Resurseffektiva kyl- och värmepumpssystem

Supermarket refrigeration and heat recovery using CO_2 as refrigerant

A comprehensive evaluation based on field measurements and modelling

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Summary

This project investigates the potentials, challenges and opportunities of using CO_2 as refrigerant in the supermarket refrigeration and heat recovery systems. The focus is on CO_2 trans-critical booster system, as the emerging state-of-the-art system in supermarket refrigeration field.

The CO2 booster system performance is studied using computer modeling and field measurement analysis to find the most energy efficient ways for providing simultaneous cooling and heating demands in supermarkets. Through this research work, the solutions available on the market are investigated, suggestions on system modifications and optimization are made, and new system solutions are suggested.

Sammanfattning

Detta projekt undersöker utmaningar och möjligheter att använda CO2 som köldmedium i livsmedelsbutiker. Fokus ligger på CO2 transkritiska system som är den bästa tillgängliga tekniken för kylsystemet i livsmedelsbutiker.

Med hjälp av datamodeller och fältmätningar har prestanda av CO2 systemet studerats för att hitta den mest energieffektiva lösningen som levererar både kyl-och värmebehov i livsmedelsbutiker. I forskningsprojektet har olika systemlösningar som finns på marknaden med CO2 som köldmedium undersökts där förslag på systemändringar och optimering presenteras.

1. Introduction

1.1. Background

Supermarkets are energy intensive buildings consuming 3-4% of the total annual electricity in industrialized countries (Sjöberg, 1997)(Reinholdt and Madsen, 2010)(Orphelin and Marchio, 1997)(Tassou et al., 2011) and 35-50% of this total electricity is consumed in the supermarket refrigeration systems (Arias, 2005). Supermarket refrigeration systems are one of the largest consumers and emitters of high GWP refrigerants; 30% of Europe HFC consumption (SKM Enviros, 2012) with 3-22% annual leakage rate (IPCC/TEAP, 2005).

Due to its significant negative environmental impacts, supermarket refrigeration is among the sectors that will be most impacted by the new European Parliament F-gas regulation; to ban use of any refrigerant with GWP higher than 150 for centralized systems larger than 40 kW from year 2022, with exception for primary cycle in cascade configurations to use refrigerants with GWP up to 1500 (The European Parliament, 2014). As a consequence, HFCs cannot be seen as long term solutions for supermarket refrigeration systems.

To avoid the damages to the environment, it was proposed to use the natural refrigerants. They exist in nature and this implies that their usage will not lead to any unforeseen risks. In the supermarket refrigeration field, among the natural refrigerants, CO2 gains market acceptance due to its good safety characteristics. The state-of-the-art system in supermarket refrigeration in Sweden is CO2 trans-critical booster systems. This system is an only-CO2 solution which provides cooling in the medium temperature (MT) cabinets and low temperature (LT) freezers. The system is considered as one of latest developments towards using climate friendly refrigerants in Swedish supermarkets and it has become the main stream of new installations (Kyla, 2012). System’s independency of using other refrigerants such as HFCs, ammonia or hydrocarbons in indirect or cascade configurations results in less negative environmental impacts (compared to HFC) and better safety (compared to NH3-HC).

The trans-critical booster system has been reported to be able to cover the entire heating demands of an average size supermarket in Sweden. The integration of refrigeration and heating in this system leads to less annual energy use when compared to a conventional R404A refrigeration system with separate heat pump for heating needs (Sawalha, 2013). The CO2 system can be controlled to match well the simultaneous cooling and heating needs in the supermarket; however, it has to be controlled properly in order to run with the highest efficiency possible.

Research at KTH on supermarket refrigeration and using CO2 as the refrigerant at KTH dates back to early 2000s. Samer Sawalha (Sawalha, 2008) and Jaime Arias (Arias,

2005) carried out their doctoral studies on this subject. During these years there is a good collaboration between KTH and the industrial partners. This Effsys+ project is a continuation of this collaboration.

1.2. **Objectives**

- Literature survey on CO_2 refrigeration research and development in national and international level
- Case studies of different system configurations in different countries
- Produce guidelines on how to instrument and analyse the supermarket refrigeration systems
- Studying control strategies for refrigeration and heat recovery
- Heat recovery investigation in various system solutions
- Develop/propose standardized methods for field measurements and evaluation of supermarket refrigeration - heat recovery
- Field measurements investigation of system and key components performance and tracing the main losses in different system solutions
- Computer modelling of different system solutions, with focus on CO_2 trans-critical booster system
- Present the definition of a state-of-the-art system, its characteristics and examples
- Establish the importance of appropriate (adapted) heat exchangers for CO_2-systems
- Identify good and relevant indicators for system and component comparison
- Refine an existing simulation model to allow comparison of supermarket energy systems.
- Extend the number of investigated supermarkets
- Produce a basis for developing guidelines in heat recovery system design

1.3. **Methodology**

The work in this project is divided in two main work packages:

1.3.1. Computer simulation modelling

Computer modelling is used to analyse the performance of different supermarket refrigeration systems including CO_2 trans-critical booster system and conventional R404a system. The inputs and boundaries of the models are to a great extent based on field measurement experiences. Different refrigeration systems performance, heat recovery investigation, study on booster system improvement, internal heat exchangers configurations and an alternative method for system analysis are some parts of the research that have been fulfilled by computer modelling.

1.3.2. Field measurements

Access to real-time field measurements of about 15 supermarkets gives an opportunity for studying real system performances. Data process is a crucial part of these field measurements evaluations. Real-time field measurements and web-monitoring are used for data acquisition, synchronization and analysis. Analysis of refrigeration system-component performance, evaluation of heat recovery in two supermarkets and control strategies study in four supermarkets are some parts of the research that have been fulfilled by filed measurements.

1.4. Project partners and acknowledgement

The project is co-financed by Swedish Energy Agency (Sveriges Energimyndigheten) and Swedish Refrigeration Organization (Kylbranschens Samarbetsstiftelse) and several industrial partners. The current research is within the Effsys+ program, as a continuation for previous research programs "Alternativa Köldmedier", "Klimat 21", "eff-sys" and "Effsys2".

The participating institutes and companies in the project are:

- Royal Institute of Technology (KTH): Samer Sawalha / Mazyar Karampour / Jaime Arias
- Energi & Kylanalys: Jörgen Rogstam
- Green & Cool: Micael Antonsson
- ICA Sverige: Per-Erik Jansson
- Huurre: Kenneth Lindberg
- Alfa Laval: Göran Hammarson
- IWMAC: Conny Andersson
- Carrier Ref: Björn Staf
- Wica Cold: Niclas Rindhagen
- Cupori: David Sharp
- Ahlsell Kyl: Roger Wranér (the company left the project 2013)

2. Supermarket refrigeration system

In general, two temperature levels are required in supermarkets for chilled and frozen products; temperatures of around +3°C and -18°C are commonly maintained respectively. In such application, with a large difference between evaporating and condensing temperatures, the cascade or other two-stage solutions become favorable and are adaptable for the two temperature level requirements of the supermarket.

The use of CO_2 in supermarket refrigeration started with applying CO_2 as the working fluid in indirect systems for freezing applications. Then it has been applied in cascade solutions mainly with R404A in the high temperature stage. After gaining experience and the availability of components CO_2 has been used in trans-critical system solutions, mainly in northern Europe. The two main trans-critical system solutions applied are the parallel and the booster, the later is the latest in CO_2 systems development series and has been applied in most of the installations in Sweden.

CO2 trans-critical booster system is the main system that has been investigated in this research work. A simple schematic of this system is shown in Figure 1. Two temperature levels are used to maintain the quality of fresh and frozen products. The absorbed heat rejected in a gas cooler/condenser. In case of demand for space /tap water heating, part of or the entire rejected heat can be recovered in a de-superheater.

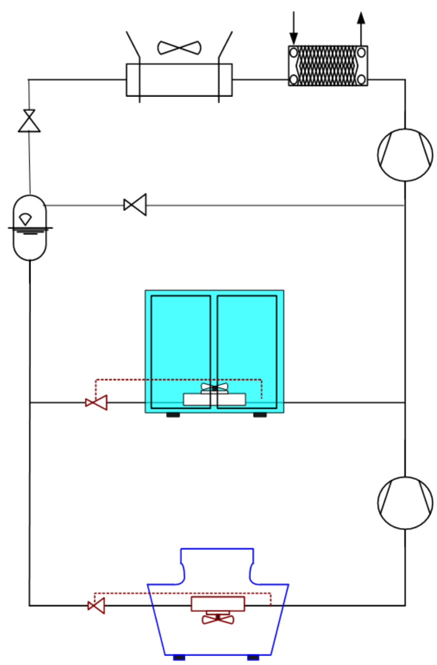

Figure 1: a CO2 trans-critical booster system

2.1. **Market status**

There are about 3000 CO2 trans-critical booster systems installed in the world where more than 90% of these systems operates in Europe (Shecco, 2014). Majority of the systems are installed in the last few years. Denmark, UK, Germany, Switzerland, Norway and Sweden have the highest numbers of these systems in Europe. The driving force for this development of these systems varies from heavy taxes on HFC trade to incentives to use natural refrigerants. Japan and Canada have the highest numbers of CO2 trans-critical booster systems outside Europe.

CO2 cascade and indirect systems are other types of supermarket refrigeration systems that partly use CO2 as the refrigerant, mainly in low temperature level.

Maps of CO2-system installations, including trans-critical, cascade and indirect, are shown in Figure 2.

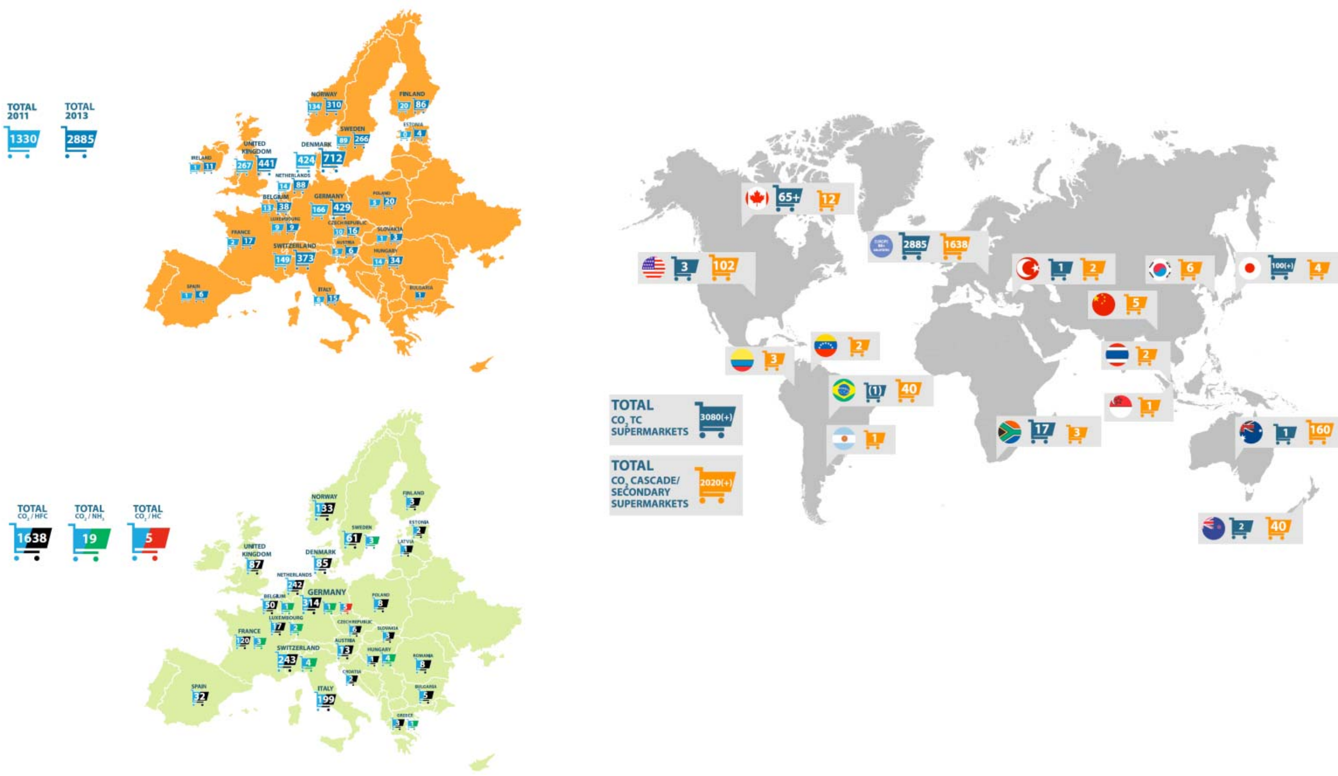

Figure 2: Year 2013 map of CO_2 trans-critical booster systems in Europe (top-left), CO_2 cascade systems in Europe (bottom-left) and CO2 trans-critical and cascade/secondary stores in world (right) (Shecco, 2014)

3. Research summaries

A summary of main research results in this project is presented in this chapter. Section 3.1 summaries the evaluation on the performance comparison of eight supermarkets, five CO_2 trans-critical booster systems versus three conventional R404a systems. Section 3.2 presents an alternative simplified method for performance evaluation of supermarket systems. Section 3.3 investigates the heat recovery control strategy in CO_2 trans-critical booster systems. Filed measurements of two supermarkets using CO_2 as refrigerant is studied and their refrigeration/heat recovery performance in practice is compared with optimum theoretical method in section 3.4. An annual performance analysis of a system following the discussed heat recovery control strategy is presented in section 3.5. An analysis on system performance improvement, with focus on internal heat exchangers arrangement, is reviewed in section 3.6.

