Field Measurements and Evaluation of CO2 Refrigeration Systems for Supermarkets

JOHAN KULLHEIM

Master of Science Thesis
Stockholm, Sweden 2011
Field Measurements and Evaluation of CO2 Refrigeration Systems for Supermarket

Johan Kullheim

Master of Science Thesis Energy Technology 2011:113
KTH School of Industrial Engineering and Management
Division of Applied Thermodynamics and Refrigeration
SE-100 44 Stockholm
Field Measurements and Evaluation of CO2 Refrigeration Systems for Supermarkets

Johan Kullheim

<table>
<thead>
<tr>
<th>Approved Date</th>
<th>Examiner</th>
<th>Supervisor</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Prof. Björn Palm</td>
<td>Dr Samer Sawalha</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Commissioner</th>
<th>Contact person</th>
</tr>
</thead>
<tbody>
<tr>
<td>Installatörernas Utbildningscentrum and Sveriges Energi och Kylcentrum (IUC &amp; SEK)</td>
<td>Jörgen Rogstam</td>
</tr>
</tbody>
</table>
Abstract

This thesis is a part of a larger project initiated by Sveriges Energi & Kylcentrum (SEK) and Installatörernas Utbildningscentrum (IUC) in cooperation with Royal Institute of Technology (KTH). The project aims at evaluating the performance of CO2 refrigeration systems for supermarkets in Sweden and has been co-financed by the Swedish Energy Agency.

In this thesis, three supermarket refrigeration systems are evaluated; two trans-critical DX CO2 systems and one R404A/CO2 cascade system. Field measurements of energy consumption, temperatures, pressures, and compressor motor frequency are made using the computer programs IWMAC [IWM10] and LDS [LDS10]. Calculations are performed using the program Refprop [NIS10] and performance data from the compressor manufacturer. The cooling capacity and coefficient of performance (COP) are calculated and the analysis is made for a period of four months during the summer, June-September 2010, for all systems.

The results show that the cascade system has a significantly lower value of COP than the trans-critical CO2 systems. This is due to the unusual design and control of the cascade system. The COP also drops because of the extra temperature levels and pumps present in this system. The use of floating condensation and the fact that no pumps are required for refrigerant circulation is beneficial for the trans-critical systems, resulting in higher COP. The COP of the systems is dependent on the ambient temperature, mainly for the medium temperature stage, showing the lowest values during the warmest summer months. Working with different operating parameters and ambient conditions, the total COP for the cascade system was about 1.8 compared to 2.7 and 2.8 for the two trans-critical installations. When using the same load ratio for all three systems, the cascade system had a total COP of about 1.7 and the two trans-critical systems achieved values of 2.7 and 2.9 respectively.

The operation of the heat recovery systems is investigated and the results suggest some differences in operation between the two trans-critical systems. There seems to be no heat recovery at all for the cascade system during the time of the study. However, a more thorough analysis of the heat recovery including the cold winter months is required in order to draw conclusions.

The energy consumption and influence of parasites like pumps and fans is analyzed and the results show that about 18 % of the total energy consumption is lost to parasites in the cascade system (including the dry cooler and all the pumps). This is much more than for the two trans-critical systems at 7 % and 1 %, where the only parasites are the gas cooler fans. The total COP when excluding the influence of parasites becomes 2.1 for the cascade system while the trans-critical systems reach values of 2.8 and 3.0. When using the same load ratio without the influence of parasites, significant change can only be observed for one of the trans-critical systems where the COP increases from 3.0 to 3.2
Acknowledgements

This master’s thesis has been a part of a project initiated by Sveriges Energi & Kylcentrum (SEK), Installatörernas Utbildningscentrum (IUC) in cooperation with Royal Institute of Technology (KTH). It has been managed by IUC and financially supported by the companies ICA, Green and Cool, Partor, Carrier, Ahlsell, Huurre, Wica Cold, Cupori, Woodley and IWMAC. The project was also funded by the Swedish Energy Agency.

First of all, I would like to express my sincere gratitude to Jörgen Rogstam, the project manager in SEK, and to my supervisor Dr Samer Sawalha for their support throughout this project. You helped me with calculations, gave me advice and new ideas as well as inspired and motivated. It was a pleasure to work with both of you, thank you.

Second, I would like to thank all project partners that have been involved. I would also like to extend my gratitude to Professor Björn Palm at the Applied Thermodynamics and Refrigeration Division of the Energy Technology Department in Royal Institute of Technology (KTH) for his support.
# Table of Contents

Abstract .................................................................................................................. I
Acknowledgements ................................................................................................. II
Table of Contents .................................................................................................... III
List of Figures .......................................................................................................... V
List of Tables ........................................................................................................... VII
Nomenclature ......................................................................................................... VIII

1 Introduction ......................................................................................................... 1
   1.1 Background .................................................................................................. 1
   1.2 Energy use and refrigerant emissions ....................................................... 1

2 Objectives ......................................................................................................... 4
   2.1 Specific Objectives ..................................................................................... 4
   2.2 Project ......................................................................................................... 4

3 CO2 Refrigeration .............................................................................................. 6
   3.1 Properties of CO2 ....................................................................................... 6
   3.2 Design options ............................................................................................ 7
      3.2.1 Indirect System ................................................................................... 7
      3.2.2 Cascade DX System ........................................................................... 8
      3.2.3 Trans-critical DX System ................................................................. 9

4 System descriptions .......................................................................................... 11
   4.1 Cascade system CC2 .................................................................................. 11
      4.1.1 Overall system description ............................................................... 11
   4.2 Trans-critical systems TR4 and TR5 ....................................................... 14
      4.2.1 Overall system description for TR4 and TR5 ................................... 14
      4.2.2 P-h diagram for TR4 and TR5 ......................................................... 18

5 Methods of Evaluation ...................................................................................... 19
   5.1 Cascade system CC2 calculation ............................................................... 19
      5.1.1 Data acquisition for CC2 ................................................................. 19
      5.1.2 Calculation of main parameters CC2 ............................................. 21
   5.2 Calculations for trans-critical systems TR4 and TR5 ......................... 24
      5.2.1 Data acquisition for TR4 and TR5 ............................................... 24
      5.2.2 Calculations of Main Parameters for TR4 and TR5 ..................... 26
   5.3 Load ratio correction ................................................................................. 28
   5.4 Compressor data ........................................................................................ 29

6 Results .............................................................................................................. 31
   6.1 Cascade System CC2 ............................................................................... 31
      6.1.1 Individual units ................................................................................. 31
List of Figures

Figure 1.1: Energy use in a typical medium size Swedish supermarket [ARI05].................................................. 2
Figure 3.1: Simplified indirect system.................................................................................................................. 8
Figure 3.2: Cascade system with intermediate brine circuit.................................................................................. 9
Figure 3.3: Simplified trans-critical CO2 system with two-stage compression and inter-cooling for the low
temperature unit and single stage compression for the medium temperature unit........................................ 10
Figure 4.1: Freezer unit KS4 in system CC2......................................................................................................... 11
Figure 4.2: Full system schematic for CC2 including all components and measurement points.......................... 13
Figure 4.3: Chiller and freezer compressors and electrical panels for system TR4.............................................. 14
Figure 4.4: Freezer cycle (left) and chiller cycle (right) with compressors and electrical panels for system TR5... 15
Figure 4.5: System schematic for TR4 and TR5 with important components and measurement points............. 16
Figure 4.6: Simplified P-h diagram for the booster-type systems TR4 and TR5 during trans-critical operation.......... 18
Figure 5.1: Pressure-enthalpy diagram for a system with a liquid suction heat exchanger............................... 22
Figure 5.2: Total compressor efficiencies as functions of pressure ratios based on manufacturer data.............. 30
Figure 5.3: Compressor volumetric efficiencies as functions of pressure ratios based on manufacturer data.... 30
Figure 6.1: Monthly averages of outdoor temperature and compressor power consumption for the units of CC2... 31
Figure 6.2: Monthly averages of outdoor temperature and cooling capacity for the different units of CC2........ 32
Figure 6.3: Monthly averages of COP (excluding parasites) and outdoor temperature for the CC2 units. The
evaporation and condensing temperatures are shown as: \([t_{\text{evap}} (t_{\text{cond}})]\)........................................... 33
Figure 6.4: Ratios of cooling capacity that VKA1, VKA2 and VKA3 each supply to the medium temperature
 cabinets based on monthly averages............................................................................................................. 34
Figure 6.5: Ratios of cooling capacity used for low and medium temperature cabinets for VKA3 and for the
total system CC2 based on monthly averages.................................................................................................. 35
Figure 6.6: Power consumption of VKA3 for LT and MT use based on monthly averages................................. 35
Figure 6.7: Monthly averages of LT, MT and total cooling capacities and power consumption for CC2............ 36
Figure 6.8: LT, MT, total and load ratio corrected total COP (LR=3) for CC2, with and without parasites, based on
monthly averages. The condensing temperature is taken as the average for VKA1 and VKA3......................... 37
Figure 6.9: Monthly averages of brine supply- and return temperatures for MT cabinets in CC2......................... 37
Figure 6.10: Monthly averages of ambient and condensing temperatures, LT, MT and total power
consumption and cooling capacity, for TR4........................................................................................................ 39
Figure 6.11: Monthly averages of LT, MT and total COP (excluding parasites), ambient temperature,
evaporation (in brackets) and condensing temperatures and total COP corrected for a load ratio of 3 for TR4... 40
Figure 6.12: Monthly averages of LT, MT and total COP (including parasites), ambient temperature,
evaporation (in brackets) and condensing temperatures and total COP corrected for a load ratio of 3 for TR4.... 40
Figure 6.13: Monthly averages of LT, MT and total power consumption and cooling capacity, ambient and
condensing temperature for TR5..................................................................................................................... 41
Figure 6.14: Monthly averages of LT, MT and total COP (excluding parasites), ambient temperature,
evaporation (in brackets) and condensing temperatures and total COP corrected for a load ratio of 3 for TR5.... 42
Figure 6.15: Monthly averages of LT, MT and total COP (including parasites), ambient temperature,
evaporation (in brackets) and condensing temperatures and total COP corrected for a load ratio of 3 for TR5.... 42
Figure 6.16: Condensation temperatures for TR4, TR5 and the units in CC2 based on monthly averages............ 43
Figure 6.17: Evaporation temperatures for TR4, TR5 and the units in CC2 based on monthly averages............. 44
Figure 6.18: Monthly averages of LT cooling capacity and power consumption for CC2, TR4 and TR5......... 45
Figure 6.19: Monthly averages of MT cooling capacity and power consumption for CC2, TR4 and TR5......... 46
Figure 6.20: Monthly averages of total cooling capacity and power consumption for CC2, TR4 and TR5....... 46
Figure 6.21: Monthly averages of LT-, MT- and total COP for CC2, TR4 and TR5.............................................. 47
Figure 6.22: Monthly averages of LT-, MT- and total COP as functions of condensation temperature for TR4,
TR5 and for the VKA-units in CC2. The evaporation temperatures (°C) are also shown for each point.............. 48
Figure 6.23: Condensation temperature as a function of ambient temperature for CC2, TR4 and TR5 based on
monthly averages............................................................................................................................................. 49
Figure 6.24: Heat rejected in the dry- and gas coolers for CC2, TR4 and TR5 and the gas cooler fan operation
for TR4 and TR5 in percent of full capacity, based on monthly averages..................................................... 50
Figure 6.25: Amount of sub-cooling in condensers (only sub-critical operation) and in gas bypass line heat
exchangers for TR4 & TR5 based on monthly averages.................................................................................... 50
Figure 6.26: Monthly average defrosting time for LT and MT cabinets divided by the number of days that data
is available for each month................................................................................................................................ 51
Figure 6.27: Internal superheat for LT cabinets of CC2, TR4 and TR5, internal superheat for MT cabinets of
TR4 and TR5 and brine temperature difference over MT cabinets for CC2 based on monthly averages........... 52
## List of Tables

