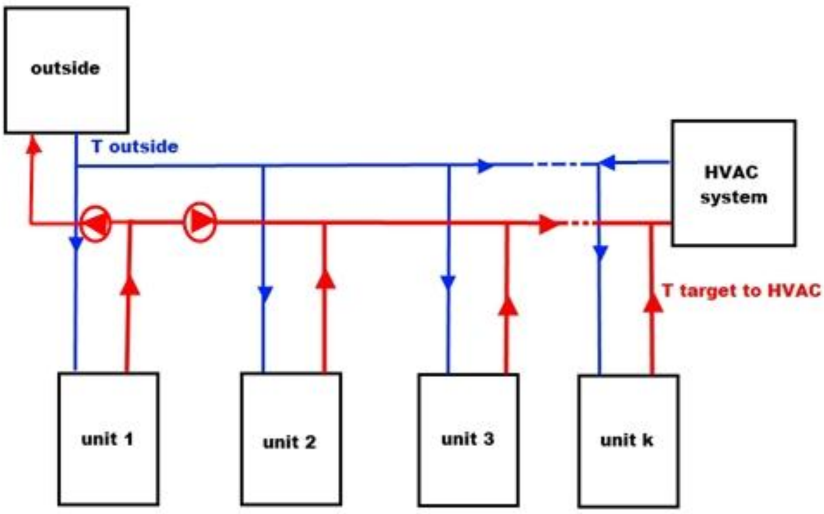

Figure 4.2.3.1: schematic of a multiple unit system

The units are divided in independent sub-systems characterized by a condensing temperature (T_cond), a condensing capacity (Q_cond), a cooling capacity (Q_evap), a mass flow (m), a coefficient of performance (COP) and compressor power consumption (E). The target temperature which has been to provide to the HVAC system is calculated with the following formula:

$$T_{Target\ to\ HVAC} = \frac{\sum_{i=1}^{k} m_i . T_i}{\sum_{i}^{k} m_i} \quad (4\text{-}1)$$

Ti is the outlet temperature of the unit i, mi is the coolant mass flow necessary to ensure the condensation of the unit i. The target temperature supplied to the HVAC system depends on heat recovery system, if a heat pump is implemented into the system the target temperature is between 13-20°C according to the heat pump design, if not usually this temperature is about 35-40°C.

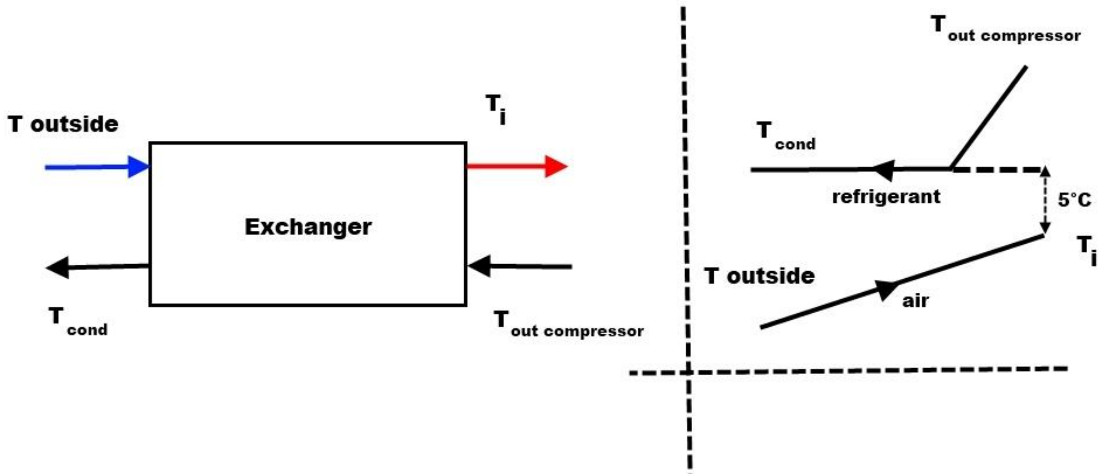

Figure 4.2.3.2: assumptions considered for the exchanger in the model

In the condenser or gas cooler, the difference temperature between the condensing temperature and the coolant outlet temperature is assumed to be 5°C and no sub-cooling occurs after the condensation(figure 4.2.3.2).