3.1. Summary of past project: Field measurements of eight Swedish supermarkets (Karampour et al., 2013)

In this section the findings of early research projects, funded by the Swedish Energy Agency, related to the subject of supermarket refrigeration are reviewed. This included summarizing extensive investigation field measurement investigation of several CO_2 and HFC supermarket refrigeration systems.

As an example, performance of eight supermarkets (five CO_2 trans-critical and three R404/R410) have been analysed and compared. The system configurations are shown in Figure 3.

It can be observed in Figure 4 that the older CO_2 systems, TR1, TR2 and TR3, have lower COP than the reference systems. Some of the main reasons for the low COP are: high vapour fraction in LT cabinets inlet, relatively high amount of internal and external superheating and 10-15% lower overall efficiency of LT compressors than the other systems.

But, the newer CO_2 systems, TR4 and TR5, proved to have higher efficiency, or at least, as energy efficient as the conventional HFC systems, shown in Figure 4. This originates from the modifications in the system design, including receiver and flash gas by-pass, and more suitable CO_2 components. Some of the main factors are: the new CO_2 systems have relatively high efficiency for MT level compressors; 5-10% higher overall efficiency than other HFC and CO_2 systems. The systems also have cabinets with lower LMTD (lower superheating and higher evaporation temperature) in comparison with the older CO_2 system cabinets, which were not especially designed to handle CO_2. This can be seen in Figure 5. The conclusion is that CO_2 systems have comparable COP to advanced conventional systems applied in Sweden.

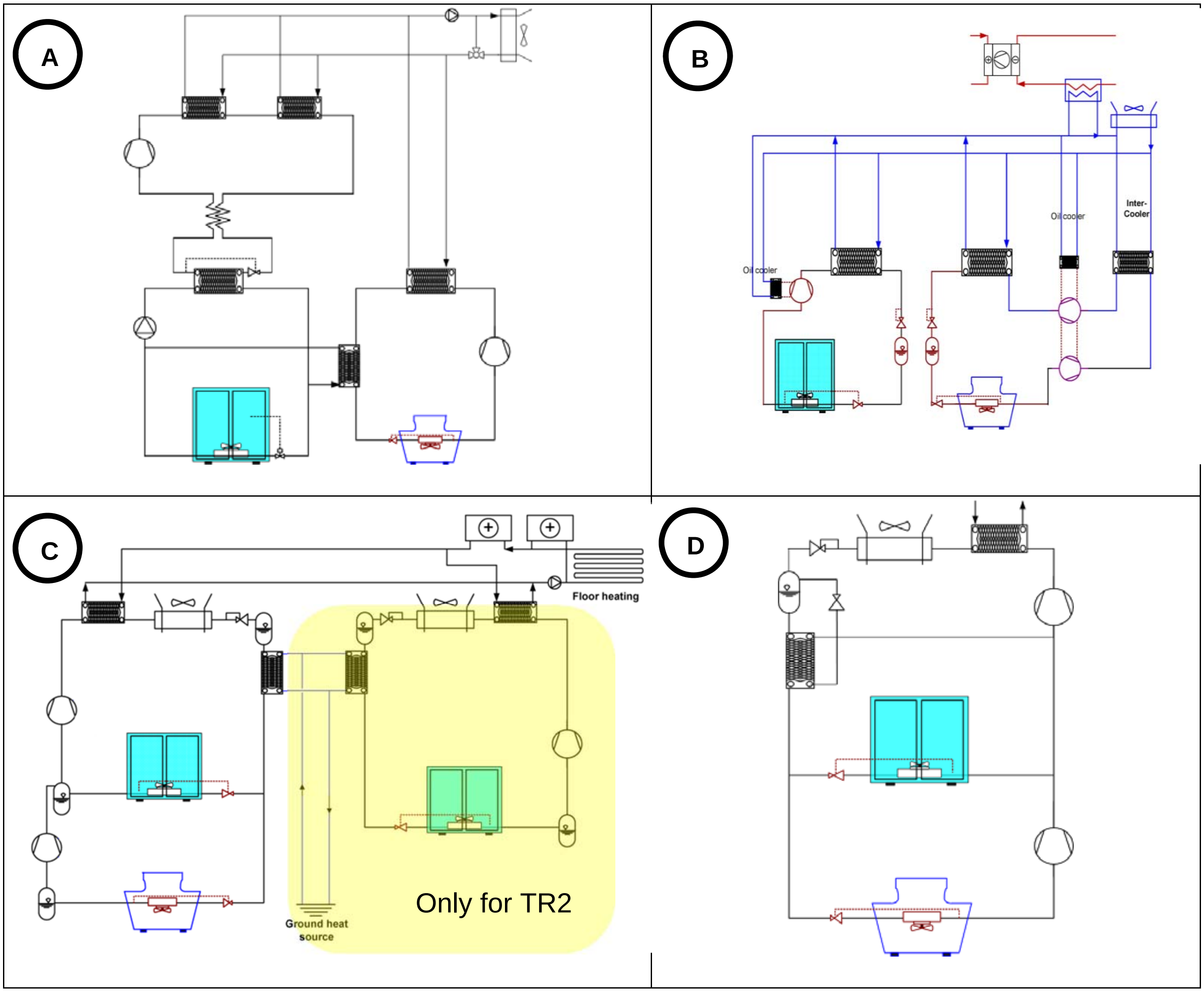

Figure 3: Systems schematics A) HFC reference systems RS1-RS2-RS3. B) TR1-Parallel transcritical. C) TR2-transcritical booster + Parallel medium temperature cycle. TR3- has only transcritical booster units D) TR4 and TR5-transcritical booster with flash gas by-pass.

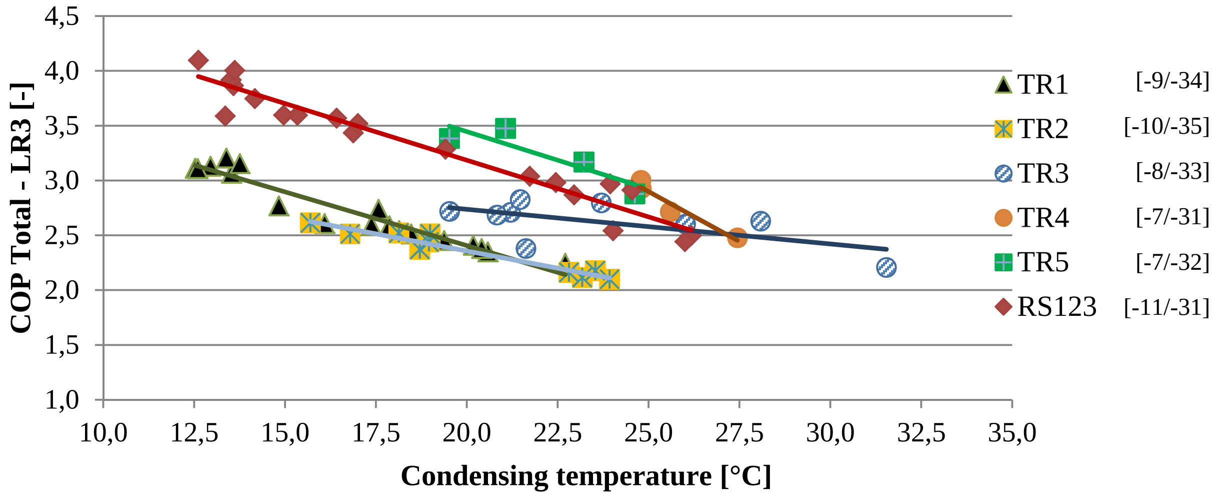

Figure 4: COP comparison of five CO2 trans-critical booster systems (TR1, TR2, TR3, TR4 and TR5) and three R404a conventional systems (RS123)

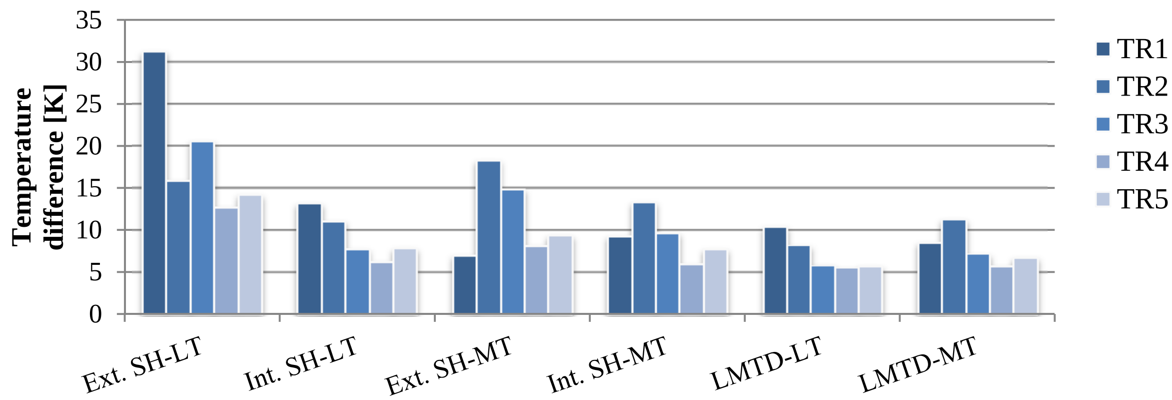

Figure 5: Comparison of internal superheating, external superheating and LMTD of five CO2 systems

3.2. An alternative method to evaluate refrigeration system performance

An alternative method is developed to find COP and cooling power of CO2 refrigeration systems without the need to measure refrigerant mass flow rate directly and calculate enthalpies. The method is referred to as “Y-method”, it is based on finding graphically the impacts of superheating and sub-cooling (Y-multipliers) on a basic cycle COP. The motivation to develop this method is to make system performance indicators easily-accurately available, both on-site in the supermarket and through web monitoring softwares. The calculations are based on simple algebraic equations where the input parameters required are basic system measurements. The results show good accuracy with 5-10% maximum difference between conventional method and Y-method.

The results of calculating the cooling capacities and COPs for a supermarket using CO_2, based on the conventional method and the alternative method, are shown in Figure 6 and Figure 7.

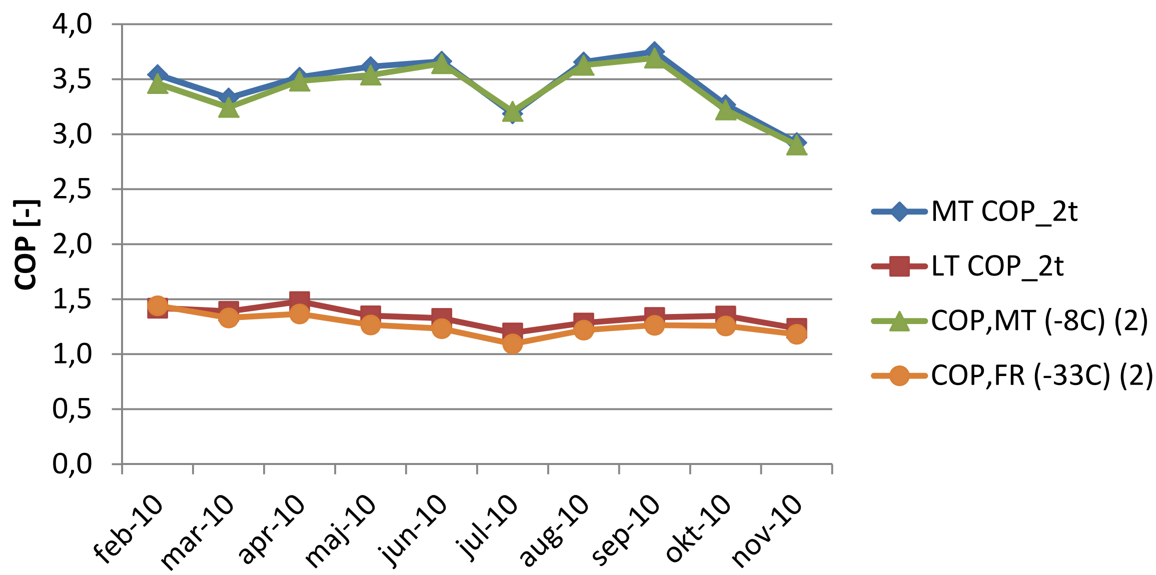

Figure 6: Comparison of cooling COPs between the conventional and alternative methods

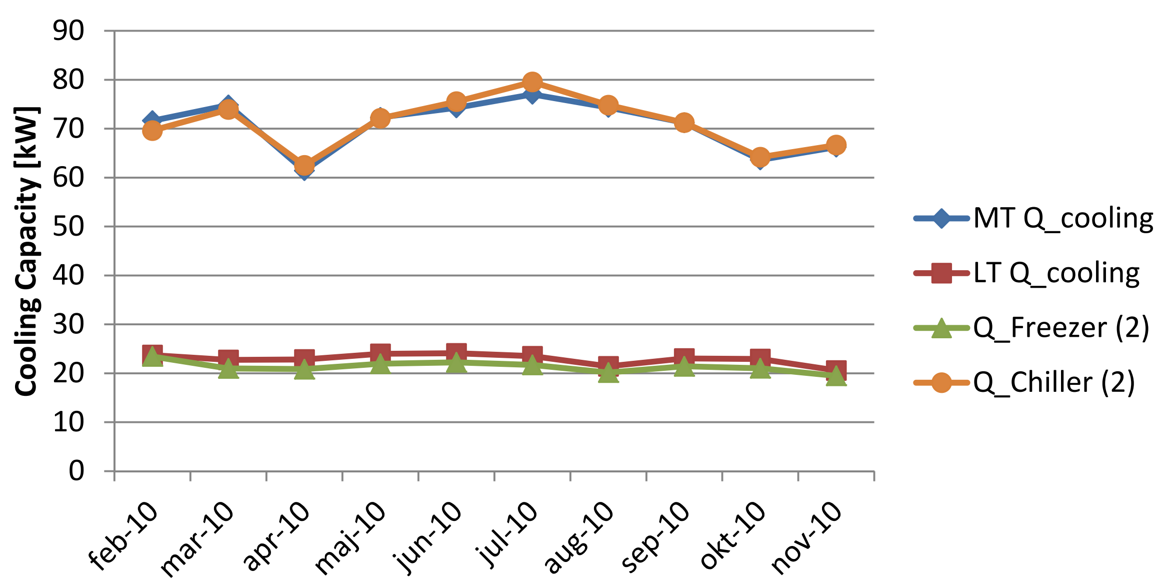

Figure 7: Comparison of cooling capacities between the conventional and alternative method

3.3. Heat recovery in CO2 trans-critical booster system (Sawalha, 2013)

Using computer simulation modelling this study investigates the performance of a CO2 trans-critical system with heat recovery from the de-superheater. The influence of sub-cooling (or further cooling) in the condenser/gas cooler on system performance is investigated.