Table 3-1: Global warming potential (GWP), ozone depletion potential (ODP) and atmospheric life time for natural and synthetic refrigerants - [BEL02]. ................................................................. 6  
Table 5-1: Complete list of measured parameters for cascade system CC2......................................................... 19  
Table 5-2: All parameters collected with LDS for system TR4 and TR5 and reference to Figure 4.5............... 25  
Table 6-1: Temperature samples in the MT cabinets of CC2 for three months................................................. 38  
Table 6-2: Distribution of energy consumption for June to September for CC2, TR4 and TR5............... 63
## Nomenclature

<table>
<thead>
<tr>
<th><strong>Roman</strong></th>
<th><strong>Definition</strong></th>
</tr>
</thead>
<tbody>
<tr>
<td>CC</td>
<td>Cascade system</td>
</tr>
<tr>
<td>CFC:s</td>
<td>Chlorofluorocarbons</td>
</tr>
<tr>
<td>CO2</td>
<td>Carbon dioxide</td>
</tr>
<tr>
<td>COP</td>
<td>Coefficient of performance</td>
</tr>
<tr>
<td>DC</td>
<td>Dry cooler</td>
</tr>
<tr>
<td>DX</td>
<td>Direct expansion</td>
</tr>
<tr>
<td>(\dot{\varepsilon})</td>
<td>Power consumption ([\text{kW}])</td>
</tr>
<tr>
<td>Etha</td>
<td>Efficiency</td>
</tr>
<tr>
<td>Excl.</td>
<td>Excluding</td>
</tr>
<tr>
<td>(f)</td>
<td>Frequency ([\text{Hz}])</td>
</tr>
<tr>
<td>GWP</td>
<td>Global Warming Potential</td>
</tr>
<tr>
<td>(h)</td>
<td>Enthalpy ([\text{kJ/kg}])</td>
</tr>
<tr>
<td>HCFC:s</td>
<td>Hydrochlorofluorocarbons</td>
</tr>
<tr>
<td>HR</td>
<td>Heat recovery</td>
</tr>
<tr>
<td>IHE</td>
<td>Internal heat exchanger</td>
</tr>
<tr>
<td>Incl.</td>
<td>Including</td>
</tr>
<tr>
<td>ISH</td>
<td>Internal superheat</td>
</tr>
<tr>
<td>K</td>
<td>Kelvin</td>
</tr>
<tr>
<td>KB1</td>
<td>Brine loop in system (\text{CC2}).</td>
</tr>
<tr>
<td>KS4</td>
<td>Low temperature unit in system (\text{CC2})</td>
</tr>
<tr>
<td>kW</td>
<td>Kilowatt</td>
</tr>
<tr>
<td>LDS</td>
<td>Long Distance Service</td>
</tr>
<tr>
<td>LR</td>
<td>Load ratio</td>
</tr>
<tr>
<td>LR-corr.</td>
<td>Correction of load ratio to a fixed value</td>
</tr>
<tr>
<td>LT</td>
<td>Low temperature</td>
</tr>
<tr>
<td>Symbol</td>
<td>Description</td>
</tr>
<tr>
<td>--------</td>
<td>-------------</td>
</tr>
<tr>
<td>MT</td>
<td>Medium temperature</td>
</tr>
<tr>
<td>( \dot{m} )</td>
<td>Mass flow rate [kg/s]</td>
</tr>
<tr>
<td>n</td>
<td>Rotational speed (rpm)</td>
</tr>
<tr>
<td>No.</td>
<td>Number</td>
</tr>
<tr>
<td>ODP</td>
<td>Ozone Depleting Potential</td>
</tr>
<tr>
<td>P</td>
<td>Pump</td>
</tr>
<tr>
<td>p</td>
<td>Pressure [bar]</td>
</tr>
<tr>
<td>p-h diagram</td>
<td>Pressure-enthalpy diagram</td>
</tr>
<tr>
<td>( \dot{Q}_1 )</td>
<td>Heat rejected in the condenser per unit time [kW]</td>
</tr>
<tr>
<td>( \dot{Q}_2 )</td>
<td>Cooling capacity [kW]</td>
</tr>
<tr>
<td>R404A &amp; R410A</td>
<td>Refrigerants that are mixtures of other refrigerants</td>
</tr>
<tr>
<td>SC</td>
<td>Sub cooling [K]</td>
</tr>
<tr>
<td>SH</td>
<td>Superheat [K]</td>
</tr>
<tr>
<td>TR</td>
<td>Trans-critical refrigeration system</td>
</tr>
<tr>
<td>T, t</td>
<td>Temperature [°C]</td>
</tr>
<tr>
<td>V</td>
<td>Volume [m³]</td>
</tr>
<tr>
<td>( \dot{V} )</td>
<td>Volumetric flow rate [m³/s]</td>
</tr>
<tr>
<td>VB1</td>
<td>Coolant loop in system CC2.</td>
</tr>
<tr>
<td>VKA1-3</td>
<td>Medium temperature units 1-3 in system CC2</td>
</tr>
</tbody>
</table>

**Greek**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \Delta )</td>
<td>Difference</td>
</tr>
<tr>
<td>( \rho )</td>
<td>Density [kg/m³]</td>
</tr>
<tr>
<td>( \eta_{is} )</td>
<td>Isentropic efficiency</td>
</tr>
<tr>
<td>( \eta_{v} )</td>
<td>Volumetric efficiency</td>
</tr>
<tr>
<td>( \eta_{tot} )</td>
<td>Total efficiency</td>
</tr>
<tr>
<td>( v )</td>
<td>Specific volume [m³/kg]</td>
</tr>
<tr>
<td>Subscript</td>
<td>Description</td>
</tr>
<tr>
<td>-----------</td>
<td>-------------</td>
</tr>
<tr>
<td>amb</td>
<td>Ambient</td>
</tr>
<tr>
<td>calc</td>
<td>Calculated</td>
</tr>
<tr>
<td>cond</td>
<td>Condenser</td>
</tr>
<tr>
<td>comp</td>
<td>Compressor</td>
</tr>
<tr>
<td>CO2</td>
<td>CO2 side of for example a heat exchanger</td>
</tr>
<tr>
<td>defr</td>
<td>Defrosting</td>
</tr>
<tr>
<td>disks</td>
<td>Cabinets</td>
</tr>
<tr>
<td>DC</td>
<td>Dry cooler</td>
</tr>
<tr>
<td>dt</td>
<td>Temperature difference</td>
</tr>
<tr>
<td>evap</td>
<td>Evaporator</td>
</tr>
<tr>
<td>extr</td>
<td>Extrapolation</td>
</tr>
<tr>
<td>fans</td>
<td>Parameter is related to fans</td>
</tr>
<tr>
<td>GC</td>
<td>Gas cooler</td>
</tr>
<tr>
<td>high</td>
<td>High stage</td>
</tr>
<tr>
<td>hex</td>
<td>Heat exchanger</td>
</tr>
<tr>
<td>in</td>
<td>Inlet</td>
</tr>
<tr>
<td>is</td>
<td>Isentropic</td>
</tr>
<tr>
<td>int</td>
<td>Internal</td>
</tr>
<tr>
<td>liq</td>
<td>Liquid</td>
</tr>
<tr>
<td>LR-corr</td>
<td>Correction of load ratio to a set value</td>
</tr>
<tr>
<td>meas</td>
<td>Measured</td>
</tr>
<tr>
<td>min-max</td>
<td>Minimum – maximum</td>
</tr>
<tr>
<td>nom</td>
<td>Nominal</td>
</tr>
<tr>
<td>out</td>
<td>Outlet</td>
</tr>
<tr>
<td>rec</td>
<td>Receiver</td>
</tr>
<tr>
<td>refr</td>
<td>Refrigerant</td>
</tr>
<tr>
<td>s</td>
<td>Volumetric</td>
</tr>
<tr>
<td>Abbreviation</td>
<td>Full Form</td>
</tr>
<tr>
<td>-------------</td>
<td>-------------</td>
</tr>
<tr>
<td>SH</td>
<td>Superheat</td>
</tr>
<tr>
<td>subcr</td>
<td>Subcritical</td>
</tr>
<tr>
<td>transcr</td>
<td>Trans-critical</td>
</tr>
<tr>
<td>tot</td>
<td>total</td>
</tr>
<tr>
<td>vap</td>
<td>vapor</td>
</tr>
</tbody>
</table>
1 Introduction

1.1 Background

The concept of using carbon dioxide as a refrigerant dates back as far as the end of the 19th century. It was one of the first refrigerants to be used in compression type refrigeration systems and soon became widely used in shipping of frozen food products [CAV07]. Prior to 1930, the most commonly used refrigerants were all natural substances like ammonia, sulphur dioxide and carbon dioxide or hydrocarbons such as propane and ethane. However, these refrigerants presented major problems for system design and safety due to their toxicity, flammability and high pressures.