The system key parameters are described in the following table:

	Medium temperature	Low temperature
T_evap (C)	-10	-35
Q_evap (kW)	100	35
COP	0.0027*T_cond^2-0.2667*T_cond+7.6195	0.0007*T_cond^2-0.0806*T_cond+2.7605
Refrigerant Mass flow (kg/s)	0.0067*T_cond+0.2355	0.0015*T_cond+0.0506
External superheat (K)	15	15
Internal superheat (K)	10	10

Table 4.2.3: Assumptions and correlation taking in account for the system solution optimization. COP and mass flow correlations depending on the condensing temperature are based on a EES model with two separated low and medium units (TR1).

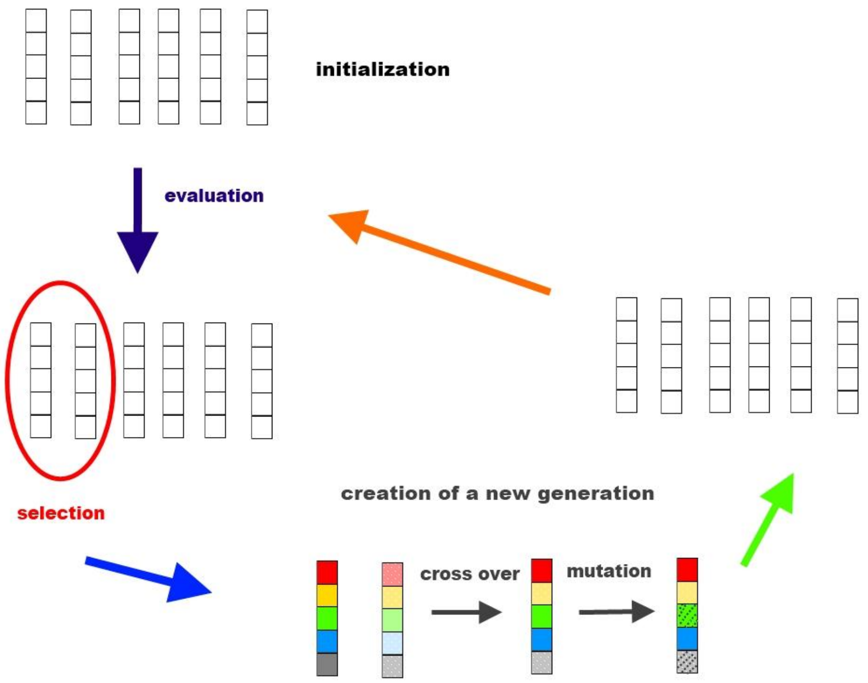

Figure 4.2.3.3: schematic of the steps of the genetic algorithm

A genetic algorithm adapted to the refrigeration model, as described in figure 4.2.3.3, has been programed in MATLAB. This algorithm is constituted by the following functions:

GA_cab_6_2: This is the main function, the number of medium and low temperature units, iteration, chromosome's number of the initial population and outside temperature have to be set in order to allow the start of the algorithm run. All functions implemented for the algorithm are called in this one by following the step order describes in figure 4.2.3.3. The solution of the algorithm is a vector which contains the condensing temperatures corresponding to the highest system COP possible according to the target temperature which is necessary to provide to the HVAC.