The research work presented in this section has been continuation of the work performed in a project within Effsys2 program, project number is EFFSYS2-P21. In this project, the heat recovery analysis have been developed further and the results have been publishing in an international journal (Sawalha, 2013), field measurements were carried out in two supermarkets to analyse the heat recovery concept.

3.3.1. System description

The system analysed in this study is a CO2 booster system solution with heat recovery from the de-superheater. The discharge pressure of the system is controlled according to the required heat in the supermarket, when there is no heating need in the supermarket the discharge pressure is kept as low as possible following the ambient temperature. When the system operates in the trans-critical mode the discharge pressure is calculated according to the following equation (Sawalha, 2008;Liao et al., 2000):

$$P_{opt,disch} = 2.7 \cdot t_{GC,exit} - 6\ [bars] \qquad \text{Eq(1)}$$

The cooling demand of the refrigeration system at the medium temperature level is dependent on the ambient conditions and assumed to change linearly between full capacity of 200 kW at 35°C and 50% of the full capacity at 10°C ambient, below which the demand remains constant. The cooling demand at the low temperature level is assummed to be constant at 35 kW independently of the ambient temperature. The heating demand for a medium size Swedish supermarket is obtained by the program CyberMart. Detailed descriptions and calculations of the program can be found in the Doctoral Thesis of Arias (2005). The main heating demand in supermarkets is space heating and the set point to start supplying the heat is 10°C ambient temperature. It is estimated that at 10°C the heating demand is 40 kW for a medium size Swedish supermarket and it increases linearly to 190 kW at -20°C ambient temperature.

3.3.2. System analysis

In order to evaluate the system performance in heat recovery mode the heating COP ($COP1_{HR}$) of the refrigeration system is used, it is expressed in the following equation:

$$COP1_{HR} = \frac{\dot{Q}_1}{\dot{E}_{HR} - \dot{E}_{FC}} \ [-] \qquad \text{Eq(2)}$$

It is defined as the ratio between the heating demand to the power consumed to provide the heating demand, which is the difference between the power consumption of the refrigeration system in heat recovery mode and floating condensing mode. The refrigeration system in this study is assumed to recover all the heating demand in the supermarket; therefore, $\dot{Q}_1 = \dot{Q}_{HR}$. $COP1_{HR}$ of the booster system with 30°C and 20°C return temperature from the heating system are plotted in Figure 8.

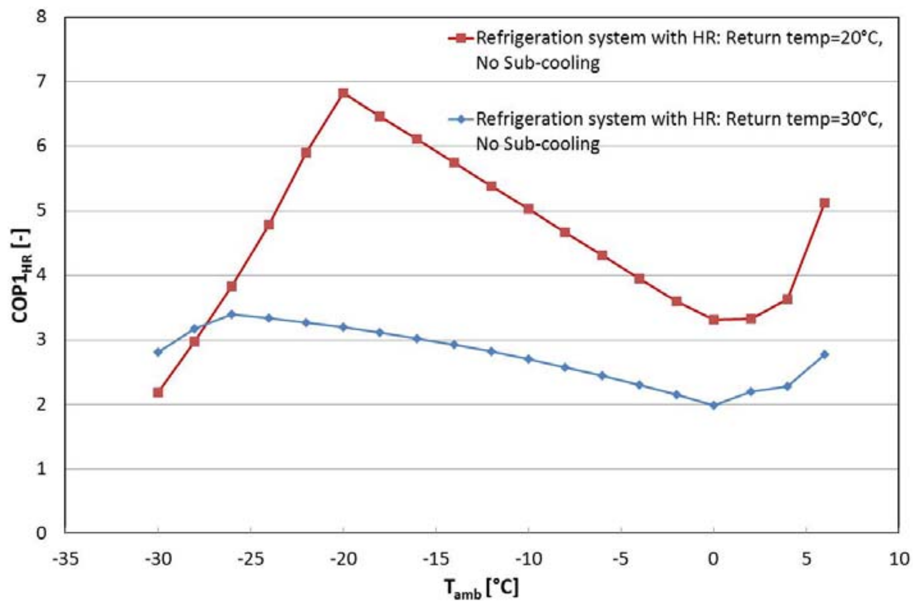

Figure 8: $COP1_{HR}$ of CO2 booster system (heating system return temperature of 20 and 30°C).

The refrigeration system in these calculations is not controlled for sub/further-cooling in the condenser/gas cooler when the system is running in the heat recovery mode. When the ambient temperature is low the heating demand is high and the discharge pressure is raised to recover heat, in this case the gas cooler can be operated to further cool the refrigerant before passing the expansion valve. This further cooling is referred to as sub-cooling in this study.

3.3.3. Influence of sub-cooling on the system's COP

Running the gas cooler to cool the refrigerant down in the heat recovery mode has two main effects on the system performance, it increased the system's COP2 which is a positive influence but it also reduces the available heating energy for recovery from the de-superheater at certain discharge pressure; this is due to the smaller refrigerant mass flow rate running in the system with sub-cooling. In order to analyse the influence of the condenser/gas cooler operation on the system performance the system is analysed for the following two cases:

3.3.3.1. Case 1

The condenser/gas cooler is operated to provide no sub-cooling in the sub-critical region. In the trans-critical operation the system rejects all the heat in the de-superheater. The heating demand is presented on the x-axis of Figure 9 as the heat recovery ratio (HRR), which is defined as the ratio of the heating demand ($\dot{Q}_1$) (totally covered by the de-superheater; $\dot{Q}_1 = \dot{Q}_{1_{HR}}$) to the total cooling demand at the medium temperature level; including the heat rejected from the low temperature cycle. HRR is expressed in the following equation:

$$HRR = \frac{\dot{Q}_1}{\dot{Q}_{m,tot}} \ [\%] \qquad \text{Eq(3)}$$

where

$$\dot{Q}_{m,tot} = \dot{Q}_{m,cab} + \dot{Q}_{f,cab} + \dot{E}_{f,shaft} \ [kW] \qquad \text{Eq(4)}$$

Three regions in Figure 9 can be observed; “Region 1” is where the system is in sub-critical operation, the decrease in medium temperature COP (COP2,m) is sharp with increasing the demand for heat recovery. “Region 2” is where the system is about to switch to trans-critical operation; this region can be identified to start at few bars below the critical point, about 69 bar. In this region much larger amount of heating energy can be recovered with a slight increase in discharge pressure. “Region 3” is where relatively large increase in discharge pressure results in slight increase in the amount of heating energy to be recovered, consequently sharper drop of COP2,m is observed. “Region 3” is where the isotherm of the CO_2 temperature at the exit of the de-superheater starts to become steep, the 35°C isotherm.

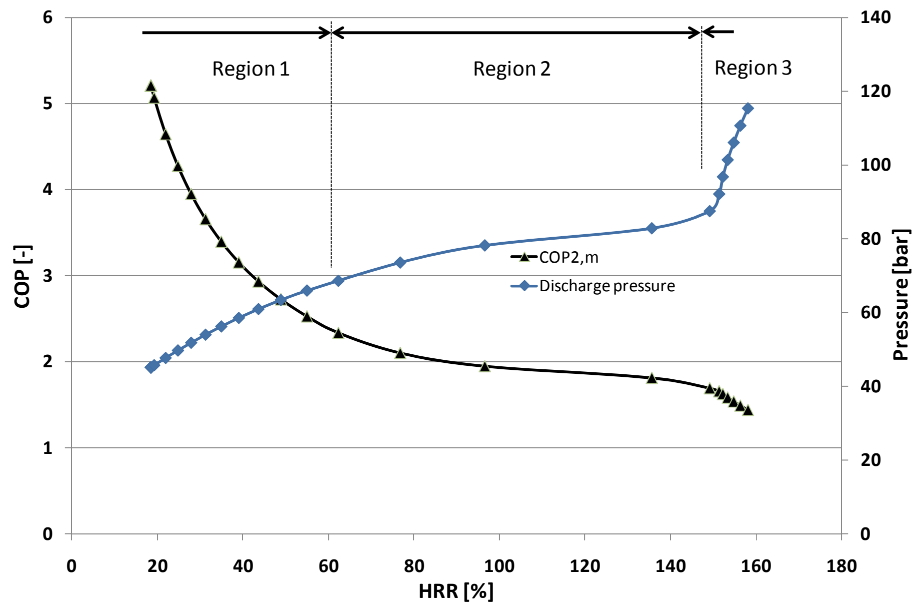

Figure 9: COP2,m and the corresponding discharge pressure of the booster system as a function of HRR. The case is for the system without sub-cooling in the condenser/gas cooler.

3.3.3.2. Case 2

The condenser/gas cooler is operating to provide sub-cooling in the sub-critical region and further cooling after the de-superheater in the trans-critical operation. The degree of sub-cooling that can be achieved in the system depends on the ambient temperature and the capacity at which the condenser/gas cooler operates.

Calculations are made for different condenser/gas cooler exit temperatures, COP2,m and the corresponding discharge pressures are presented in Figure 10 and Figure 11 respectively. The positive influence of sub-cooling can be observed in Figure 10; for example, when the system operates to provide heating that corresponds to HRR of 60%, the COP2,m increases from about 2.4 in the case of without sub-cooling to about 3.0 in the case of 5°C exit temperature of the condenser/gas cooler, this is despite the need to operate the system at higher discharge pressure; 68 compared to about 78 bars respectively, see Figure 11.

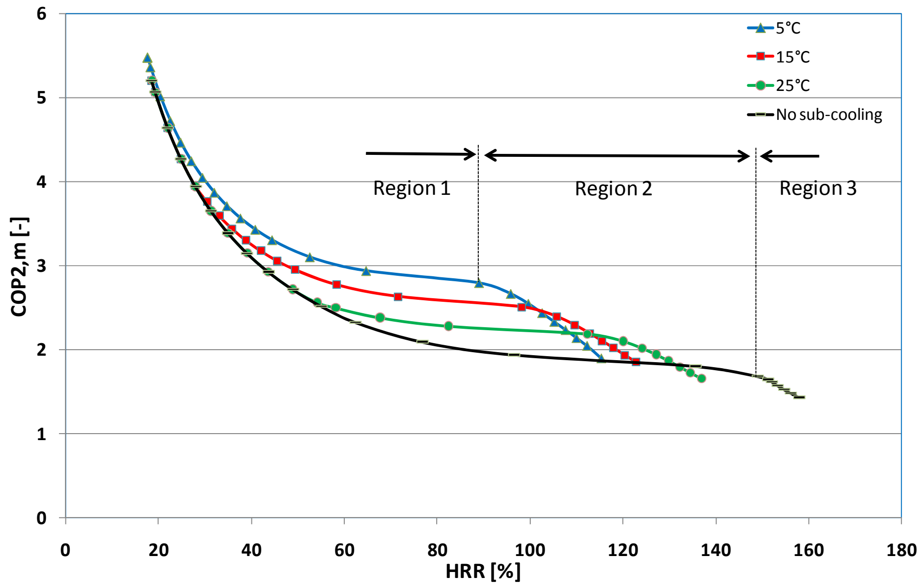

Figure 10: COP2,m of the booster system as a function of HRR. The cases are for different condenser/gas cooler exit temperatures.

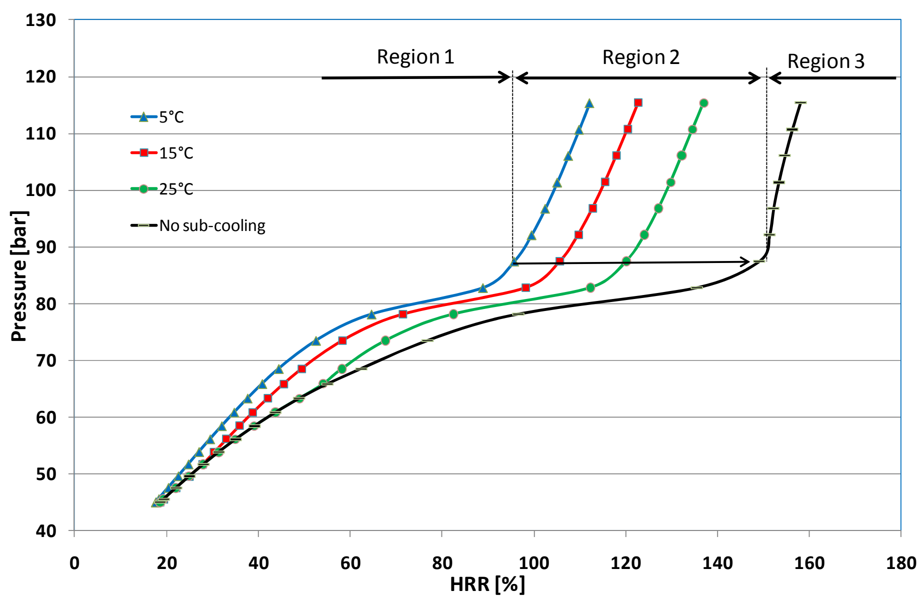

Figure 11: Discharge pressure as a function of HRR for different condenser/gas cooler exit temperatures.