In the early 1930’s, natural refrigerants started to be replaced by synthetic substances like chlorofluorocarbons (CFC’s); R11, R12 and R13, and hydrochlorofluorocarbons (HCFC’s); R22 and R502. These were regarded as superior in terms of safety and performance and soon the use of natural refrigerants had decreased to a few specific niches, for example industrial refrigeration with ammonia [BEL02].

During the 1970’s, the discovery that the use of CFC’s and HCFC’s had an adverse effect on the ozone layer resulted in the Montreal Protocol in 1987 with several later additions. It is an international agreement about reducing and banning the use of CFC’s and HCFC’s and to replace these substances, new chlorine-free refrigerants, the HFC’s were developed [GRA05]. The HFC’s have no ozone-depletion potential (ODP) and dominate today’s market. However, the global warming potential (GWP) of some of these refrigerants is relatively high but still much lower than that of the CFC’s and HCFC’s [JOH09]. The challenge now lies in finding safe, suitable alternatives for refrigerants with high GWP in order to reduce the effects on global warming.

The two main reasons why CO2 was phased out in the first place was the low critical temperature of 31 °C and the high operating pressures [BEL02]. With the technology available at the time, CO2 refrigeration was difficult to manage safely and efficiently but today, with specially designed components available, this is no longer an issue. Carbon dioxide is the only natural refrigerant that is non-flammable, non-toxic and can operate as a free fluid in a vapor compression cycle below 0 °C and this makes it the best option for supermarket refrigeration [JOH09].

1.2 Energy use and refrigerant emissions

To keep food products cold or frozen is essential in today’s society and will become even more important in the future as urban populations grow. This is of course associated with increased energy consumption and refrigerant emissions which both have a negative impact on the environment.

In Sweden, about 3 % of the total consumption of electricity derives from energy use in supermarkets [NID09]. A typical Swedish supermarket uses about 35-50 percent of its total energy consumption for refrigeration purposes. This can be seen in figure 1.1 where the breakdown of energy use is shown for a typical medium size Swedish supermarket. The
single largest consumer of energy is the refrigeration process at 47 percent followed by lighting at 27 percent and the ventilation at 13 percent, including fans and climate control [ARI05].

![Energy Usage in A Supermarket in Sweden](image)

**Figure 1.1:** Energy use in a typical medium size Swedish supermarket [ARI05].

There are a number of ways to increase the energy efficiency of a supermarket. For example, the design of the display cabinets has a big influence on the energy consumption. Large open cabinets are ideal for displaying the food items to the customers in an attractive way but at the same time, such cabinets have a large infiltration of warm air from the store area, reducing the efficiency of the system [JOH09]. The required cooling capacity also differs between low and medium temperature display cases, or cabinets. Vertical, open display cases are normally the worst in terms of energy efficiency as they usually contain a large amount of food products on a small surface and have large open front areas. In a typical vertical display cabinet, as much as 60-70 percent of the cooling load can be caused by infiltration [ARI05].

The losses due to infiltration can be reduced by having glass doors or lids on the display cabinets, keeping the cold and warm air separated. It is also beneficial to have the food products properly stored in the cabinets to avoid disturbing the circulation of cold air. The climate outdoors and in the store also affects the cooling load. During the summer, the ambient temperature is high, resulting in high relative air humidity in the store area [JOH09]. High humidity increases the cooling load of the cabinets because of increased condensation of water and frost forming on the evaporator, cabinets and products [LIK07]. In winter, the ambient temperature drops, resulting in lower outdoor and indoor relative humidity, reducing the cooling load. For standard indoor conditions of 22 °C and 65 % relative humidity, the effect of outdoor temperature can reduce the cooling load with as much as 50-70% [NID09].

Typical energy saving technologies in supermarket refrigeration includes heat recovery, floating head condensing pressure, energy efficient lighting, defrost control, efficient motors.
and energy efficient display cabinets [ARI05]. Other than energy consumption, one of the main considerations when discussing environmental impact refrigeration systems is the refrigerant emissions.

Commercial refrigeration (supermarkets, larger kitchens, shops etc.) and cold storage account for about 28 percent of the world’s total refrigerant consumption, only surpassed by mobile air conditioning at 31 percent [SAW08]. At the same time, the commercial sector is also a strong contributor to refrigerant emissions worldwide. In a study of 220 Norwegian supermarkets, a leakage rate of 14 % was found and in an earlier study from 1993, the annual leakage was as much as 30 percent of the total refrigerant charge [GIRO4]. This shows that there is much room for improvement when it comes to refrigerant containment. Nevertheless, emissions have been decreasing due to actions taken by the industry and governmental regulations regarding containment, recovery, improved service procedures, increased personal training and usage record keeping [FRE09].
2 Objectives

The main objective of this master's thesis is to analyze and evaluate the refrigeration systems of three different supermarkets that use CO2 as a refrigerant. The analysis is based on field measurements that are collected by using two different computer programs and calculations made based on these data. The evaluation consists of calculating the cooling capacity and coefficient of performance and by making comparisons between the three systems. Heat recovery and the influence of parasites are also included in the evaluation.

2.1 Specific Objectives

The specific objectives for the thesis are:

- Collecting data for a period of four months
- Creating calculation templates
- Data processing (filtering etc.)
- Calculating main parameters (cooling capacity and COP)
- Investigating heat recovery operation
- Investigating the influence of parasites
- Comparing the different systems

2.2 Project

The thesis work presented in this report is only one part of a more comprehensive study of Swedish supermarkets which aims at evaluating the application of CO2 technology in Sweden. This project was initiated by Sveriges Energi & Kylcentrum (SEK) and Inställtörenas Utbildningscentrum (IUC) in Katrineholm, in cooperation with KTH. Several previous thesis works have been linked to this project and supermarkets with transcritical CO2 or cascade solutions have been evaluated. For comparison, conventional systems with R404A have also been investigated.

Project partners:

<table>
<thead>
<tr>
<th>Organization</th>
<th>Participant/s</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sveriges Energi &amp; Kylcentrum</td>
<td>Jörgen Rogstam</td>
</tr>
<tr>
<td>KTH-Energiteknik</td>
<td>Björn Palm/Samer Sawalha</td>
</tr>
<tr>
<td>ICA</td>
<td>Per-Erik Jansson</td>
</tr>
<tr>
<td>ICA</td>
<td>Micael Antonsson</td>
</tr>
<tr>
<td>Partor</td>
<td>Martin Johanson</td>
</tr>
<tr>
<td>WICA</td>
<td>Peter Rylander</td>
</tr>
<tr>
<td>Ahlsell</td>
<td>Torbjörn Larsson</td>
</tr>
<tr>
<td>Institution</td>
<td>Name</td>
</tr>
<tr>
<td>-------------------</td>
<td>-----------------------------</td>
</tr>
<tr>
<td>Huurre</td>
<td>Fredrik Strengbohm</td>
</tr>
<tr>
<td>Carrier</td>
<td>Jakob Granström/Björn Staf</td>
</tr>
<tr>
<td>AGA</td>
<td>Christer Hens</td>
</tr>
<tr>
<td>Tranter</td>
<td>Ulf Vestergren</td>
</tr>
<tr>
<td>Cupori</td>
<td>David Sharp</td>
</tr>
<tr>
<td>Woodley</td>
<td>Freddy Stendahl</td>
</tr>
<tr>
<td>IWMAC</td>
<td>Conny Andersson</td>
</tr>
<tr>
<td>Oppunda Svets</td>
<td>Ken Johansson</td>
</tr>
<tr>
<td>Energimyndigheten</td>
<td>Conny Ryytty</td>
</tr>
</tbody>
</table>
3 CO2 Refrigeration

Following the implementation of the Montreal and Kyoto protocols and the phase out of synthetic refrigerants like CFC’s and HCFC’s, there has been a renewed interest in natural refrigerants like CO2 for use in supermarkets. Different technologies for achieving energy efficient refrigeration exist today and CO2 systems can be constructed in many different ways. This chapter is a review of the most common system configurations for CO2 and a description of the properties of CO2.

3.1 Properties of CO2

HFC’s do not have the harmful chlorine molecule responsible for the breakdown of the ozone layer and which could be found in CFC’s and HCFC’s. Therefore, the ozone depletion potential of these refrigerants is equal to zero but their global warming potential can still be very high. The global warming potential is measured in carbon dioxide equivalents, which means that the GWP of CO2 is equal to 1. This is very small compared to many other refrigerants, for example R404A which has a GWP value of 3800. At the same time, CO2 has no ozone depletion potential, making it one of the most environmentally friendly refrigerants [FRE09], when it comes to the direct impact on environment. The GWP and ODP of different refrigerants as well as their atmospheric life time are shown in table 3-1.