Initpop: this function creates the first generation of chromosome which the genes are the different system condensing temperatures. The condensing temperature values are generated randomly in order to obey to the following constraints:

$$10°C < T_{cond} < 50°C$$

$$Q_{heat\ recovered} = Q_{heating\ demand} \qquad (4\text{-}2)$$

$$T_{Target\ to\ HVAC} = \frac{\sum_{i=1}^{k} m_i . T_i}{\sum_i^k m_i}$$

Evalpop: this function calculates the solution of each chromosome and determines the two one which present the best total COP.

Copybest: this function duplicates the two best chromosomes from a generation i in to the generation i+1.

Croisement: this function performs with a probability of 0.3 a crossover of genes between one of the best chromosomes and a chromosome randomly choose by the function *choix_parents.*

Mutation: this function replaces with a probability of 0.3 gene value by another one generated randomly.

Check_newpop: this function checks after cross over and mutation if each chromosomes of the next generation follows the constraints' (*), if not a new chromosome is generated.

F_prob: this function calculates based on correlations determined with EES model, the coolant outlet temperature, the COP, energy consumption and heat of condensation of each unit.

4.2.4 Optimization results

The system optimized is at first constituted by one low temperature and one medium temperature unit. This system is successively studied for 13, 18, 25 and 35 C as target temperatures which have to be supplied to the HVAC system. The target temperature 13 and 18C implies the addition of heat pump in to the HVAC system, but the heat pump itself is not considered in the system.

Figure 4.2.4.1 shows the condensing temperatures of the low and medium units which give the best total COP. According to the optimization results the condensing temperature of the low temperature unit would have to be lowered as much as possible whereas the condensing temperature of the medium temperature unit would have to be increased enough to obtain the right target temperature provided to the HVAC system. Therefore a good way to regulate such two unit refrigeration system could be to control the condensing pressure and temperature of the medium unit in order to provide heat to the supermarket and in parallel let the low temperature unit running in floating condensing mode. This operating way is similar to the floating condensing and heat recovery system described in part 2.4 [3].