It can be observed in "Region 1" in the plots of Figure 10 that the system will have the highest COP when controlled to achieve the lowest gas cooler exit temperature.

Therefore, the condenser/gas cooler should operate at full capacity and the discharge pressure should be regulated to match the required heating demand from the refrigeration system.

This is correct up to a point, start of “Region 2” in Figure 10, where more heating energy will be needed and if the condenser/gas cooler is still running at full capacity the drop in COP will be steep, as can be seen in all plots in Figure 10, this is where the isotherm line starts to become steep. The discharge pressure at which the COP starts to drop in a steeper trend is about 88 bars, as can be observed in the start of “Region 2” in Figure 11.

In order to operate the system at the highest COP possible in “Region 2” in Figure 10, the system must be operated at the maximum discharge pressure to achieve the highest COP, about 88 bars in this case, and the condenser/gas cooler capacity should be reduced so more heating energy will be available in the de-superheater to be supplied to the heating system. The operation will follow the arrow crossing “Region 2” in Figure 11. The maximum operating pressure for highest COP in heat recovery mode is dependent on refrigerant exit temperature from the de-superheated; it follows the same correlation for the optimum discharge pressure for maximum COP in a CO2 refrigeration system, expressed in equation (1).

With increasing the heating demand the condenser/gas cooler should eventually be switched off so all the system’s heating energy can be rejected in the de-superheater, this is where “Region 3” in Figure 10 and Figure 11 starts. Beyond this point the only way to recover more heating energy from the system is by increasing the discharge pressure. It can be observed in Figure 10 and Figure 11 that higher than a discharge pressure of about 88 bars the system will have a relatively steeper drop in the COP and sharp increase in discharge pressure. If we assume that due to the sharp increase in discharge pressure, the start of “Region 3” is the limit of the refrigeration system to provide heating at reasonable efficiency then the refrigeration system can provide HRR of about 150%, i.e. 225 kW of heat in this case study.

Calculating for the system to run with optimum control to achieve the highest COP possible, the approach temperature difference in the condenser/gas cooler is assumed to be 5K when the condenser/gas cooler is running at full capacity in “Region 1” in Figure 10. The resulting medium and low temperature COP’s are plotted in Figure 12. The operating discharge pressure and condenser/gas cooler exit temperature are plotted in Figure 13.

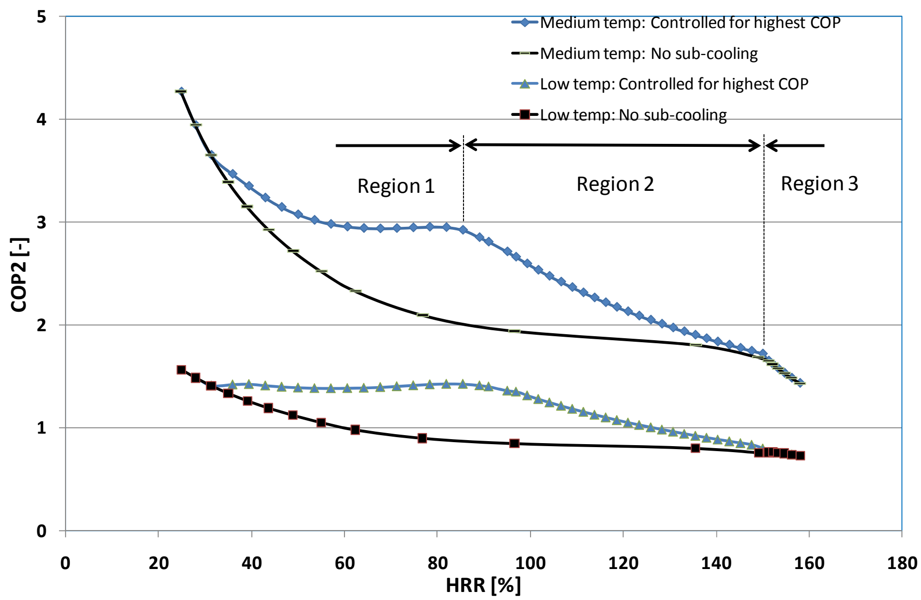

Figure 12: COP2,m and COP2,f as a function of HRR. The cases are for operation with no sub-cooling in the condenser/gas cooler and for system controlled for highest COP possible.

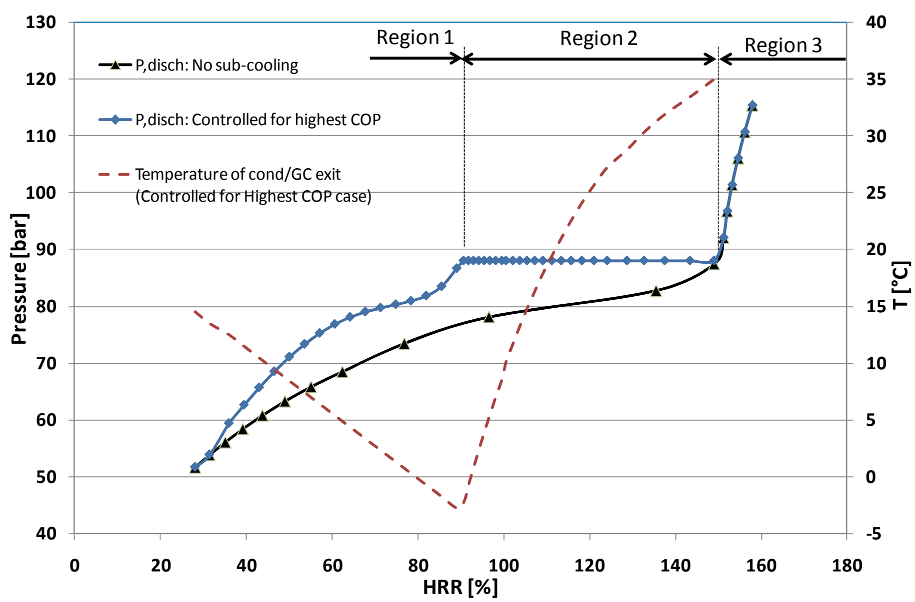

Figure 13: Discharge pressure and condenser/gas cooler exit temperature as a function of HRR. The cases are for operation with no sub-cooling in the condenser/gas cooler and for system controlled for highest COP.

As can be seen in Figure 12, the COP is higher for the system using the condenser/gas cooler for sub-cooling. The three regions identified in earlier plots can also be identified in this plot. At the curves' edges with low HRR there is almost no difference in the COP between both cases, this is due to the relatively high ambient temperature so negligible or little sub-cooling can be achieved in the system.

In Figure 13, it can be observed that in “Region 1” the condenser/gas cooler exit temperature decreases following ambient temperature (increasing HRR) because the condenser/gas cooler is running at full capacity. The discharge pressure increases compared to the “no sub-cooling” case so the same amount of heating energy can be recovered. “Region 2” starts when the system reaches the maximum discharge pressure for highest COP; about 88 bars in this case. The pressure is fixed, as can be seen in “Region 2”, and the condenser/gas cooler fans are controlled for reduced capacity to recover more heat in the de-superheater, this results in higher condenser/gas cooler exit temperature.

The resulting total power consumption of the refrigeration system when controlled for the highest COP is plotted in Figure 14. The power consumption for the cases of heat recovery without sub-cooling and for floating condensing are also plotted. The corresponding $COP1_{HR}$ are plotted in Figure 15.

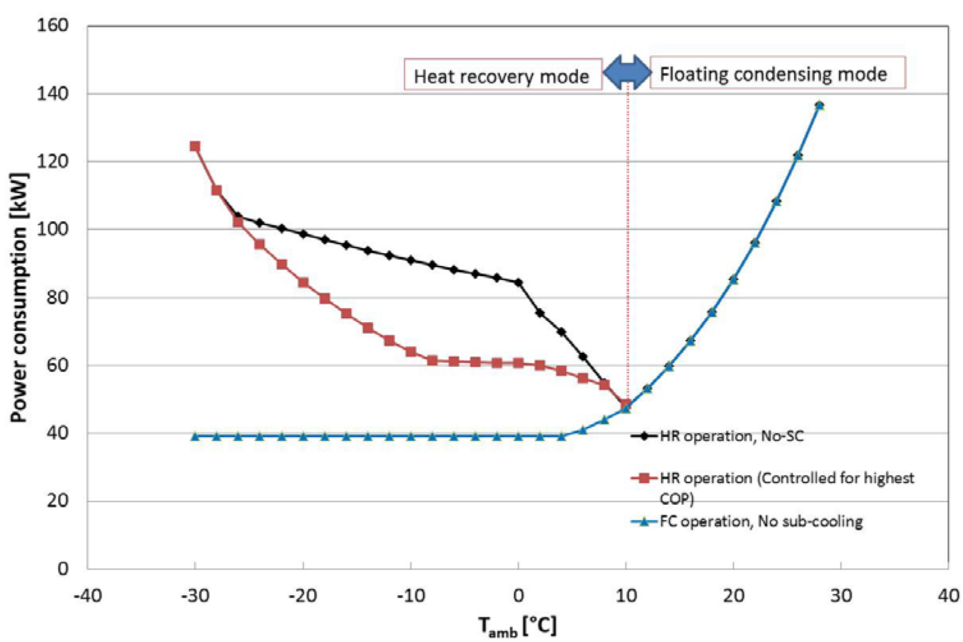

Figure 14: Power consumption of the refrigeration system at different ambient temperatures. The cases are for heat recovery mode with no sub-cooling, system controlled for highest COP possible and for floating condensing without heat recovery.

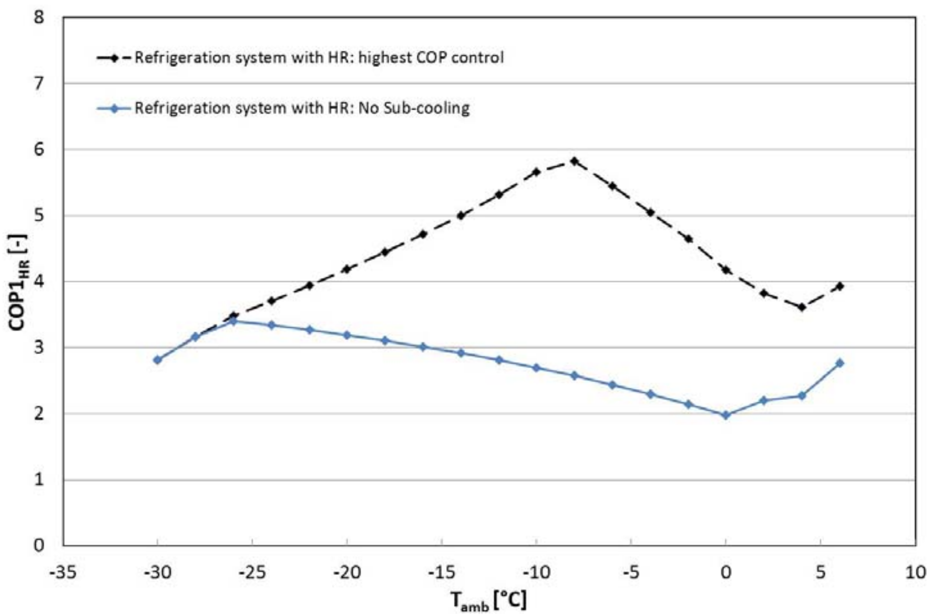

Figure 15: $COP1_{HR}$ for the refrigeration system with heat recovery from the de-superheater. For the cases of heating system return temperatures of 30°C with no sub-cooling and for controlled for highest COP possible.

It can be observed in the plot that the booster system's $COP1_{HR}$ increases significantly, for most of the ambient temperature range, due to the use of sub-cooling.

3.3.4. Annual energy use

Annual energy use of the discussed CO2 booster system providing heating and cooling is compared with a conventional R404a system without control for heat recovery and with a separate heat pump. It is found that CO2 system consumes 6% less annual energy, 540 MWh versus 565 MWh for conventional system with separate heat pump.

To summarize, following the suggested control strategy in this study, the extra operating energy demand required to recover the needed heating energy from the analysed CO2 system is smaller than what a typical heat pump would require for the same load. This is the case for almost all ambient temperatures over a full season. When taking the simultaneous heating and cooling loads into account, the CO2 trans-critical system has lower annual energy usage in an average size supermarket in Sweden when compared to a conventional R404A refrigeration system with separate heat pump for heating needs. CO2 trans-critical systems are efficient solutions for simultaneous cooling and heating needs in supermarkets in relatively cold climates.

Suggested control strategy: Sub-cooling in the condenser/gas cooler in the heat recovery mode increases the system's COP; therefore, **1-** The condenser/gas cooler should be operated at full capacity in the heat recovery mode as long as the pressure is lower than the maximum value to achieve the highest COP. **2-** When the heating needs reach a high value where the maximum discharge pressure for the highest COP is reached then the pressure should not be increased and the condenser/gas cooler fans speed should be reduced to increase the recovered heat from the system. **3-** The maximum heating capacity of the refrigeration system is reached when the discharge pressure is at the maximum value for highest COP and the condenser/gas cooler is switched off, or by-passed. For the case analysed in this study the system can provide heating energy about 1.5 times the total demand at the medium temperature level (i.e. $\dot{\mathbf{Q}}_{\mathbf{m,tot}}$).

3.4. Heat recovery – Field measurements and modelling (Abdi et al., 2014)

This study investigates the heat reclaim of trans-critical CO_2 booster refrigeration unit in two supermarkets in Sweden. The aim is to compare the control strategy for heat recovery in real supermarket installation to the optimum control strategy. The results of this research has been presented in a master thesis report (Abdi, 2014) and a conference paper (Abdi et al., 2014). The main findings for one of the supermarkets are discussed here in brief.