![Image](https://example.com/image.png)

<table>
<thead>
<tr>
<th>Refrigerant</th>
<th>GWP (100 years, relative to CO2)</th>
<th>ODP (relative to R11)</th>
<th>Atmospheric life (years)</th>
</tr>
</thead>
<tbody>
<tr>
<td>R11</td>
<td>3,800</td>
<td>1</td>
<td>50</td>
</tr>
<tr>
<td>R12</td>
<td>8,100</td>
<td>1</td>
<td>102</td>
</tr>
<tr>
<td>R13b1</td>
<td>5,400</td>
<td>10</td>
<td>65</td>
</tr>
<tr>
<td>R22</td>
<td>1,500</td>
<td>0.055</td>
<td>13.3</td>
</tr>
<tr>
<td>R23</td>
<td>12,000</td>
<td>0</td>
<td>260</td>
</tr>
<tr>
<td>R134a</td>
<td>1,300</td>
<td>0</td>
<td>13.6</td>
</tr>
<tr>
<td>R717</td>
<td>0</td>
<td>0</td>
<td>-</td>
</tr>
<tr>
<td>- Ammonia</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>R744</td>
<td>1</td>
<td>0</td>
<td>-</td>
</tr>
<tr>
<td>- CO2</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>R290</td>
<td>3</td>
<td>0</td>
<td>-</td>
</tr>
<tr>
<td>- Propane</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

The critical temperature for CO2 is very low, only 31 °C. This means that if the ambient temperature is close to or above this value, the refrigeration cycle would have to reject heat
in trans-critical operation while the evaporation stage remains sub-critical. Generally, Trans-
critical operation requires a high compressor discharge pressure resulting in an increase of
compressor power consumption and reduced compressor performance [Ooi07]. For CO₂,
the compressor pressure ratio is usually lower than for other refrigerants, resulting in higher
volumetric efficiency. However, operation close to the critical point requires high condensing
temperatures for CO₂ and this result in low values of COP compared to other refrigerants. If
a floating head pressure (floating condensing) is used to match the variation in ambient
temperature, the COP will increase and can even exceed the COP of systems using other
refrigerants with floating head pressure [SAW07].

CO₂ has very good safety characteristics compared to many other refrigerants since it is
non-toxic, non-flammable and non-explosive. It is compatible with most materials and oils
used in refrigeration technology and it also has a high availability which makes it a very
cheap refrigerant. The thermal properties of CO₂ are excellent. High vapor density, low
surface tension and low vapor viscosity results in a heat transfer coefficient that is 2-3 times
greater than those of conventional refrigerants at the same saturation temperature since the
pressure drop in the two-phase flow is much smaller [Kar06].

CO₂ is suitable for use as a secondary fluid for low temperature applications. Compared to
other secondary fluids like potassium based salts, CO₂ has lower pumping power, smaller
pipe dimensions, good material compatibility and very good heat transfer properties [Hin09].
The low pressure drop and low volume flow rate reduce the power requirement of the
circulation pump which gives CO₂ a great advantage over different types of brines in an
indirect circuit [Lik07]. High operating pressures in CO₂ systems yields high vapor density,
which in turn results in high volumetric refrigeration capacity. This means that components
like pipes and compressors can be made smaller, leading to more compact systems and that
less amount of material is required in manufacturing [Joh09].

3.2 Design options

Carbon dioxide can be utilized for refrigeration in several different ways depending on how
the system is constructed. It can either be used as a secondary refrigerant, where it is not
compressed but circulated by pumps, or as the primary refrigerant with compression. For
CO₂ refrigeration, there are three different main system solutions used in supermarkets;
indirect system, cascade DX system or trans-critical DX system [Tam09]. When the interest
in CO₂ as a refrigerant was renewed in the end of the 20th century, the first commercial
application was as a secondary working fluid in indirect systems. The successful operation of
these early systems soon resulted in a wider use of CO₂ and today, there are more than 400
installations in Europe using direct expansion for LT and MT applications with two stage
compression for the LT side [KAU10]

3.2.1 Indirect System

In an indirect system, the refrigerating effect is obtained by using two different fluids, a
primary refrigerant in a conventional vapor compression cycle, and a secondary refrigerant
that is usually brine. The secondary fluid is not compressed but circulated from the machine
room to the display cabinets in the store area and back by using pumps. The CO₂ would
typically be used as the secondary fluid while the primary refrigerant could be for example
ammonia, R404A or propane. Using CO₂ in this manner suits well for public areas like a
supermarket as it is non-flammable and rather non-toxic [JOH09]. The required pumping power is also very small when comparing with other refrigerants in conventional brine circuits. The main reason for this is the fact that the CO2 changes phase on the CO2 side which results in a small volumetric flow rate and subsequently, a small pressure drop [KAR06]. A simplified schematic of an indirect system is shown in figure 3.1.

![Figure 3.1: Simplified indirect system.](image)

The secondary refrigerant evaporates by absorbing heat from low and medium temperature display cabinets and cold storages. It is pumped back to the machine room where it is heat exchanged with the primary refrigerant causing it to evaporate. The operating temperature of CO2 can be between -40 and -10 °C [TAM09]. The primary refrigerant is compressed and enters the condenser where heat is transferred to a coolant circuit and the heat is finally rejected to the ambient in the dry coolers. The primary refrigerant loop can be contained in the machine room which means that toxicity of the refrigerant is not an issue. The small pressure drop for CO2 in the secondary circuit means that pipe dimensions and other components can be made smaller which favours the design of compact systems [SAW08].

### 3.2.2 Cascade DX System

For evaporation temperatures below 30 °C, a cascade system is usually the best option [FRE09]. This type of system has two refrigerants each operating in a separate cycle. Since the total required temperature lift has been divided over two units, higher efficiencies can be achieved than for one stage compression systems. However, a disadvantage with this type of system is the extra temperature difference due to the heat exchanger between the stages [GRA05]. This is especially true for the system shown in figure 3.2, where an additional brine circuit has been introduced between low and medium temperature stages. The CO2 circuit operates in the low stage with direct expansion, rejecting heat to the brine in the condenser. The brine supports the medium temperature cabinets and is circulated by a pump. Heat is transferred to the high stage refrigerant and released to the ambient via a coolant loop.
3.2.3 Trans-critical DX System

What separates trans-critical systems from other systems is the ability to reject heat at a state above the critical point. At pressures higher than the critical point, there is no saturated condition and temperature is independent of pressure [KAR06]. This means that both temperature and pressure have a separate influence on cooling capacity and COP. In the trans-critical region, COP is a function of the gas cooler outlet temperature and the discharge pressure. This means, that for each gas cooler outlet temperature, there is an optimum value of high stage discharge pressure [LIK07]. Since there is no condensation but single phase heat exchange above the critical point, the heat rejection process is referred to as “gas cooling” for trans-critical operation. Figure 3.3 shows a simplified schematic of a trans-critical CO2 system. The low and high stages are separated and both connected to a coolant loop for the heat rejection. Ideally, direct heat rejection in gas coolers should be used instead of a coolant loop in order to keep the condensing temperature as low as possible. The medium temperature unit has single-stage compression and the low temperature unit has two-stage compression with inter-cooling.

The ambient conditions affect the operation of this type of system. When the ambient temperature is high, the system mainly operates in a trans-critical mode, but for low ambient temperatures, the cycle is sub-critical. Because of the high compressor energy consumption that is required for trans-critical operation, trans-critical CO2 systems are best suited for cold climates.
Figure 3.3: Simplified trans-critical CO2 system with two-stage compression and inter-cooling for the low temperature unit and single stage compression for the medium temperature unit.

One of the main advantages with using CO2 as the only refrigerant in the cycle is that it eliminates the temperature differences that are present in the heat exchangers of indirect and cascade systems [JOH09]. The use of CO2 as the only refrigerant is also beneficial for the environment when considering the direct impact due to refrigerant release.
4 System descriptions

This chapter describes the structure and function of the different refrigeration systems included in this master’s thesis. Three supermarkets are evaluated; one is a cascade solution using R404A and CO2, and two are trans-critical CO2 systems. They are all located in the southern part of Sweden and therefore, they have reasonably similar ambient conditions such as outdoor temperature. Following the principles of the larger, overall supermarket investigation, of which this thesis is one part, the systems have been named CC2, TR4 and TR5 for cascade system 2 and trans-critical systems no. 4 and 5 respectively.

4.1 Cascade system CC2

The refrigeration system CC2 has been in operation since the end of Mars 2009. With more than 50 medium temperature cabinets and 30 low temperature cabinets, it is the largest of the three supermarkets included in this thesis report. It has a design capacity of about 240 kW for the chillers and about 50 kW for the freezers. It is also the system located furthest to the north and unlike the trans-critical CO2 systems, it is situated inland and not on the coast.

4.1.1 Overall system description

CC2 is a cascade system that has three R404A-circuits in the high temperature stage and one DX CO2- circuit in the low temperature stage. The CO2-circuit supports the low temperature cabinets while the medium temperature cabinets are connected to a propylene-glycol brine circuit in the intermediate stage. An indirect system with an ethylene-glycol coolant loop is used for heat rejection and heat recovery. Figure 4.1 shows the CO2 unit KS4 that serves the freezers.

![Figure 4.1: Freezer unit KS4 in system CC2.](image-url)
This unit has five LG rotary compressors with number GP290PA that are on-off controlled. These are originally designed for R410A but used for CO2 in this case. The number of compressors in operation is available as a percentage of the full capacity and this is used as input data to the calculations in chapter 5.

Figure 4.2 shows the full system schematic for CC2 including the different components and the measurement points. In the figure, two of the R404A circuits are placed in parallel (VKA1 and VKA2) and produces cooling capacity for the medium temperature cabinets via the brine circuit. This parallel solution enables VKA2 to work as a backup-system in case VKA1 fails. VKA3 is a separate unit that is connected to the low temperature CO2 unit KS4 via the brine loop. The cooling capacity produced in this unit is thus used for both low and medium temperature applications. The VKA3 cooling capacity for LT use is equal to the heat rejected in the condenser of KS4. The heat from the VKA units is rejected to the coolant loop in the condensers and in the sub-coolers located after the condensers. The VKA-units are all equipped with liquid suction heat exchangers (IHE 1-3) which further increases the amount of sub-cooling and heats the refrigerant before the compressors. Each of the VKA-units has two compressors mounted in parallel. The compressors are Copeland ZB220CE-TWM hermetic scroll compressors [COP09] and they are all frequency controlled. This makes the system more adaptable to changes in the cooling load.

The brine circuit is equipped with three pumps that circulate the brine from the machine room to the medium temperature cabinets in the store area and back. Pump 2A and 2B are placed in parallel and pump the brine from the KS4 condenser out to the MT cabinets. P1 is the main pump that circulates the brine with a volumetric flow of about 52 m$^3$ per hour (average measured value over four months). This is the flow that goes through the evaporators of VKA1 and VKA2 which have the same evaporation temperature. One part of the flow is pumped back to store area and one part is circulated through the evaporator of VKA3 before entering the KS4 condenser and reaching the pumps 2A and 2B. The two flows from pump 1 and 2 mix before the brine is supplied to the medium temperature cabinets.