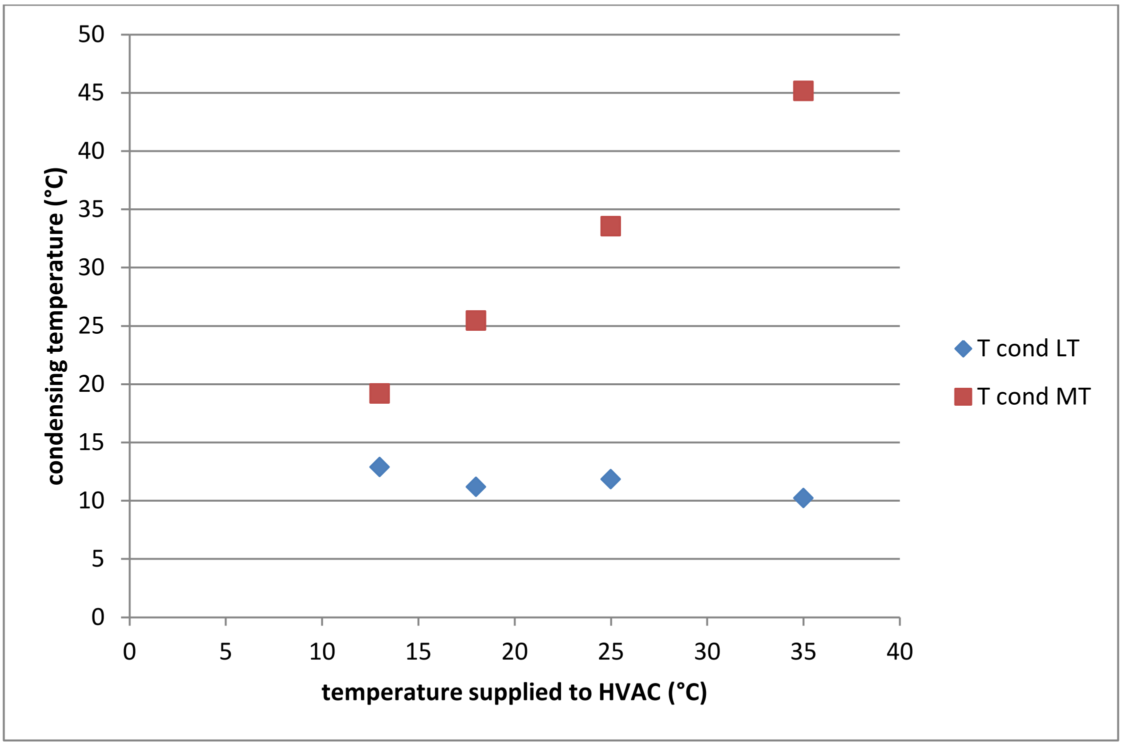

Figure 4.2.4.1 profiles of the condensing temperature of the medium and low temperature calculated by the optimization algorithm

In the following table, the total COP improvement between the optimized system and a system which operates at the same condensing temperature level for both units in heat recovery mode. The average of the potential improvement is estimated to be around 10%.

Temperature to HVAC system (°C)	Total COP improvement (%)
13	7,05
18	9,31
25	13,5
35	13,8

Table 4.2.4: Total COP improvement estimations

Usually instead of one huge refrigeration unit, several small refrigeration units are used in supermarket. Thus the distance between refrigeration cabins and refrigeration can be reduced. For CO2 system, multiple refrigeration system or compressors in series are necessary since single CO2 compressor cannot provide a cooling capacity more than 60kW [12].

According to figure 4.2.4.2, if Bitzer compressors are used for CO2 system, at the same evaporating temperature level, compressor efficiency remains similar for each size type of compressor. Therefore the use of multiple cabins has in theory no influence on the system total COP. The comparison with the optimization algorithm between the total COP of the simple two units systems and multiple unit system with 2 low temperature units and 3 medium temperature units confirms this assumption.

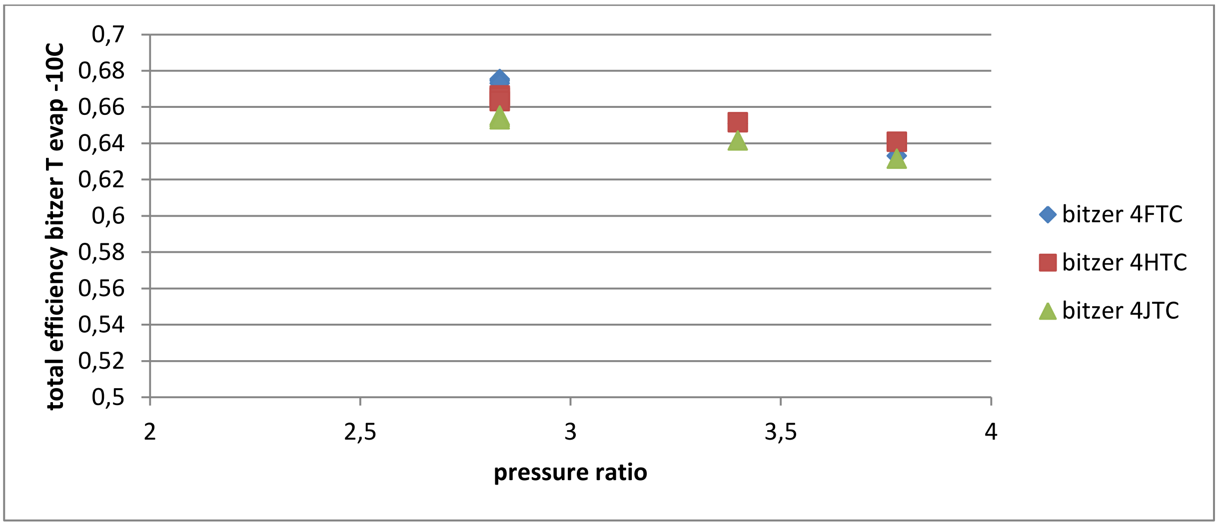

Figure 4.2.4.2: Bitzer CO_2 compressor total efficiency as function of the pressure ratio