Heat is recovered by the refrigeration system to some extent and district heating is used to complement covering the heat load. Discharge pressure is varied between 50-80bars and system is run in trans-critical area sometimes. The main remaining question is that how much potential the refrigeration system has to cover higher heat loads. The simulation model is used to investigate the higher rate of heat recovery.

The discharge pressure and gas cooler exit temperature that the system will have to be operated at for different HRR are calculated. In computer simulation fixed value of the desuperheater exit temperature was used; 35℃. However it varies considerably depending on the heating demand in the supermarket. Figure 16 shows the simulated curves of discharge pressure and gas cooler exit temperature for a wide range of HRR. The heat recovery starts from 58bars at the lowest HRR and reaches 88.5 bars at higher HRR.

Measured discharge pressure values, averaged hourly are plotted in Figure 16 match the simulated curve quite well. Measured discharge pressure values show that the highest HRR is around 80 % of cooling demand of the system. The results show that the system has potential to recover heat to much higher extent by following the explained manner. Measured gas cooler exit temperature values, averaged hourly and plotted in Figure 16 deviates from the simulated curve indicating maximum sub-cooling is not gained before discharge pressure reaches the maximum value of discharge pressure (88.5bars); however at low HRR the influence of sub-cooling is not expected to be significant on the efficiency of the system.

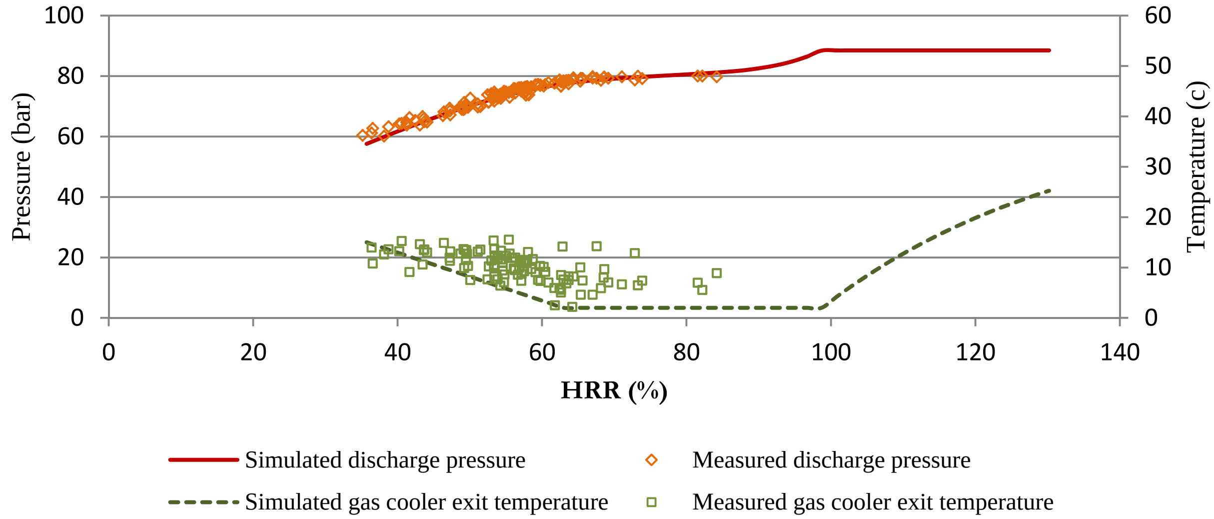

Figure 16: Calculated boundaries from simulation and hourly averaged values from field measurement

The expected $COP_{1\text{-}HR}$ from the system is plotted versus HRR in Figure 17. The difference in energy use compared to the measured values in heat recovery mode (E_{HR}) was used according to equation (2) to calculate the systems's measured $COP_{1\text{-}HR}$. The simulated curve shows that for low HRR (less than 40%) heat can be recovered at quite high $COP_{1\text{-}HR}$. For HRR higher than 40 %, $COP_{1\text{-}HR}$ varies from 3.5 to 5.2. Measured hourly averaged points of $COP_{1\text{-}HR}$ plotted in Figure 17 varies around the simulated values, the difference may be result of variation in return temperature of water from heating system and also changes in gas cooler exit temperature.

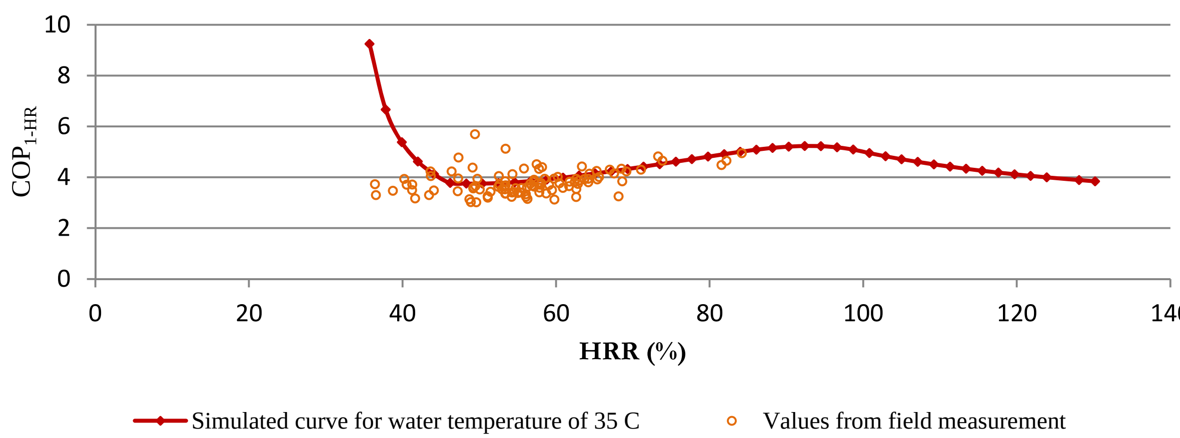

Figure 17: Calculated $COP_{1\text{-}HR}$ from simulation and hourly averaged values from field measurement

To summarize the main findings of this research, the results show that heat can be recovered at $COP_{1\text{-}HR}$ of 3-4.5. The theoretical analysis shows that the amount of heat that can be recovered from the refrigeration system is about 1.3 times (130%) the cooling demand in the system. However the analysis of the field measurements shows that only between 30-70% of the available heat to be recovered is utilized, the rest is released to outdoors. $COP_{1\text{-}HR}$ can be compared to COP of normal ground source heat pump due to its specific definition. The results show the $COP_{1\text{-}HR}$ of refrigeration system is quite competitive with heating COP of normal ground source heat pump.

3.5. System performance - annual modelling (Karampour and Sawalha, 2014a)

The controls strategy discussed in the previous sections are applied to model the annual performance of a CO2 trans-critical booster system. In these calculations modelling of the flash gas by-pass line is added.

The total monthly cooling loads in MT cabinets (Q_{MT}) and LT freezers (Q_{LT}) are shown in Figure 18, with negative values. The total monthly rejected heat in the gas cooler ($Q_{gascooler}$) and recovered heat in the de-superheater (Q_{HR}) are shown, as well. The total amount of electricity use in high stage compressors (E_{HS}), low stage compressors (E_{LS}) and gas cooler fans (E_{fan}) can be read from the right vertical axis. Average monthly ambient temperatures are shown in the horizontal axis. Total annual electricity use to provide the cooling and heating demands is calculated 536 MWh electricity. 79% (422 MWh) of the total energy use is dedicated to high stage compressors, 14% (76 MWh) is the fraction for low stage compressors and the remaining 7% (38 MWh) is used for gas cooler fans.

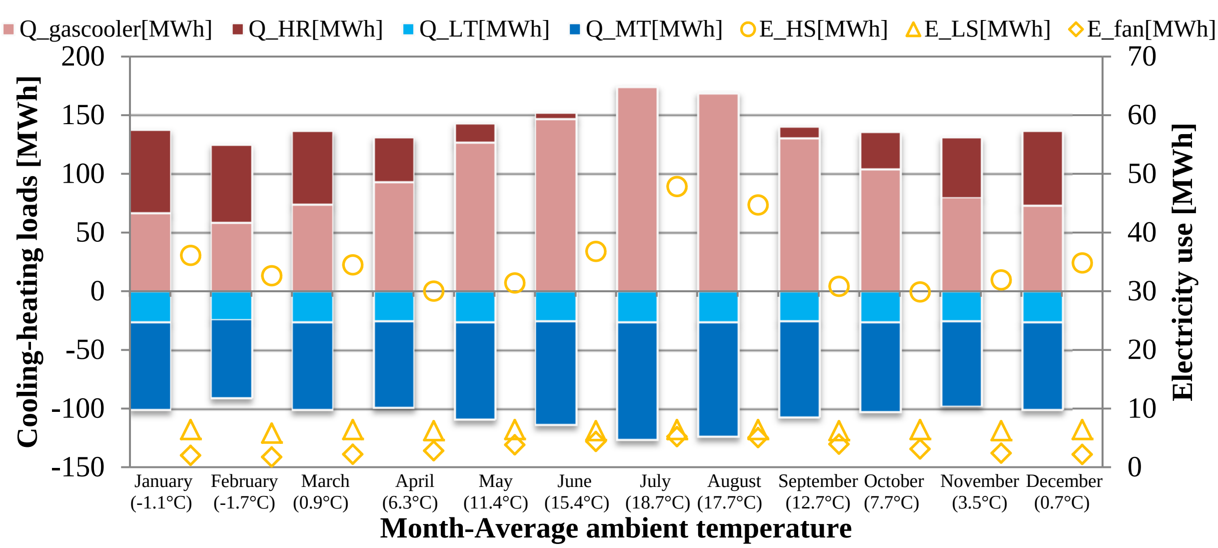

Figure 18: total monthly cooling-heating loads (left axis) and electricity use (right axis)

The COPs of medium temperature level (COP_{MT}) and COP for heat recovery (COP_{HR}) are shown in Figure 19. The corresponding discharge pressure (P_{disch}) and gas cooler exit temperature (T_{gce}) are shown for an ambient temperature range of -20° to +30°C. The ambient temperature -9.86°C is the shift point from first step of heat recovery with pressure regulation and gas cooler full capacity to second step with fixed max pressure (88.5 bars) and decreasing gas cooler fan speed and sub-cooling. This can be observed by the increasing gas cooler exit temperature for ambient temperatures lower than -9.86°C. As can be seen, COP_{MT} fluctuates between 1.5-4 with a peak at 10-11°C where the discharge pressure has the lowest value. COP_{LT} is not shown in the figure but has the same trend as COP_{MT} and varies between 1-2.

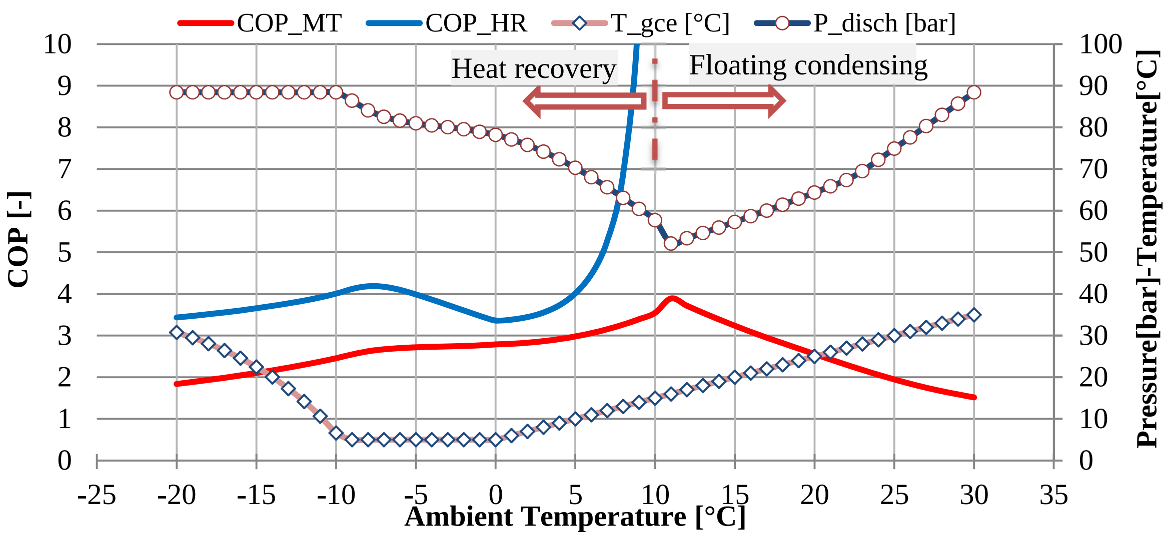

Figure 19: COP_{MT}, COP_{HR}, discharge pressure [bar] and gas cooler exit temperature [°C]

COP_{HR} trend can be explained by the ratio between the total heating demand and extra compressor work. When heat recovery starts at 10°C, the amount of heat available in floating condensing ($Q_{HR,fc}$) is close to the heating demand and minor/no extra power consumption is needed. This explains the high COP_{HR} values at high ambient temperatures. COP_{HR} trend in lower temperatures, minimum value at 0°C (COP_{HR}=3.3) and local peak value at -8°C ambient temperature (COP_{HR}=4.2) reflect the rates of change in total recovered heat (Q_{HR}) versus extra compressor work for heat recovery (E_{HS} - $E_{HS,fr}$).