The ethylene-glycol in the coolant loop is circulated with six pumps, P3-P8. The coolant loop is constructed as a two way flow circuit, having pumps located before the dry coolers and before the heat recovery system (P3 and P4 respectively). Depending on how these pumps are regulated, the heat from the VKA-units can be utilized either in the heat recovery system or be released to the ambient. The heat that reaches the heat recovery system is used for floor and space heating applications. In this report, the heat recovery system in supermarket CC2 has not been thoroughly investigated due to a lack of measurement points at the site. The pumps P5, P6 and P7 supply the coolant to the condensers of VKA1, VKA2 and VKA3 respectively. Pump P8 supplies the coolant to the sub-coolers in all three VKA-units. VKA3 is the unit closest to the dry cooler.

Unit VKA3 is highly interesting because of its dual function; producing both LT and MT cooling capacity. It is also the unit closest to the dry cooler and its operation and characteristics will be investigated in chapter 6.5.
Figure 4.2 includes:

- Three R404A DX units for the first stage (VKA 1-3):
  - Two parallel compressors per VKA-unit: Copeland ZB220KCE-TWM,
  - Three internal heat exchangers (IHE 1-3)
  - Three sub-coolers
  - Three evaporators and three condensers

- Brine loop:
  - Brine - Propylene glycol
  - Pumps P1 & P2-A&B
  - Chillers

- One CO2 DX unit for the second stage (KS4)
  - Five parallel compressors: LG GP290PA
  - Receiver
  - Condenser
  - Freezers

- Coolant loop:
  - Coolant – Ethylene glycol
  - Pumps P3-8
  - Dry cooler

Figure 4.2: Full system schematic for CC2 including all components and measurement points.
4.2 Trans-critical systems TR4 and TR5

The two systems TR4 and TR5 are located in the southern part of Sweden, on the southwest- and southeast coast respectively. TR4 is the smallest of the three systems in the investigation with only eight island freezers and five chillers. The size of TR5 is comparable to that of the cascade system CC2 but TR5 has a few cabinets less. Both TR4 and TR5 are new installations and have only been in operation since the beginning of May 2010. The systems are quite similar in design and only differ by size. For this reason, they can be described in the same chapter.

4.2.1 Overall system description for TR4 and TR5

TR4 and TR5 are CO2 trans-critical booster systems with direct expansion in both low and medium temperature stages. Figure 4.3 shows the low- and medium temperature level compressors and electrical panels of TR4. For the low temperature stage, two subcritical compressors of the model; Bitzer 2MHC-05KB-40S, are placed in parallel. The trans-critical operation is then achieved by two Bitzer compressors of the model; 4MTH-10KI-40S, also mounted in parallel. The MT compressors are used for compressing the entire refrigerant flow of the system. The section of the store area that is used for chilled or frozen products is comparatively small for TR4 with only 5 cooling cabinets and 8 island freezers. Therefore, the design capacity for this system is only about 5.8 kW for the freezers and 24.5 kW for the chillers. Both the chillers and freezers of TR4 have no glass doors or glass lids which increases the influence of the ambient temperature on cooling capacity and energy consumption during the summer period, mainly due to increased humidity in the outdoor and indoor air.

![Figure 4.3: Chiller and freezer compressors and electrical panels for system TR4.](image)
Figure 4.4 shows the freezer and chiller cycles (left and right respectively) with the compressors and electrical panels for TR5. The setup is identical to that of TR4 but in TR5, there are four parallel Bitzer compressors of the model; 2HHC-2K in the low stage cycle. The high stage cycle has five parallel Bitzer 4HTC-20KI compressors that are used for circulating the entire refrigerant flow of the system. TR5 has glass doors on almost all chillers and glass lids on all freezers which reduces the influence of the ambient.

![Image](image_url)

**Figure 4.4:** Freezer cycle (left) and chiller cycle (right) with compressors and electrical panels for system TR5.

A simplified schematic of the refrigeration system is displayed in figure 4.5 where the important components and measurement points are visible. The two systems both have a two stage compression using parallel Bitzer compressors. In the first stage, subcritical compression increases the pressure from about 12 to 30 bars. In the second stage compression, trans-critical compressors are used in parallel to raise the pressure to the heat rejection level at about 50-75 bar. The pressure levels are very similar for both TR4 and TR5. The refrigerant enters the heat recovery system which acts as a de-superheater before the condenser. The heat recovery systems are counter flow tube-tube heat exchangers where a water-glycol circuit transports heat from the refrigerant to the ventilation air. The systems have floating condensing which reduces the required high stage discharge pressure as the condensing temperature follows the level of the ambient temperature. If the outdoor temperature drops to about 10°C, more heating of the store area is required and the condensing pressure/temperature is raised to increase the amount of heat recovery. On the liquid side, there is a small amount of sub-cooling for each system in the condensers, about 3K for TR4 and 4K for TR5, before the refrigerant flows through an expansion valve and is collected in a receiver vessel.
Figure 4.5 includes:

- Low temperature stage:
  - Two parallel Bitzer 2MHC-05 compressors for TR4
  - Four parallel Bitzer 2HHC-2K compressors for TR5
  - Low temperature cabinets (freezers)
  - Direct expansion

- Medium temperature stage:
  - Two parallel Bitzer 4MTC-7K compressors for TR4
  - Five parallel Bitzer 4HTC-20K1 compressors for TR5
  - Medium temperature cabinets (chillers)
  - Direct expansion
  - Heat recovery (de-superheater)
  - Gas cooler
  - Receiver
  - Gas bypass to MT compressor inlet

Figure 4.5: System schematic for TR4 and TR5 with important components and measurement points.
The receiver vessel has two outlets, one at the bottom for the liquid that is to be introduced to the cabinets and evaporated, and one at the top for vapour extraction to a gas bypass circuit. This vapour is first expanded in two parallel expansion valves to reduce its pressure and temperature. Second, the resulting vapour-liquid mixture in this line and the saturated liquid from the bottom of the receiver tank enter a counter flow heat exchanger where the liquid is sub-cooled and the vapour-liquid mixture is heated and returned to the suction side of the high stage compressors.

There are two main reasons for using a heat exchanger in this manner. First, when expanding the saturated vapour from the receiver to the suction side of the high stage compressors, the result will be a liquid-vapour mixture and there is a risk of liquid droplets entering the compressor. Since this mixture will be mixed with the relatively hot discharge gas from the low pressure compressors, the risk of this happening is probably small but the use of the heat exchanger reduces it further. Due to the slope of the saturated vapour line in the p-h diagram, the lower the receiver pressure is, the higher the vapour quality in the bypass line will be, which reduces the risk of liquid entering the compressors and increases the COP [DAN08]. Second, sub-cooling of the liquid from the receiver reduces the vapour quality at the inlet of the evaporators. This means that a larger region of the evaporators will be filled with liquid which improves the heat transfer. The liquid sub-cooling in the heat exchanger turned out to be very small, on average about 1K or less for the time period of this study.

After the heat exchanger, the liquid refrigerant flow is divided in two parts, one leading to the medium temperature cabinets and one to the freezers. There, evaporation takes place after the refrigerant has passed through expansion valves. The refrigerant from the freezers is returned to the low stage compressors and mixes with the flow from the chillers at the compressor discharge. It also mixes with the flow from the receiver before being compressed to a heat sink level. A p-h diagram for the entire cycle is shown in figure 4.6 including explanations.

Some features have been left out of the simplified sketch in Figure 4.5. For example, for TR4, there is a pipe leading from the receiver to the roof of the building, acting as a safety valve. Similar pipes are connected to the suction sides for the flows coming from the cooling cabinets and the freezers. There is also a pipe connecting the high stage compressor discharge with the receiver tank vapour outlet. This pipe has a relatively small diameter compared to the other parts of the system which indicates a smaller mass flow. However, this connection is not always in operation. Its purpose is rather to serve as a pressure regulator as hot gas can be introduced to the tank from the compressor discharge, thus increasing the temperature and pressure if necessary.

If instead, the high stage compressor discharge temperature is too high, there is a sensor located after the compressors that is connected to a valve controlling a pipe used for liquid injection. This pipe is situated between the liquid side outlet from the heat exchanger and the vapour side inlet. Due to the difference in pressure, liquid will be injected into the vapour-liquid mixture line if the valve is opened, thereby reducing the temperature at the second stage compressor inlet and outlet. Another part that has been excluded from the simplified system schematic is a bypass from the heat exchanger outlet on the liquid side to the pipe that leads to the cabinets but this bypass is only used for maintenance and does not affect the system in normal operation.
4.2.2 P-h diagram for TR4 and TR5

Figure 4.6 shows a simplified pressure-enthalpy diagram with explanations that also relates to measurement points in Figure 4.5. The distance between some of the parameters and the vapor-liquid saturation lines has been exaggerated to clarify how the systems operate. The figure includes:

1) Cooling of the refrigerant in the condenser [e-f]
2) Expansion of the refrigerant [f-g]
3) The refrigerant entering the receiver where liquid and vapour separation takes place [g-h] and [g-l] respectively.
4) Expansion of the vapour in two parallel expansion valves [l-m].
5) The vapour-liquid mixture and the liquid from the receiver entering the heat exchanger. The vapour-liquid mixture is heated up and the liquid is sub-cooled [m-n] and [h-j] respectively.
6) Expansion of the refrigerant before the cooling cabinets [i-j].
7) Evaporation of the refrigerant in the medium temperature cabinets [j-(m)]
8) Expansion of the refrigerant before the freezers [j-k].
9) Evaporation of the refrigerant in the freezers [k-a] including external super heat.
10) Subcritical compression of the refrigerant [a-b]
11) The refrigerant from the 1:st stage compressor discharge being mixed with the flow from the medium temperature cabinets and the vapour from the heat exchanger [b+n+MT=c].
12) The refrigerant being compressed before entering the heat recovery system [c-d].
13) The refrigerant rejecting heat to the ventilation air in the heat recovery system via a counter flow tube-tube heat exchanger with a water-glycol loop on the heat sink side [d-e].