5. Discussion

Due to global warming concern, the interest in the use of natural refrigerant significantly grows. Nevertheless systems using artificial refrigerant remain reliable and safe. In order to compete with them, systems using natural refrigerant have to be as efficient and safe as them. Since its very high pressure at relatively low temperature, CO2 systems needed in the past further developments in compressor technology, but nowadays the technology advancements allows CO2 systems to compete with other systems. NH3 systems, which used to be voluminous systems, require regulations less restrictive in term of safety in supermarket [4].

With the current technology level, this study has shown that CO_2 system could compete with HFC systems by integrating heat pump or de-superheater. The addition of heat pump connected to the condenser or sub-cooler estimated the annual energy consumption just slightly higher than conventional systems using artificial refrigerant and operating at fixed head pressure. With de-superheater simulations point out that acceptable COP can be obtained with the assistance of external heating systems or sub-cooling.

Since NH_3 systems generally require a high practical charge, NH_3/CO_2 cascade system permits to reduce this charge. Due to the usual high COP of NH_3 systems, NH_3/CO_2 system presents higher COP than other systems according to the simulations. The less energy consuming system in heat recovery mode is the fixe head pressure. Actually the total COP increase with the use of heat pump but the energy used by the heat pump is superior to the energy saving relating to the total COP enhancement.

Refrigeration systems have to be efficient, safe and harmless for the environment. From this study, it can be imply that R404A systems tend to be safe and efficient, NH3 systems efficient and harmless for the environment and CO2 systems safe and harmless for the environment. However, for NH3 and CO2 systems technology may able in the future to fix those system problems.

External superheating and sub-cooling have also an influence on the system performance. Since superheating through the pipes are losses to the ambient, external superheating tends to decrease the system performance. This negative influence is globally equivalent for all system solutions and depends mainly on the pipe network size. Thus it is coherent to fix the superheating at the same value for all system solutions studied. Concerning sub-cooling, it has been observed that sub-cooling increase significantly the performance and CO2 systems whereas the performance enhancement is negligible for NH3 system.

A way to improve refrigeration systems performances is also to regulate key parameters in order to fit perfectly the system requirements in heat recovery mode with the maximum efficiency. In this study a genetic algorithm has been coded to calculate the condensing temperatures which provide the best total COP for a multiple unit system. It has been observed that to get a better total COP, it is more efficient to maintain the condensing temperature of the low temperature unit remaining at low level and increase the condensing temperature of the medium temperature unit rather than increase all unit condensing temperatures at same level. The potential COP improvement has been estimated by simulation to be 10%. Nevertheless experimental results are necessary to validate this assumption.

6. Conclusion

In this thesis, conventional and new refrigeration system solutions have been analyzed through computer simulations. Unlike field data analysis, simulations permit more variation concerning input data, external conditions and control parameters. Thus the comparison of different system solutions is facilitated.

In term of performance, system solutions using natural refrigerant are able to compete with conventional solutions using artificial refrigerant. The limits of natural refrigerant systems are set by the current technology used. The multiplication of the use of such systems will certainly be followed by the development of technologies operating with natural refrigerant. It has also been pointed out that the control of the condensing temperature has clearly an influence on system performance.

In this study, it has been state by simulation that NH_3/CO_2 cascade system should be more efficient than R404A refrigeration system. In future projects, the feasibility of NH_3/CO_2 cascade system implementation in supermarket could be analyzed experimentally. This investigation could take a particular interest in the safety issues of NH_3 systems.