Seasonal performance factor (SPF) is another performance indicator that is commonly used to compare heat pumps. It is defined as the sum of annual heat supply over annual sum of electricity consumption. SPF for a heat pump with capacity of 150-200 kW in Swedish climate ranges between 2.2-3.3, depending on the heat source (Granryd, 2005). The trans-critical booster system provides 417 MWh of annual heating demand by consuming 104 MWh electricity. Therefore, SPF of the system is 4 using equation (5), which is higher than conventional heat pumps.

$$SPF = \sum Q_{HR} / \sum (E_{HS} - E_{HS\_fc}) \qquad \text{Eq(5)}$$

where E_{HS_fc} [kW] is the amount of power consumption of high stage compressors in floating condensing mode and E_{HS} is the power consumption in heat recovery mode where the pressure is increased to cover the heating demand. $E_{HS} - E_{HS\_fc}$ shows the difference between real consumption in heat recovery mode and virtual consumption if the system was run with floating condensing mode for low ambient temperatures.

By heat recovery from the refrigeration system, the cost of installing a separate heating system is cut, as well.

To summarize, this computer simulation model has been built to analyze a state-of-the-art refrigeration system for simultaneous heating, cooling, and air conditioning. The model allows for calculating for several system configurations, in different running modes and boundary conditions.

3.6. Study on performance of internal heat exchangers in a CO2 booster system (Karampour and Sawalha, 2014b)

This study evaluates the application of internal heat exchangers (IHE) in a CO2 trans-critical booster system. As shown in Figure 20, four IHE A, B, C and D with 50% effectiveness have been examined in 8 configurations of A, AC, AD, B, BC, BD, C and D. These configurations performance are compared with a reference “NO IHE” case. The compared parameters are total cooling COP (COP_2), total cooling and heating combined COP (COP_{tot}) and the amount of recovered heat (Q_{HR}). Boundary conditions of the model are similar to section 3.5. The calculations are done both for a system *with by-pass line (WBP)* and *without by-pass line (WOBP).*

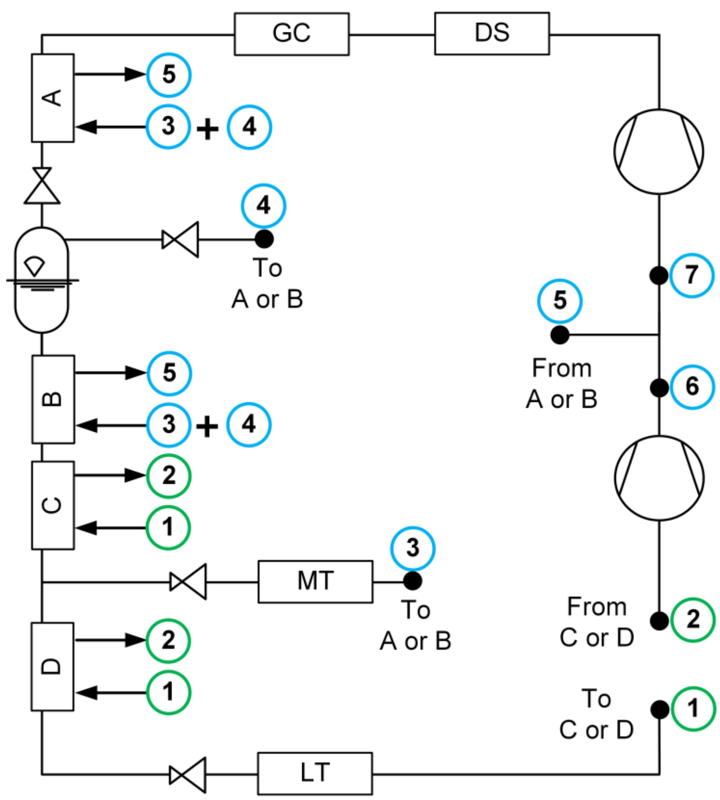

Figure 20: Internal heat exchangers arrangement in a CO2 booster system

Results of the calculation for 65bar discharge pressure are shown in Figure 21. Considering COP_2, no significant difference is seen between different arrangements. However, using IHE A, AC or AD has a positive impact on heat recovery potential of the system. The main reason for this is the high amount of MT superheating that is occurred by applying these IHEs. For a system without by-pass use of IHEs B, BC or BD increases the amount of available heat by about 15%.

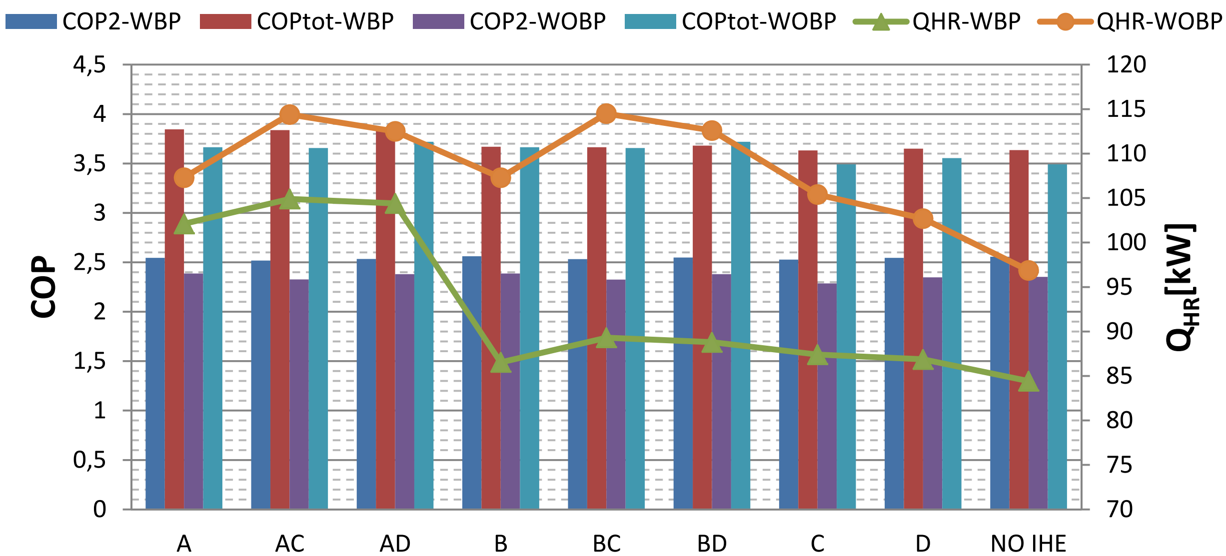

Figure 21: COP_2, COPtot and QHR for nine IHE configurations and two with by-pass (WBP) and without by-pass (WOBP) system designs @65bar

Taking the “NO IHE” with by-pass as the reference, COP_{tot} values normalized for 85kW heat recovery and the amount of COP_{tot} change is shown in Figure 22. It can be seen in a system with by-pass line using IHE A configurations, A-AC-AD, can increase COP_{tot} up to 12%. IHE B configurations increase COP_{tot} up to 11% in a system without by-pass line. The COP increase is due to the fact that with IHE more heat can be recovered which means that the system will operate at lower discharge pressure to recover the same amount of heat.

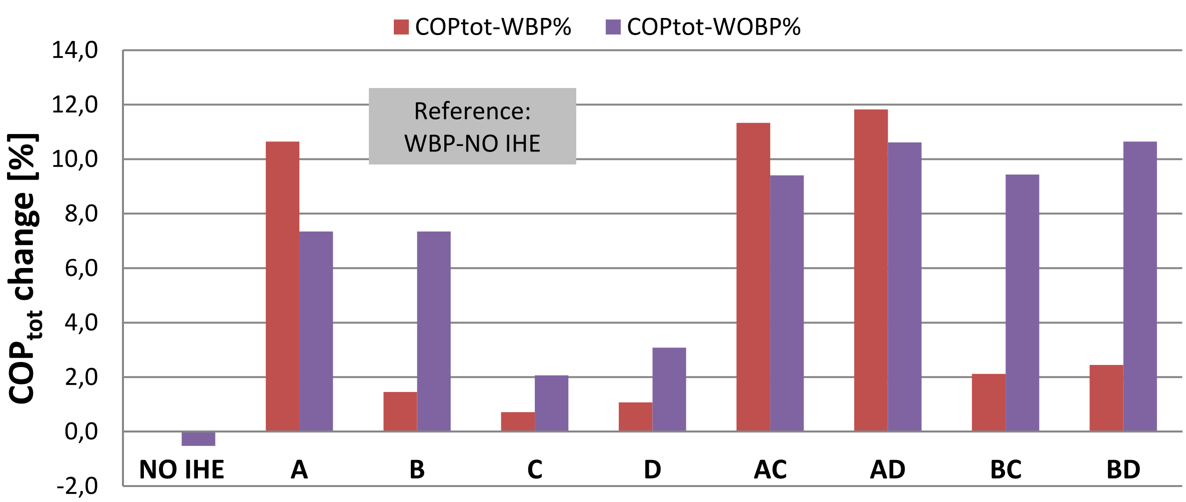

Figure 22: COP_{tot} change -65 BAR- With and Without By-pass - normalized (QHR=85kW]

4. State-of-the-art system: System definition and case studies

According to this research project CO2 trans-critical booster system is found to be the state-of-the-art system in the supermarket refrigeration field for Sweden climate condition. A simple schematic of this system is shown in Figure 23. However, some modifications and improvements can be applied which are discussed in the following sections.

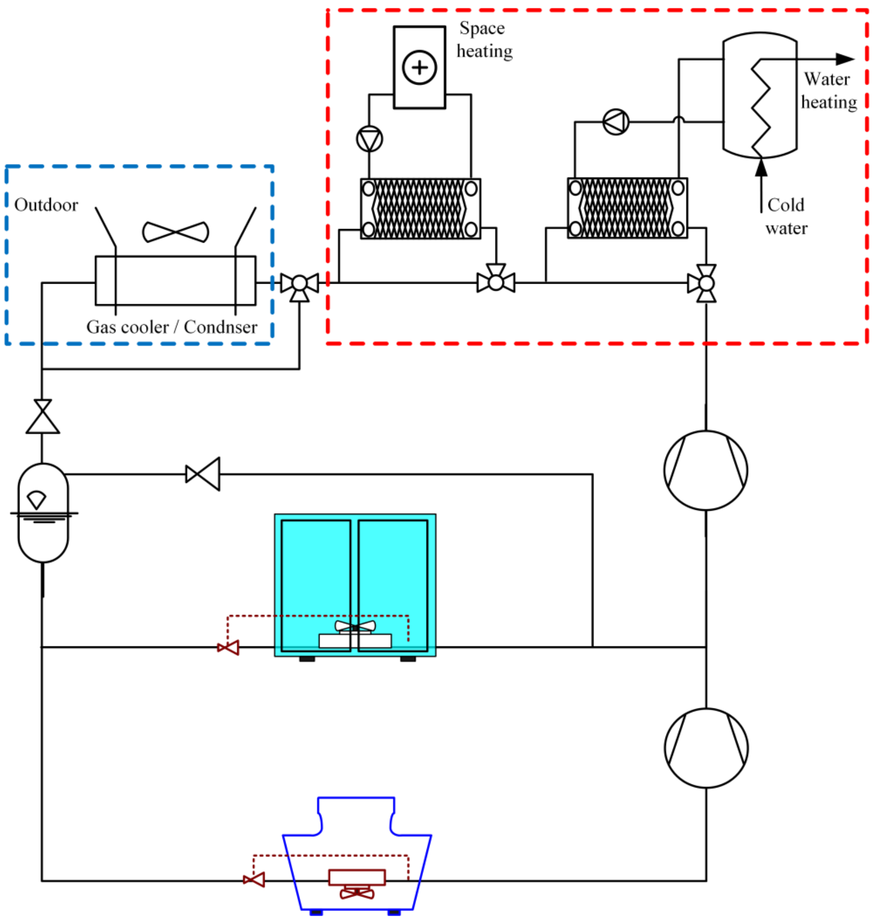

Figure 23: Schematic of a state-of-the-art CO2 trans-critical booster system

Characteristics of this system and its integration into the supermarket energy systems can be reviewed in the following categories:

4.1. Refrigeration

- In a state-of-the-art CO2 trans-critical booster system the fraction of the vapour into the evaporator coils is designed to be as low as possible. There are different methods applied to decrease this vapour fraction. The standard solution is flash gas removal from the receiver at intermediate pressure. Intermediate pressure level should be as close as possible to the pressure of the medium temperature level. A by-pass line conducts the vapour in the receiver to high stage compressors suction line, expanding through a by-pass valve. An alternative method, reported to be more energy-efficient (Bella and Kaemmer, 2011), is parallel compression. The vapour is compressed directly to the high pressure level by a parallel compressor. The effect of parallel compression is more significant when there is considerable amount of pressure difference between receiver pressure and pressure of MT level.

In section 3.6 it has been shown that using an internal heat exchanger (IHE) in a system with flash gas by-pass doesn't have a significant impact on the refrigeration system performance. However, it can provide a more fraction of liquid feeding the evaporators. An alternative solution seen in the market is to use a cooling coil internal heat exchanger inside the receiver to condense a portion of the vapour.

- Reduction of parasitic heat loads on cabinets and freezers is crucial in performance of the state-of-the-art system. Using glass doors or LED lighting in LT freezers and MT cabinets is well-established examples of reducing these loads.
- Heat rejection from the discharge line of LT compressors can be considered as an efficient method to reduce the MT compressors inlet temperature (higher density vapour) and, consequently, MT compressors power consumption, refer to section 4.5.2, HE100.

4.2. **Heat recovery**

- CO2 trans-critical booster system is able to provide the entire heating demands in an average size supermarket in Sweden. Consequently, this system should be considered as the primary heating system. If the provided heat is not sufficient, for example in larger supermarkets or during low-refrigeration-duty periods, an integrated ground source heat pump using parallel compression is a proper auxiliary heater. Using an integrated evaporator in the gas cooler as a heat source for this heat pump is an alternative technique applied by some companies. The heat pump evaporator is integrated in condenser/gas cooler and the method is called "false load".
- Heat recovery from the LT compressors discharge reduces the heating load on high stage de-superheater(s) which allows the system to operate at lower discharge pressure for the same amount of recovered heat. This method is more effective when there is a high amount of suction line super-heating (i.e. IHE) in low pressure side.