Figure 4.6: Simplified P-h diagram for the booster-type systems TR4 and TR5 during trans-critical operation.
5 Methods of Evaluation

The main parameters in the evaluation of the different systems are cooling capacity and coefficient of performance. The calculations are made in Microsoft Excel based on measurements of temperature, pressures and compressor speed and using two different programs for the data collection, LDS [LDS10] and IWMAC [IWM10]. Enthalpies are calculated from temperatures and pressures by using the NIST reference properties through the software Refprop 7.0 [NIS10]. The mass flow is calculated using these enthalpies, measured compressor speed and compressor manufacturer data. The cooling capacity is then calculated from mass flow and enthalpies and the energy consumption is calculated using mass flow, enthalpies and total efficiency from compressor manufacturer data. In some cases, energy consumption is also obtained from direct measurements.

5.1 Cascade system CC2 calculation

This chapter treats the data acquisition and calculations for system CC2. Each measured parameter is listed and the methods used to obtain the results are explained. The main parameters; cooling capacity and coefficient of performance are calculated for the low temperature (LT) and medium temperature (MT) stages and on the basis of the total system.

5.1.1 Data acquisition for CC2

The program IWMAC is used for collecting data for system CC2. IWMAC is a web-based program used for surveillance of different types of refrigeration systems [IWM10]. Each measured value is stored in a database which can be accessed on line and data can be downloaded to text files. In case of the CC2 data acquisition, this presented some problems due to inconsistent time intervals in the collected data. The differences occurred mainly between different parameters but the time interval also varied for each individual parameter. The reason for the inconsistency could be seen in the data series as IWMAC only had registered a new measured value when that value had changed. If a measured temperature had remained constant over a certain time, no values were added to the data series until that parameter changed. Since different parameters changed with different frequency, the collected data series had different lengths and time intervals. The inconsistency complicated the calculations in excel and another software, Python [PYT10], had to be used to convert the collected data to 10 minute averages. If the time interval exceeded 10 minutes, the previous value was used. The collected data includes temperatures, energy measurements and the signals for the compressors to determine their rpm. Figure 4.2 shows the different temperature measurements in the system schematic and these are also listed in Table 5-1.

<table>
<thead>
<tr>
<th>Component</th>
<th>Parameter</th>
<th>IWMAC nomenclature</th>
<th>Reference to Fig.4.2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Compressor KS4:</td>
<td>Compressor inlet temperature</td>
<td>Suggas (Se)</td>
<td>T1</td>
</tr>
<tr>
<td></td>
<td>Compressor outlet temperature</td>
<td>Hetgas (Sd)</td>
<td>T2</td>
</tr>
</tbody>
</table>

Table 5-1: Complete list of measured parameters for cascade system CC2.
<table>
<thead>
<tr>
<th>Description</th>
<th>Symbol</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Compressor power consumption</td>
<td>E02</td>
<td></td>
</tr>
<tr>
<td>Condenser KS4:</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Condensing temperature</td>
<td>Hp (t1)</td>
<td>Tcond4</td>
</tr>
<tr>
<td>Condenser outlet temperature</td>
<td>GT32</td>
<td>T3</td>
</tr>
<tr>
<td>Evaporator KS4:</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Evaporation temperature</td>
<td>Lp (t2)</td>
<td>Tevap4</td>
</tr>
<tr>
<td>Internal superheat in LT cabinets</td>
<td>Superheat AK</td>
<td></td>
</tr>
<tr>
<td>Brine loop KB1:</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Brine inlet to MT cabinets</td>
<td>KB1 GT1</td>
<td>T4</td>
</tr>
<tr>
<td>Brine return temperature</td>
<td>KB1 GT2</td>
<td>T5</td>
</tr>
<tr>
<td>VKA evaporators 1-2:</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Evaporation temperature</td>
<td>Fa (t2)</td>
<td>Tevap1&amp;2</td>
</tr>
<tr>
<td>Compressor VKA 1-2:</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Compressor inlet temperatures</td>
<td>GT27 and GT24</td>
<td>T6-7</td>
</tr>
<tr>
<td>Compressor outlet temperatures</td>
<td>GT26 and GT23</td>
<td>T8-9</td>
</tr>
<tr>
<td>Compressor frequency</td>
<td>Frekv Hz</td>
<td></td>
</tr>
<tr>
<td>Compressor signal on/off</td>
<td>Driftind. on/off</td>
<td></td>
</tr>
<tr>
<td>Condenser VKA 1-2:</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Condensing temperatures</td>
<td>2 x Kond (t1)</td>
<td>Tcond1&amp;2</td>
</tr>
<tr>
<td>Condenser outlet temperatures</td>
<td>GT28 and GT19</td>
<td>T10-11</td>
</tr>
<tr>
<td>Sub-cooler VKA 1-2:</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Sub-cooler outlet temperatures</td>
<td>GT7 and GT8</td>
<td>T12-13</td>
</tr>
<tr>
<td>Internal heat exchangers</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Internal heat exchanger outlet temp.</td>
<td>GT29 and GT25</td>
<td>T14-15</td>
</tr>
<tr>
<td>Compressor VKA3:</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Compressor inlet temperature</td>
<td>GT22</td>
<td>T16</td>
</tr>
<tr>
<td>Compressor outlet temperature</td>
<td>GT21</td>
<td>T17</td>
</tr>
<tr>
<td>Compressor frequency</td>
<td>Frekv Hz</td>
<td></td>
</tr>
<tr>
<td>Compressor signal on/off</td>
<td>Driftind. on/off</td>
<td></td>
</tr>
<tr>
<td>Condenser VKA3:</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Condensing temperature</td>
<td>Kond (t1)</td>
<td>Tcond3</td>
</tr>
<tr>
<td>Condenser outlet temperature</td>
<td>GT20</td>
<td>T18</td>
</tr>
<tr>
<td>Sub-cooler VKA3:</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Sub-cooler outlet temperature</td>
<td>GT9</td>
<td>T19</td>
</tr>
<tr>
<td>Internal heat exchanger</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Internal heat exchanger outlet temp.</td>
<td>GT18</td>
<td>T20</td>
</tr>
<tr>
<td>Coolant loop VB1:</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Heat recovery inlet temperature</td>
<td>VB1 GT10</td>
<td>T21</td>
</tr>
<tr>
<td>Heat recovery outlet temperature</td>
<td>VB1 GT11</td>
<td>T22</td>
</tr>
<tr>
<td>Energy consumption for dry cooler</td>
<td>E02</td>
<td></td>
</tr>
<tr>
<td>Energy consumption for pump 3-8</td>
<td>E04-05, U01-04</td>
<td></td>
</tr>
<tr>
<td>Brine loop KB1:</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Condenser 4 brine inlet temperature</td>
<td>GT4</td>
<td>T23</td>
</tr>
<tr>
<td>Condenser 4 brine outlet temperature</td>
<td>GT30</td>
<td>T24</td>
</tr>
<tr>
<td>Evaporator 1-2 brine outlet temp.</td>
<td>GT5 and GT6</td>
<td>T25-26</td>
</tr>
<tr>
<td>Evaporator 3 brine inlet temperature</td>
<td>GT3</td>
<td>T27</td>
</tr>
<tr>
<td>Evaporator 3 brine outlet temperature</td>
<td>GT17</td>
<td>T28</td>
</tr>
<tr>
<td>Energy consumption for pump 1-2</td>
<td>Power P1, E03</td>
<td></td>
</tr>
<tr>
<td>Total system:</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Total energy consumption</td>
<td>E01</td>
<td></td>
</tr>
</tbody>
</table>
5.1.2 Calculation of main parameters CC2

The important temperature measurements for system CC2 are shown in the system schematic in chapter 4. Pressures are not obtained directly from IWMAC but calculated from temperatures at the saturation lines by using the program Refprop. The same software is used to calculate enthalpies, entropies and specific volume from temperatures and pressures. The cooling capacity is calculated using equation 5.1.

\[ \dot{Q}_{\text{evap}} = \dot{m}_{\text{refr}} \cdot \Delta h_{\text{evap}} \]  \hspace{1cm} (5.1)

Where \( \Delta h_{\text{evap}} \) is the enthalpy difference over the evaporator (cabinets) calculated from temperatures and pressure and \( \dot{m}_{\text{refr}} \) is the refrigerant mass flow in the evaporator. This equation is used for the VKA units and the KS4 unit. The refrigerant mass flows are calculated with equation 5.2 for the VKA-units:

\[ \dot{m}_{\text{refr}} = \frac{\dot{V}_s \cdot \eta_s}{v_{\text{comp,in}}} \]  \hspace{1cm} (5.2)

where, \( \eta_s \) is the volumetric efficiency and \( \dot{V}_s \) is the swept volume flows of the compressor in m\(^3\)/s, both are based on compressor manufacturer data [COP09]. \( v_{\text{comp,in}} \) is the specific volume at the compressor inlet in m\(^3\)/kg, calculated with Refprop from temperature and pressure. The swept volume is calculated with equation 5.3.

\[ \dot{V}_s = \frac{V_s \cdot n}{60} \]  \hspace{1cm} (5.3)

The swept volume (\( V_s \)) is the space inside the compressor where the refrigerant collects before compression and this parameter is known from the compressor manufacturer data. The frequency for the compressors is available in IWMAC for the VKA units and for KS4, which has on-off controlled compressors, the frequency (\( f \)) is known from the manufacturer data. The compressor speed (\( n \)) in revolutions per minute is calculated from frequency using equation 5.4, where \( n_{\text{nom}} \) and \( f_{\text{nom}} \) are nominal compressor speed and frequency from the manufacturer data.

\[ n = f \cdot \frac{\eta_{\text{nom}}}{f_{\text{nom}}} \]  \hspace{1cm} (5.4)

The total and volumetric efficiencies are also taken from the compressor manufacturer data and plotted as a function of pressure ratio. By using the curve-fit option in Excel, equations for these functions can be extracted and by using the real pressure ratios based on measurements, the actual efficiencies can be calculated. Equation 5.5 shows an example of a function generated from compressor data and used in the calculations of volumetric efficiency. The pressure \( p_1 \) represents discharge pressure and \( p_2 \) suction pressure.

\[ \eta_s = 0.0003 \cdot \left( \frac{p_1}{p_2} \right)^2 - 0.031 \cdot \frac{p_1}{p_2} + 0.9769 \]  \hspace{1cm} (5.5)