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8. Annexes

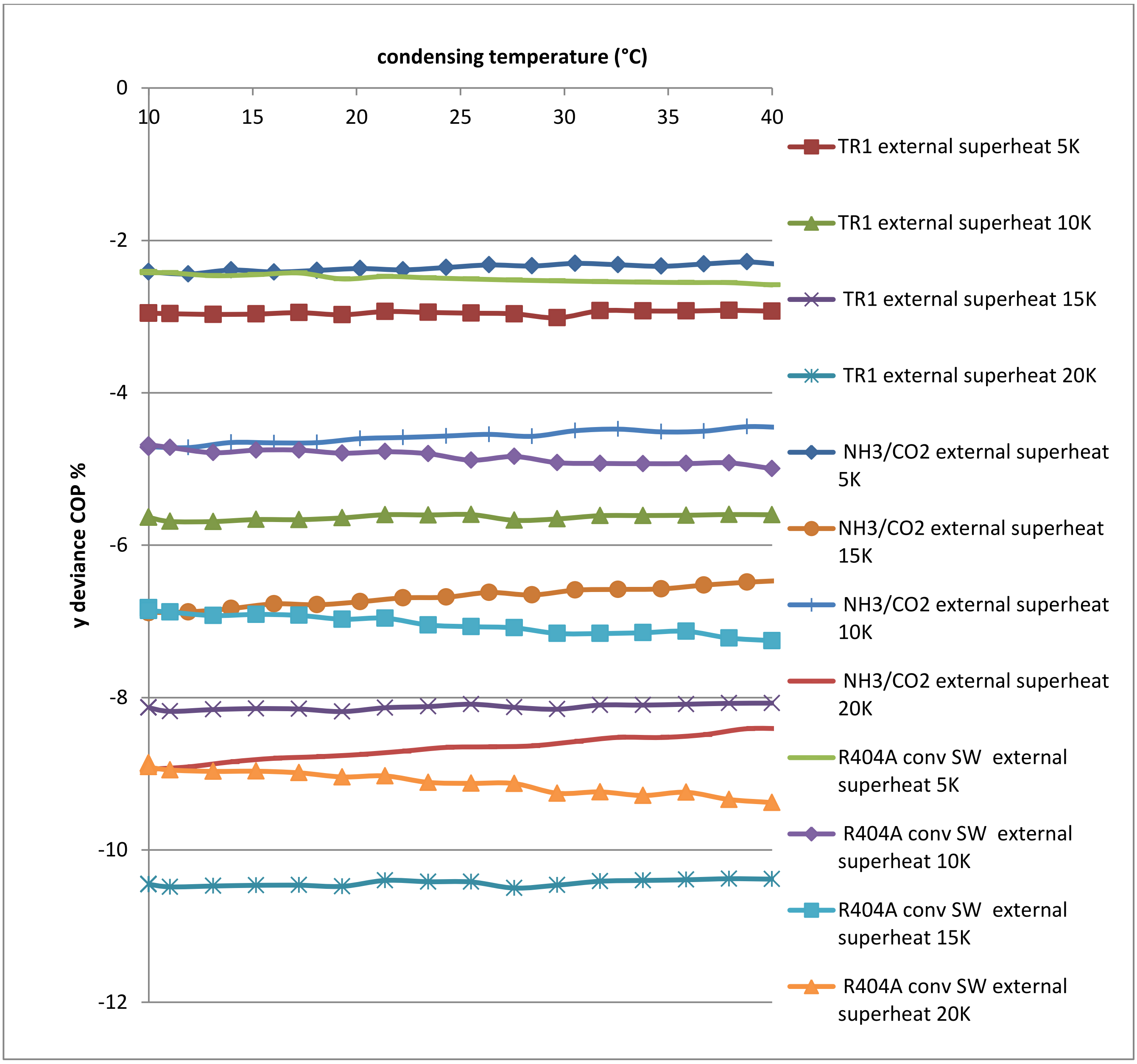

Figure 8.1: low temperature COP deviance as a function of the condensing temperature for R404A Swedish system, NH_3/CO_2 cascade system and TR1