Control strategy for heat recovery is discussed in detail in section 3.3. The steps are in brief: 1-gas cooler full capacity and pressure regulation. 2-fixed max discharge pressure and sub-cooling decrease. 3- Switching off gas cooler fans, or gas cooler by-pass. 5- Auxiliary heater. The reason for using the auxiliary heater as the last choice is high COP and SPF of CO2 booster heat recovery system (see section …). Regarding heating COP, Sawalha (2013) showed for a R407 heat pump with heat source temperature of 5°C providing water at 35°C, the heating COP is about 3,8 while CO2 system can provide the heat with a COP of higher than 4 for a wide range of cold ambient temperature, refer to Figure 15. Furthermore, following the suggested control strategy and with 30°C heating system return temperature, a SPF of 4 is calculated for CO2 system (Karampour and Sawalha, 2014a) while the average SPF of 56 ground source heat pumps is reported to be 3.88 (Miara et al., 2011).

- Different heating demands in supermarkets require different temperature levels. Heat can be recovered more efficiently in CO2 refrigeration system with more than one heat exchanger (de-superheater). This helps to avoid the pinch point and to extract the heat more efficiently considering the isobar shape of CO2 in a T-s diagram.
- The system should be able to provide the heating demands at proper required temperatures. Tap water heating should be kept higher than 50°C to avoid legionella growth and lower than 60°C to avoid scalding risk according to Swedish building regulations (reference). The right temperature for space heating depends on the system used and the ambient temperature: radiators, coils in the air handling unit, floor heating or a combination of them. The system should be designed for the lowest possible return temperature from the heating system.
- Although it has been seen that using of an IHE doesn't have a significant positive or negative impact on cooling side, higher amount of superheating in the compressors suction line can increase the amount of available heat for heat reclaim.

4.3. Control

- Control unit of a CO2 trans-critical booster system should have the possibility for communication with the HVAC systems control to run in harmony. This is important when the CO2 system provides heating or air conditioning for the HVAC system.
- Receiver pressure control is crucial to ensure proper liquid feed to the evaporators. This is done typically by a motorized by-pass control valve or a parallel compressor. The receiver pressure should be kept as close as possible to the pressure of the medium temperature level.
- Gas cooler pressure in floating condensing mode should maintain the lowest possible approach temperature in sub-critical region. Pressure must be regulated

in the super-critical region to achieve optimum COP. This is done by a high pressure regulating valve after the gas cooler. Correct gas cooler exit temperature reading is vital and the sensor should be placed as close as possible to gas cooler exit. The recommendations in section 3.3 should be applied in the heat recovery mode.

- One compressor in the compressor packs and gas cooler fans should be equipped with frequency converters instead of the traditional on-off control.
- Evaporator control uses electronic expansion valve to maintain a certain amount of superheating. Some auxiliary controls including fans control, night blind, anti-sweating heaters, light control and demand base defrosting instead of time-scheduled are examples of auxiliary control for case controller in freezers and cabinets.

4.4. Monitoring, data collection and performance analysis

- Access to real-time measurements is an important feature of a state-of-the-art system. The measurements should be accessible through web-based monitoring. Furthermore, the measured data should be available to collect for deeper studies.
- The measured data should be synchronized and for same time intervals. One of the web-monitoring softwares used in this project doesn't have synchronized measurements and this data synchronization is a time consuming process.
- The new state-of-the-art monitoring systems provide some performance indicators including COP and cooling capacity. These performance indicators can be used alarming about the failure of system in energy-efficient performance. In this project a simple and accurate method is introduced for system performance analysis in section 3.2: An alternative method to evaluate refrigeration system performance.
- Guidelines of how to instrument, measure and evaluate refrigeration systems in supermarkets is a master thesis done in this project (Giménez Gavarell, 2011). This thesis investigates the important parameters required for a proper system analysis. It is available online on the project research website, “publications” section.
- Separate power consumption measurement of main electricity consumers, i.e compressors, is a must in a state-of-the-art system.

KTH research group is active in some other research projects, indirectly connected to development of this state-of-the-art system in the market. Our team joined Annex 44, which deals with performance indicators in supermarkets. Also we joined Horizon2020 application on Energy Efficient Supermarket Platform; proposal is submitted the goal is removing market barriers for applying energy efficient supermarket systems with integrated cooling, heating and air conditioning.

4.5. **Case studies**

The presented supermarkets in the following sections have some features of a state-of-the-art system.

4.5.1. Case study 1: Sollefteå, Sweden

The supermarket is located in mid-north Sweden and started to work November 2013. The refrigeration system consists of three MT compressors, two LT compressors and two parallel-air conditioning compressor.

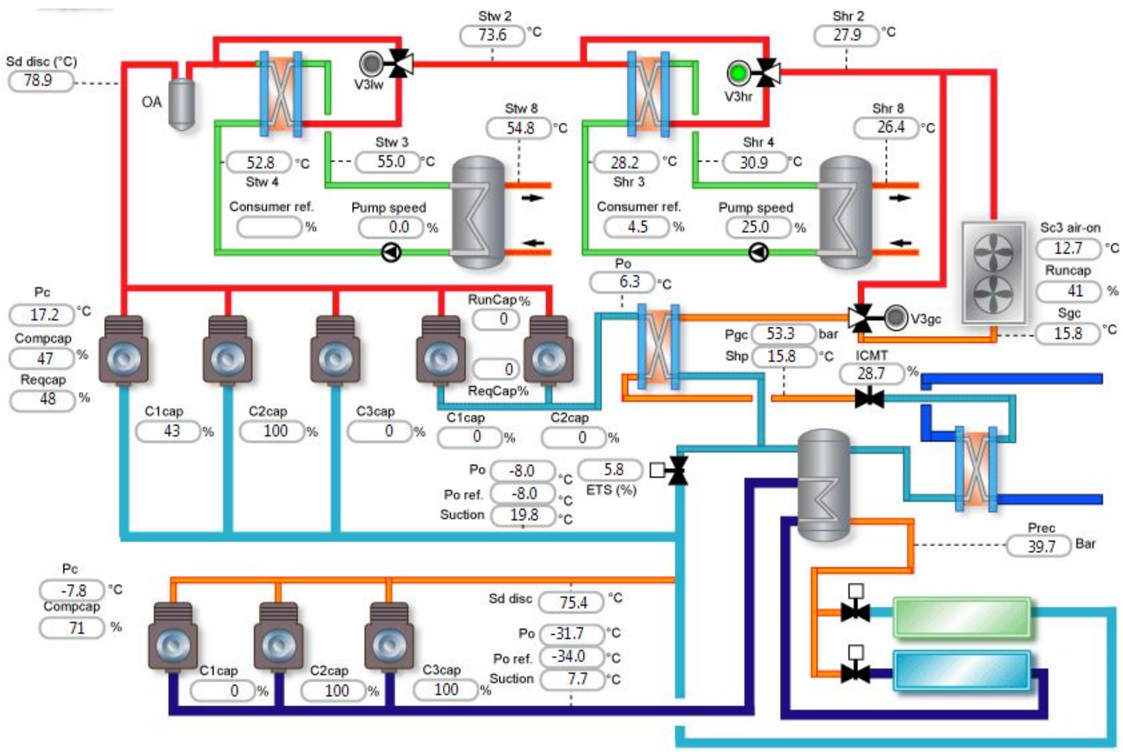

Figure 24: Web-monitoring interface of Sollefteå supermarket

Some state-of-the-art characteristics of this system are:

- Integrated solution: The CO2 trans-critical booster system provides various cooling and heating demands including LT and MT refrigeration, space and tap water heating and air conditioning. The auxiliary heating system is district heating.
- Refrigeration
    - o Reduction of vapour fraction: Flash gas by-pass + parallel compression + internal IHE inside the receiver
- Heat recovery: a large portion of the rejected heat is recovered for (I) space heating (radiators, air handling units, floor heating, entrance air curtain and snow

melting) and (II) tap water heating. Some features of the heat recovery system are:

  - three 3-way valves for enrolling or by-passing (I) tap water heating heat exchanger, (II) space heating heat exchanger and (III) gas cooler/condenser
  - 55°C set point temperature for tap water heating and a variable set point temperature, function of ambient temperature, for space heating

- Control and monitoring:
  - Frequency-controlled compressors and fans: (I) one LT compressor (II) one MT compressor (III) one AC compressor (IV) Gas cooler fans
  - Whole case (evaporators) and pack (compressors rack) controllers including optimum high pressure discharge control for refrigeration and heat recovery, receiver pressure control, evaporators control (liquid injection, defrost, light, fan, anti-sweating heater, …), oil management, …
  - IWMAC real-time measurements web-monitoring; however, the measured data are not synchronized and a data processing is required to normalize the measurements intervals into common time intervals.
  - COP and cooling capacity calculation/monitoring

What could be added to this system is heat recovery form the low stage compressors discharge. There is a considerable amount of super-heating in the suction line and this leads to high-temperature discharge flow.

4.5.2. Case study 2: Evanston, Illinois, USA

This retail store sells both food and non-food products, mainly pharmaceuticals. The store is opened in November 2013. It is awarded as US first net zero energy store and various sustainable energy solutions including 850 roof-top PV panels and two wind turbines are used in the building. A triple-level temperature trans-critical CO2 refrigeration system is selected as the heart of the system for cooling and heating.

The CO2 refrigeration system serves the LT freezers, MT cabinets, and provides both heating and chilled water to the store’s HVAC systems. Waste heat from the system also feeds a pre-heat tank for domestic hot water. The system has a refrigeration gas-cooler but also uses eight 150-meters deep geo-exchange wells to store heat for use during heating season.

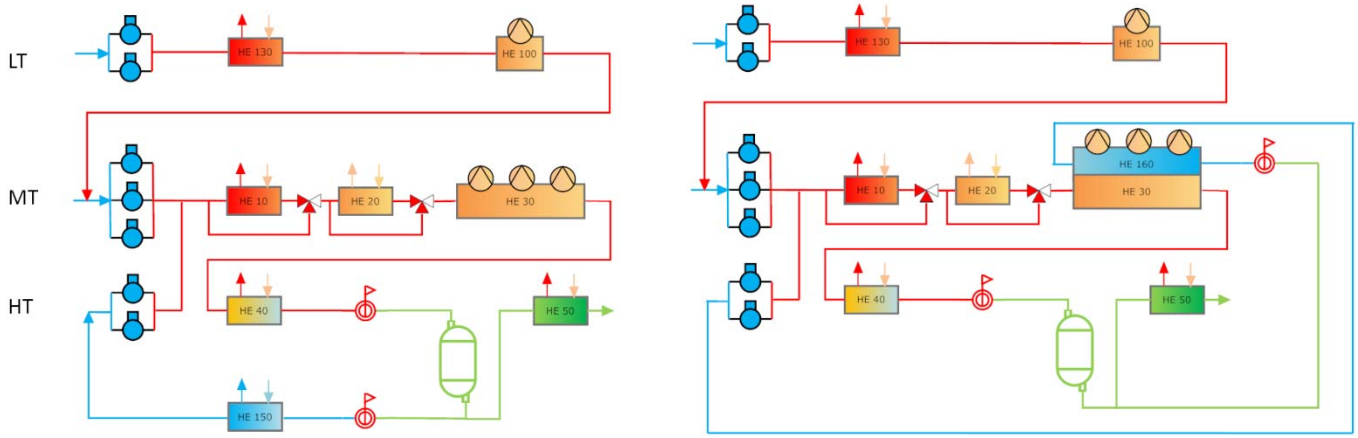

Figure 25: System and heat exchangers configuration of Evanston's store; summer mode (left) and winter mode (right)

The system is designed to have a full set of heat exchangers including evaporators for MT and LT refrigeration, tap water heating (HE10), space heating (HE20), air conditioning (HE150), borehole subcooling and thermal storage (HE40), LT compressors discharge heat recovery (HE130) and gas cooling (HE100), IHE after receiver (HE50), gas cooler (HE30) and false load heat exchanger (HE160) in the winter period. The HT compressor works for air conditioning in summer and for providing heat from the false load HE in the winter.

Some state-of-the-art characteristics of this system are:

This system has several similar characteristics to the previous supermarket. It includes an integrated refrigeration and heat recovery solution for responding to cooling and heating demands in the store. Air conditioning is provided by parallel compression. The measurements and COP indication are available via web-monitoring.

Some differences of this system and Sollefteå supermarket are using ground thermal storage, IHE after receiver and heat recovery/gas cooling in the LT compressors discharge.

4.5.3. Case study 3: Trondheim, Norway

The supermarket started its operation in August 2013. Several energy saving measures are applied in the design of the supermarket aiming to reduce the energy consumption by 30%, in comparison with a standard Norwegian supermarket. The supermarket chose a CO_2 trans-critical booster system to provide the cooling and heating demands.

The rejected heat from the refrigeration system is recovered in two gas coolers. These heat exchangers transfer the heat to two high and low temperature glycol loops. The recovered heat is used in the air handling units, entrance air curtain, floor heating and snow melting.