Figure 5.2 and 5.3 show the total and volumetric efficiencies for all compressors except for unit KS4 as functions of pressure ratios. For KS4, the energy consumption is measured directly and it is therefore considered more accurate than a calculated value. Since this accurate value is available, the mass flow for KS4 is not calculated from volumetric efficiency.
and compressor manufacturer data but from measured power consumption, enthalpy difference and by assuming 7 \% losses according to equation 5.6 [FRE09]:

\[
\dot{m}_{refr,KS4} = \frac{\dot{E}_{measured,KS4} \cdot 0.93}{\Delta h_{comp}}
\]

5.6

In Figure 4.2, temperature measurements are not available directly before and after the evaporators for the VKA circuits. In order to obtain the enthalpy difference over the evaporators, heat balances over the internal heat exchangers must be made. Figure 5.1 shows the liquid suction heat exchanger in a p-h diagram. Since the mass flow is the same on both sides of the heat exchanger, equation 5.7 is valid:

\[
h_{comp, in} - h_{evap, out} = h_{SC, out} - h_{HHE, out}
\]

5.7

This can be re-organized to form equation 5.8:

\[
h_{evap, out} - h_{HHE, out} = h_{comp, in} - h_{SC, out} = \Delta h_{evap}
\]

5.8

![Figure 5.1: Pressure-enthalpy diagram for a system with a liquid suction heat exchanger.](image)

Equation 5.1 can now be used to calculate the cooling capacities. For VKA3, the part of the cooling capacity used for low temperature applications is equal to the condensing power of unit KS4. The cooling capacities for LT and MT use are separated in equation 5.9 and 5.10.

\[
\dot{Q}_{evap,VKA3,LT} = \dot{Q}_{cond,KS4} = \dot{m}_{CO2} \cdot \Delta h_{cond,KS4}
\]

5.9

\[
\dot{Q}_{evap,VKA3,MT} = \dot{Q}_{evap,VKA3,tot} - \dot{Q}_{evap,VKA3,LT}
\]

5.10

The compressor power consumption is calculated for VKA1-3 using mass flow, isentropic enthalpy at compressor outlet, compressor inlet enthalpy and total efficiency according to equation 5.11.

\[
\dot{E}_{comp} = \dot{m}_{refr} \cdot \frac{h_{comp, out, is} - h_{comp, in}}{\eta_{tot}}
\]

5.11
For KS4, the power consumption is available as a direct measurement and this is used instead of a calculated value. The isentropic outlet enthalpy is calculated with Refprop using entropy, which is constant for isentropic expansion, and discharge pressure. Measurements of power consumption are also available for all the pumps and for the dry cooler fans. The compressor COP is calculated for the VKA units and for unit KS4 using equation 5.12:

\[
COP_{\text{comp}} = \frac{\dot{Q}_{\text{evap}}}{\dot{E}_{\text{comp}}}
\]  

The power consumption for the cooling of the MT cabinets is calculated with equation 5.13 where \(COP_{\text{comp,vka}}\) is the compressor COP calculated with equation 5.12. The power consumption for LT use is calculated in equation 5.14 and the cooling capacities are calculated with equation 5.15 and 5.16.

\[
\hat{E}_{\text{vka3,MT}} = \frac{\dot{Q}_{\text{evap vka3,MT}}}{COP_{\text{comp,vka}}}
\]

\[
\hat{E}_{\text{vka3,LT}} = \hat{E}_{\text{vka3,tot}} - \hat{E}_{\text{vka3,MT}}
\]

\[
\dot{Q}_{\text{evap,MT,tot}} = \dot{Q}_{\text{evap vka1}} + \dot{Q}_{\text{evap vka2}} + \dot{Q}_{\text{evap vka3,MT}}
\]

\[
\dot{Q}_{\text{evap,LT,tot}} = \dot{Q}_{\text{evap,KS4}}
\]

The total compressor power consumption for the MT-stage is the sum of the calculated MT compressor power for each of the VKA units as can be seen in equation 5.17. The total compressor power consumption for the LT-stage is the sum of the measured power consumption for KS4 and the part of the VKA3 compressor power used for LT applications as shown in equation 5.18.

\[
\hat{E}_{\text{MT,tot,comp}} = \hat{E}_{\text{vka1}} + \hat{E}_{\text{vka2}} + \hat{E}_{\text{vka3,MT}}
\]

\[
\hat{E}_{\text{LT,tot,comp}} = \hat{E}_{\text{ks4}} + \hat{E}_{\text{vka3,LT}}
\]

The compressor coefficient of performance is calculated for the LT- and MT stage and for the total system according to equation 5.19, 5.20 and 5.21.

\[
COP_{\text{comp,LT}} = \frac{\dot{Q}_{\text{evap,LT,tot}}}{\hat{E}_{\text{LT,tot,comp}}}
\]

\[
COP_{\text{comp,MT}} = \frac{\dot{Q}_{\text{evap,MT,tot}}}{\hat{E}_{\text{MT,tot,comp}}}
\]

\[
COP_{\text{comp,tot}} = \frac{\dot{Q}_{\text{evap,LT,tot}} + \dot{Q}_{\text{evap,MT,tot}}}{\hat{E}_{\text{LT,tot,comp}} + \hat{E}_{\text{MT,tot,comp}}}
\]

The total power consumption (including parasites) is calculated for the medium temperature stage by adding the total MT compressor power and the measured power for the dry cooler
fans and for the pumps P1, P3, P5, P6 and P7 as shown in equation 5.22. The total power consumption for the LT stage is calculated according to equation 5.23.

$$\dot{E}_{MT,tot} = \dot{E}_{MT,tot,comp} + \dot{E}_{P1} + \dot{E}_{P3} + \dot{E}_{P5} + \dot{E}_{P6} + \dot{E}_{Fans}$$  \hspace{1cm} 5.22

$$\dot{E}_{LT,tot} = \dot{E}_{LT,tot,comp} + \dot{E}_{P2}$$  \hspace{1cm} 5.23

The coefficient of performance is calculated for LT- and MT stage and for the total system according to equation 5.24, 5.25 and 5.26.

$$COP_{LT} = \frac{\dot{Q}_{evap,LT,tot}}{\dot{E}_{LT,tot}}$$  \hspace{1cm} 5.24

$$COP_{MT} = \frac{\dot{Q}_{evap,MT,tot}}{\dot{E}_{MT,tot}}$$  \hspace{1cm} 5.25

$$COP_{tot} = \frac{\dot{Q}_{evap,LT,tot} + \dot{Q}_{evap,MT,tot}}{\dot{E}_{LT,tot} + \dot{E}_{MT,tot}}$$  \hspace{1cm} 5.26

5.2 Calculations for trans-critical systems TR4 and TR5

This chapter contains descriptions of the data acquisition and calculations for the two trans-critical CO2 systems TR4 and TR5. As mentioned in the system descriptions, the two systems are very similar in their design and only differ by size. The measurement points are therefore the same for the two systems as seen in Figure 4.5, even though the number of measured parameters differs. The calculations of cooling capacity and coefficient of performance are made for the LT- and MT stages and for the total systems.

5.2.1 Data acquisition for TR4 and TR5

The program Long Distance Service, LDS [LDS10], is used for the data acquisition for TR4 and TR5. In both locations, a number of parameters are being measured and the data is saved on an on-site computer. Data can then be access from any computer by connecting to the logging computers via a modem, specifying which parameters to collect. In LDS, it is possible to write scripts, determining which data that should be collected, for which period of time and when the collection should take place. This way, the data acquisition is made automatically on a regular basis. One problem with this method of data collection is that the on-site logging computers have a limited capacity to store data. If the data is downloaded to another computer via a modem once every 24 hours, the data is available with a minimum time interval of two minutes. If the download is made less frequent, for example once a week, the logging computers will have started to delete data in order to save space on the hard drives. The time interval of the data will not change but there will be gaps of several hours in the data series. This has resulted in major problems with the data collection for this study. If the automatic dialling via modem or the data logging on the sites should fail, data is lost and in some cases, it cannot be recovered. An example of this can be seen in Figure 6.34 where gaps in the collected data are visible as straight lines. Due to problems with the on-site computers, data for TR5 is only available for the first half of September. Table 5-2 shows the parameters collected in LDS and the respective nomenclature in Figure 4.5.
Table 5-2: All parameters collected with LDS for system TR4 and TR5 and reference to Figure 4.5.