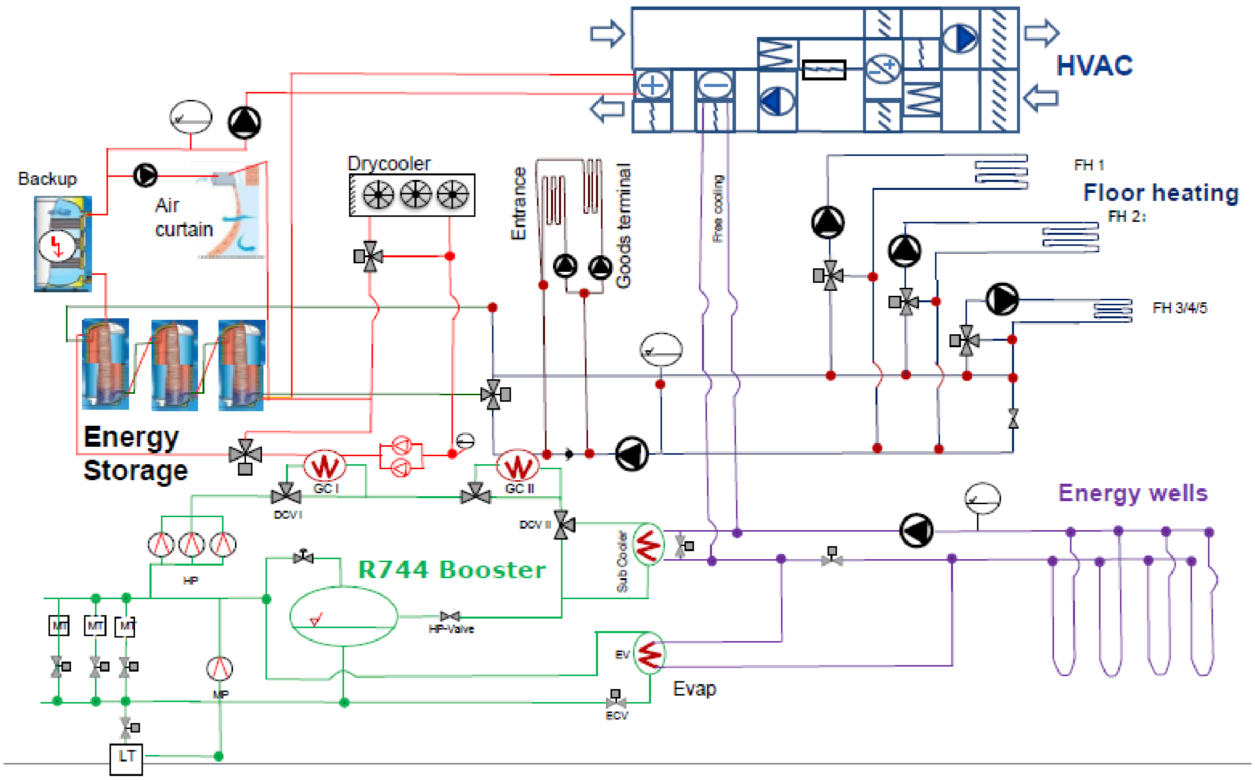

Figure 26: Refrigeration, heating and air conditioning schematic of Trondheim supermarket (Funder-Kristensen, 2013)

Four 170-meters boreholes are applied as the heat sink for refrigeration sub-cooling in summer season. The stored heat will be an extra heat source for CO2 system heat-pumping function if the absorbed heat in the freezers and cabinets would not be sufficient in winter season. Furthermore, the boreholes can provide free cooling and dehumidification for the AHU unit in summer time.

Some state-of-the-art characteristics of this system are:

Ground source summer sub-cooling and winter heat pumping – Heat recovery with two heat exchangers – usage of several 3-port valves to by-pass different loops – use a holistic control system for the refrigeration system – refrigeration and HVAC control systems communication – speed control of compressors and fans

Some notes about the system: -The use of dry cooler will mean that there is an additional temperature difference.

-The heating demand in the winter is quite high in this supermarket, it is tried try to reject all the heat in the ground during the summer time and use it during winter time.

-The ground source is used to cover the air conditioning load.

4.5.4. Case study 4: Rasttat, Germany

The last case study is a supermarket operated by German supermarket chain Aldi Sud and started to work in Rasttat 2010. The supermarket is considered as a prototype project and funded by German Federal Ministry of Economics and Industry to examine/indicate the possibilities of radical primary energy reduction, in this project 30% (Rehault and Kalz, 2012).

Among many innovative solutions, a CO2 trans-critical booster system with heat recovery was selected as an efficient and integrated solution. The system is coupled to six 100-deep boreholes to improve the refrigeration efficiency by sub-cooling in summer. A parallel CO2 compressor uses the stored ground heat in winter to enhance the heat pumping function of CO2 system.

Breakdown of consumption in a standard branch, the target case and measured 2010 and 2011 values can be seen in Figure 27.

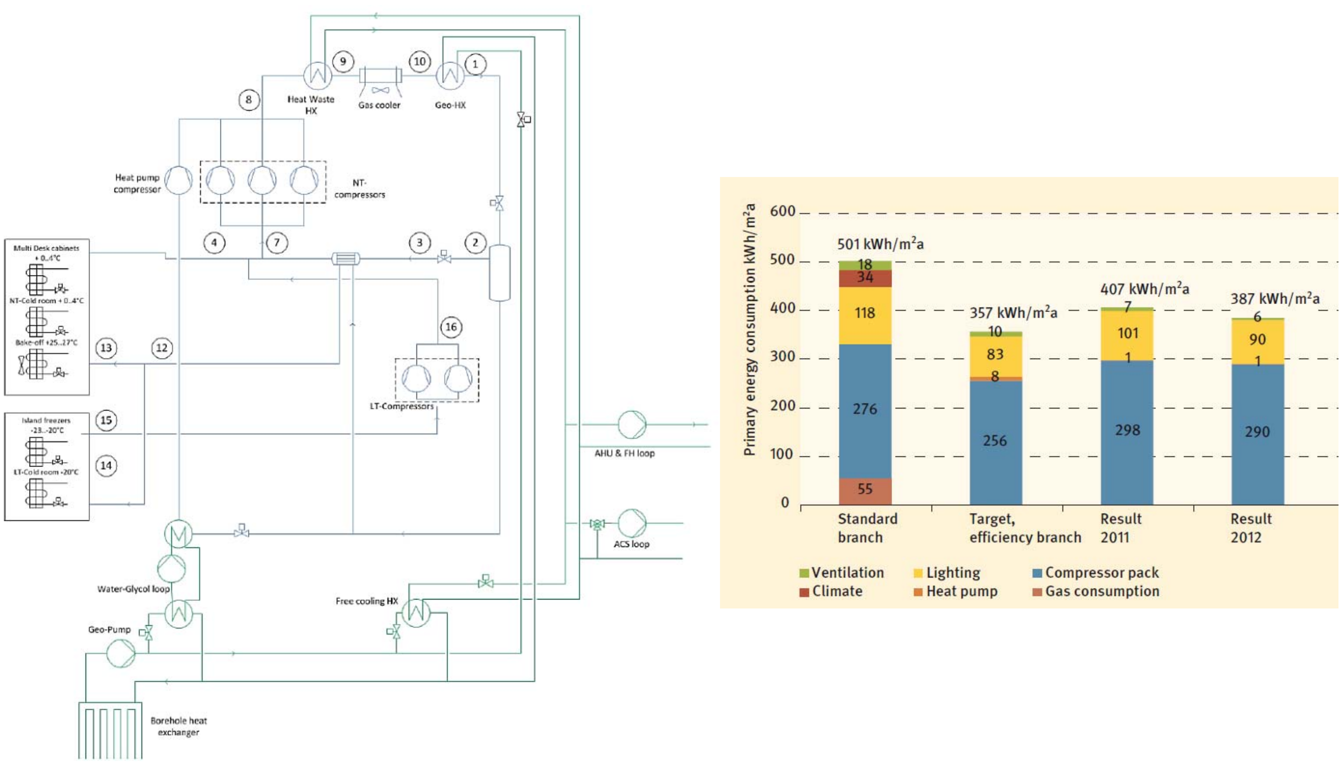

Figure 27: Rasttat supermarket system schematic (left) and breakdown of specific energy consumptions (right)

Some state-of-the-art characteristics of this system are:

Integrated solution for refrigeration and heat recovery – Parallel compression – auxiliary heater: ground source heat pump with parallel CO_2 compressor – free ground source cooling – flash gas by-pass – internal heat exchanger with by-pass line – performance monitoring

5. Conclusion

- Refrigeration

Guideline of how to instrument, measure and evaluate refrigeration systems in supermarkets is developed.

Field measurements of eight supermarkets (5 using CO2, 3 HFC) from a previous project is summarized. It is shown that while older CO2 systems have lower COP in comparison with a conventional HFC system, newer systems have comparable or better performance in moderate-cold climates like Sweden.

Control strategies used in supermarket refrigeration system, with focus on CO2 booster systems, is studied.

An alternative method for on-site COP and cooling power calculation/visualization is developed. The results show good accuracy with 5-10% maximum difference between conventional method and this method.

Study on the parameters influencing the system efficiency is done by computer modeling and field measurements analysis. The key parameters are identified in system design level and components performance level.

- Heat recovery

A heat recovery control strategy is suggested for CO2 booster system to have the optimum system performance while covering the entire heating demands of an average size supermarket in Sweden.

Following the suggested strategy, it is shown by modelling that CO2 system with heat recovery consumes 6 % less energy annually in comparison with a conventional R404 system with separate heat pump for heating demands.

Two supermarkets are analysed by field measurements. In one of the supermarkets it is shown that while available heat to recover is 130% of the cooling capacity, only 30-70% of the heat is recovered.

A computer modelling of annual performance of a CO2 booster system showed that CO2 systems can provide the entire heating demands in an average size supermarket with a high seasonal performance factor of 4, more than a majority of conventional heat pumps with same capacity.

- State-of-the-art system

Definition and characteristics of a state-of-the-art system is studied. The recent improvements in system design and case studies of some highly energy-efficient supermarkets using this system have been discussed and the state-of-the-art system solution to be installed in Sweden has been suggested.

6. Suggestion for future work

- Addition of building HVAC measurements to field measurement analysis
- Deeper study of integration of air conditioning system into refrigeration system
- Deeper study of hydronic system and water accumulator tank storage transferring the heat from CO2 system to HVAC system and tap water system

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8. Scientific publications

- Investigation of heat recovery in CO2 trans-critical solution for supermarket refrigeration (Sawalha, 2013)

- Field measurements and performance evaluation of CO2 supermarket refrigeration systems (Karampour et al., 2013)

- Performance and control strategies analysis of a CO2 trans-critical booster system (Karampour and Sawalha, 2014a)

- Heat recovery investigation of a supermarket refrigeration system using carbon dioxide as refrigerant (Abdi et al., 2014)

- A study on energy efficiency improvement of CO2 trans-critical booster system; focus on suction gas heat exchanger arrangements (Karampour and Sawalha, 2014b)

- A paper is submitted to Applied Thermal Engineering journal titled: “Field Measurements of Supermarket Refrigeration Systems. Part I: Analysis of CO2 trans-critical refrigeration systems”, S.Sawalha, M. Karampour, J.Rogstam, 2014

9. Popular science publications, oral and poster presentations

9.1. Popular science publications

- Värmeåtervinning med CO2 i livsmedelsbutikers kylaanläggningar (Sawalha and Abdi, 2013)

9.2. Oral presentations

- Jörgen Rogstam presented the project on Effsys day- October 2012, Göteborg, Sweden

- Samer Sawalha presented the research results that have been doen in the period 2009-2011. It is presented in ATMOsphere 2012 Conference – November 2012, Brussels, Belgium.

- Mazyar Karampour presented the conference paper written by him, Samer Sawalha and Jörgen Rogstam in 2nd IIR Cold Chain Conference- April 2013, Paris, France.

- Samer Sawalha, Creativ workshop, 23 October 2012

- Mazyar Karampour presented the conference paper written by him and Samer Sawalha in 3rd IIR Cold Chain Conference- June 2014, London, UK.

- Amir Abdi 2014 will present a paper in Gustav Lorentzen Conference, Hanghzou, Cina, August 2014

- Mazyar Karampour 2014 will present a paper in Gustav Lorentzen Conference, Hangzhou, China, August 2014

9.3. Poster Presentations

- EFFSYS day - 14 October 2011, Göteborg, Sweden

- ITM PhD Workshop - 22 May 2012, Stockholm, Sweden

- KTH Energy Dialogue day - 22 Nov 2012, KTH, Stockholm, Sweden

- EFFSYS day - 18 October 2013, Göteborg, Sweden

- KTH Energy Dialogue day - 11 Nov 2013, KTH, Stockholm, Sweden

10. Appendix

10.1. Master thesis and summer internships

A list of publications from KTH research team, including some of the master theses is available at: http://www.kth.se/en/itm/inst/energiteknik/forskning/ett/projekt/co2-suprmarket-refrigeration/publications-1.301166

10.1.1. Master theses

- Experimental and field measurements analysis of heat recovery in supermarket refrigeration systems with CO2 as refrigerant, Amir Abdi, Master Thesis, 2013

- Investigating control strategies in supermarket refrigeration systems, Bisrat Girma Taddesse, Master Thesis, 2013

- Optimization and calculation of supermarket refrigeration systems, Vincent Cottineau, Master Thesis, 2011

- Guidelines of how to instrument, measure and evaluate refrigeration systems in supermarkets, Pau Giménez Gavarell, Master Thesis, 2011

- Analysis of simultaneous cooling and heating in supermarket refrigeration systems, Johan Marigny, Master Thesis 2011

10.1.2. Summer internships

- Performance Analysis of CO2 Refrigeration Systems in Supermarkets, Julien Pondaven, 2012

- Performance Analysis of CO2 Refrigeration Systems in Supermarkets, Amélie Masclaux, 2012

- Evaluating Pack Calculation II software, Florence Rodriguez, 2013

- Field measurement analysis of a CO2 trans-critical booster system, Laurene Messager 2014

- Field measurement analysis of a CO2 trans-critical booster system, Simon Martinez, 2014

10.2. Attachment of scientific publications

A separate document contains the main publications of this project is sent to Effsys+ board. This document is not possible to be published on web due to copy rights restrictions.