<table>
<thead>
<tr>
<th>Component</th>
<th>Parameter</th>
<th>LDS Nomenclature</th>
<th>Reference to Fig.4.5</th>
</tr>
</thead>
<tbody>
<tr>
<td>LT Compressors:</td>
<td>Compressor inlet temperature</td>
<td>Suction gas temp.</td>
<td>T_a</td>
</tr>
<tr>
<td></td>
<td>LT Suction gas pressure (p_low)</td>
<td>(LT): po Act</td>
<td>P_low</td>
</tr>
<tr>
<td></td>
<td>Compressor outlet temperature</td>
<td>Cyl.temp comp.x</td>
<td>T_b</td>
</tr>
<tr>
<td></td>
<td>Compressor power consumption</td>
<td>Power LT consumer</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>Compressor speed (% of maximum)</td>
<td>Anal.var.speed.compr.</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>Compressor operation (on/off)</td>
<td>Control comp.stage.x</td>
<td>-</td>
</tr>
<tr>
<td>MT Compressors:</td>
<td>Compressor inlet temperature</td>
<td>Suction gas temp.</td>
<td>T_c</td>
</tr>
<tr>
<td></td>
<td>Compressor outlet temperature</td>
<td>Cyl.temp comp.x</td>
<td>T_d</td>
</tr>
<tr>
<td></td>
<td>Suction gas pressure (p_medium)</td>
<td>(LT): pc Act</td>
<td>P_medium</td>
</tr>
<tr>
<td></td>
<td>Discharge pressure (p_high)</td>
<td>(MT): pc Act</td>
<td>P_high</td>
</tr>
<tr>
<td></td>
<td>Compressor power consumption</td>
<td>Power MT consumer</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>Compressor speed (% of maximum)</td>
<td>Anal.var.speed.compr.</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>Compressor operation (on/off)</td>
<td>Control comp.stage.x</td>
<td>-</td>
</tr>
<tr>
<td>De-superheater:</td>
<td>Heat recovery outlet temperature</td>
<td>HR outlet temp.</td>
<td>T_e</td>
</tr>
<tr>
<td>Gas cooler:</td>
<td>Gas cooler outlet temperature</td>
<td>GC outlet temp.</td>
<td>T_f</td>
</tr>
<tr>
<td></td>
<td>Gas cooler fan speed.</td>
<td>Anal.var.speed.fan</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>Condensing temperature (if sub.cr.)</td>
<td>(MT): tc</td>
<td>-</td>
</tr>
<tr>
<td>Receiver:</td>
<td>Receiver pressure</td>
<td>Mean pressure</td>
<td>P_rec</td>
</tr>
<tr>
<td>Heat exchanger:</td>
<td>Liquid outlet temperature</td>
<td>Temperature Rx.1</td>
<td>T_i</td>
</tr>
<tr>
<td>MT evaporator:</td>
<td>Evaporation temperature</td>
<td>(MT): to</td>
<td>Tevap_MT</td>
</tr>
<tr>
<td></td>
<td>Internal superheat from cabinets</td>
<td>Superheating</td>
<td>ISH_MT</td>
</tr>
<tr>
<td></td>
<td>Total superheat</td>
<td>SH suction gas</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>Defrosting signal (1/0)</td>
<td>Defrosting</td>
<td>-</td>
</tr>
<tr>
<td>LT evaporator:</td>
<td>Evaporation temperature</td>
<td>(LT): to</td>
<td>Tevap_LT</td>
</tr>
<tr>
<td></td>
<td>Internal superheat from cabinets</td>
<td>Superheating</td>
<td>ISH_LT</td>
</tr>
<tr>
<td></td>
<td>Total superheat</td>
<td>SH suction gas</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>Defrosting signal (1/0)</td>
<td>Defrosting</td>
<td>-</td>
</tr>
<tr>
<td>Ambient:</td>
<td>Room temperature</td>
<td>Room temperature</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>Outdoor temperature</td>
<td>Outdoor temperature</td>
<td>-</td>
</tr>
</tbody>
</table>
5.2.2 Calculations of Main Parameters for TR4 and TR5

The input data for the calculations are shown in Table 5-2. The data available in LDS are: temperatures, pressures, compressor signal (on/off) for all compressors and compressor speed for the frequency controlled compressors (given as percent of full capacity). The power consumption is also measured for the compressors. There are no pumps in the CO2-systems but the power consumption of the gas cooler fans is included in the measured values of power consumption for the MT compressors. Bitzer compressors are used in both systems and performance data is available from the Bitzer homepage [BIT10]. The calculation procedure is essentially the same as for CC2 and all calculations of enthalpy, entropy and specific volume are made in Refprop. The internal superheat is measured for each cabinet and an average value for all cabinets is added to the evaporation temperature in order to obtain the cabinet exit temperature. The shaft power is calculated for the LT-compressors using equation 5.27.

\[ \dot{E}_{\text{comp,LT shaft}} = \dot{m}_{\text{LT}} \cdot \Delta h_{\text{comp,LT}} \]  \hspace{1cm} 5.27

The parallel compressors have the same inlet enthalpy but different outlet enthalpies. All these enthalpies are calculated in Refprop from temperatures and pressures but in order to find the common outlet enthalpy for a number \( n \) of parallel compressors, a heat balance is required as shown in equation 5.28.

\[ \sum_i (\dot{m}_{\text{comp},i} \cdot h_{\text{comp, out },i}) = \dot{m}_{\text{tot}} \cdot h_{\text{comp, out tot}} \]  \hspace{1cm} 5.28

The mass flow through the LT compressor is the same as through the LT evaporator but for MT, the mass flow through the compressor is the total mass flow of the system as seen in figure 4.5. The mass flows through the compressors (\( m_{\text{LT}} \) and \( m_{\text{MT}} \)) are obtained the same way as the compressor mass flows for CC2, by using equation 5.2-5.4 with volumetric efficiency as a function of pressure ratio taken from compressor performance data. However, the frequency in equation 5.4 is now calculated by using equation 5.29 where \( f_{\text{min}} \) and \( f_{\text{max}} \) are the minimum and maximum frequencies for the compressor and [% of maximum] is data from LDS.

\[ f = f_{\text{min}} + \frac{(f_{\text{max}} - f_{\text{min}}) \cdot [\% \text{ of maximum}]}{100} \]  \hspace{1cm} 5.29

The cooling capacities are calculated for the LT and MT stages by using equation 5.30 and 5.31.

\[ \dot{Q}_{\text{evap,LT}} = \dot{m}_{\text{LT}} \cdot \Delta h_{\text{evap,LT}} \]  \hspace{1cm} 5.30

\[ \dot{Q}_{\text{evap,MT}} = \dot{m}_{\text{MT}} \cdot \Delta h_{\text{evap,MT}} \]  \hspace{1cm} 5.31

The enthalpies at the LT- and MT evaporator inlets are the same as the enthalpy of the liquid exiting the heat exchanger in Figure 4.5. This enthalpy is calculated from measured temperature of the liquid and the receiver pressure. The temperature after the evaporators is found from the internal superheat of the cabinets and since the pressure is known, the outlet enthalpies can be calculated. The mass flow through the evaporators can be calculated through a mass balance but first, the mass flow in the gas bypass line must be found. A saturated condition is assumed in the receiver and the vapor quality is calculated from
pressure and enthalpy at the receiver inlet (position g in Figure 4.6). The mass flows in the vapor and liquid lines are calculated with equation 5.32 and 5.33 and the mass flow in the MT evaporator is found with equation 5.34.

\[
\dot{m}_{\text{rec,vap}} = \dot{m}_{\text{tot}} \cdot \text{vapor quality} \\
\dot{m}_{\text{rec,liq}} = \dot{m}_{\text{tot}} \cdot (1 - \text{vapor quality}) \\
\dot{m}_{\text{MT}} = \dot{m}_{\text{rec,liq}} - \dot{m}_{\text{LT}}
\]

In order to separate the fan power, which is included in the measured value of MT power consumption, from the compressor power consumption, the MT compressor power consumption is calculated using equation 5.35. The LT compressor power consumption is also calculated in equation 5.36. In both equations, the isentropic enthalpies are found from inlet entropies and discharge pressures.

\[
\dot{E}_{\text{comp,MT,calc}} = \dot{m}_{\text{MT}} \cdot \frac{h_{\text{comp,out,is,MT}} - h_{\text{comp,in,MT}}}{\eta_{\text{tot,MT}}} \\
\dot{E}_{\text{comp,LT,calc}} = \dot{m}_{\text{LT}} \cdot \frac{(h_{\text{comp,out,is,LT}} - h_{\text{comp,in,LT}})}{\eta_{\text{tot,LT}}}
\]

The power consumption of the gas cooler fans is calculated by equation 5.37:

\[
\dot{E}_{\text{GC,fans}} = \dot{E}_{\text{MT,measured}} - \dot{E}_{\text{comp,MT,calc}}
\]

The total cooling load on the high stage compressors is calculated using equation 5.38.

\[
\dot{Q}_{\text{2,high,tot}} = \dot{Q}_{\text{evap,LT}} + \dot{Q}_{\text{evap,MT}} + \dot{E}_{\text{comp,LT,shaft}}
\]

The coefficient of performance for the MT stage is calculated with and without the fan power included as seen in equation 5.39 and 5.40 respectively.

\[
\text{COP}_{\text{MT}} = \frac{\dot{Q}_{\text{2,high,tot}}}{\dot{E}_{\text{MT,measured}}} \\
\text{COP}_{\text{comp,MT}} = \frac{\dot{Q}_{\text{2,high,tot}}}{\dot{E}_{\text{comp,MT,calc}}}
\]

The part of the high stage power consumption used only for MT applications is calculated with and without the fan power according to equation 5.41 and 5.42.

\[
\dot{E}_{\text{high,MT,meas}} = \frac{\dot{Q}_{\text{evap,MT}}}{\text{COP}_{\text{MT}}} \\
\dot{E}_{\text{high,MT,calc}} = \frac{\dot{Q}_{\text{evap,MT}}}{\text{COP}_{\text{comp,MT}}}
\]

The part of the high stage power consumption used only for LT applications is calculated with and without the fan power according to equation 5.43 and 5.44.
The coefficient of performance for the LT stage is calculated using equation 5.45 with the fan power included and equation 5.46 without the fan power. The calculated value of LT compressor power consumption is used in both equations due to the large tolerances of the measured values.

\[
COP_{LT} = \frac{\dot{Q}_{evap,LT}}{E_{comp,LT,calc} + \dot{E}_{high,LT,meas}}
\]

\[
COP_{comp,LT} = \frac{\dot{Q}_{evap,LT}}{E_{comp,LT,calc} + \dot{E}_{high,LT,calc}}
\]

The coefficient of performance is calculated for the total system according to equation 5.47 and 5.48, with and without the fan power included respectively.

\[
COP_{tot} = \frac{\dot{Q}_{evap,LT} + \dot{Q}_{evap,MT}}{E_{MT,measured} + E_{comp,LT,calc}}
\]

\[
COP_{comp,tot} = \frac{\dot{Q}_{evap,LT} + \dot{Q}_{evap,MT}}{E_{comp,MT,calc} + E_{comp,LT,calc}}
\]

### 5.3 Load ratio correction

When calculating the total COP for the different systems, the operating conditions vary between the systems. The calculations made so far only show how well the systems operate under their respective conditions but in order to compare different systems, these conditions have to be equalized. The load ratio is the relation between the cooling capacity of the medium temperature cabinets to that of the low temperature cabinets. An approximate value for the load ratio of a typical European supermarket is 3, i.e. three times more cooling capacity for MT cabinets than for LT cabinets [FRE09]. In order to ensure that the load ratio is the same for each system, an equation for load ratio correction is used (equation 5.49) and the load ratio is set to the value 3 [Ibid]. The load ratio correction for the total compressor COP (excluding parasites) is made using equation 5.50.

\[
COP_{tot,LR,corr} = \frac{1}{LR + \frac{1}{LR \cdot COP_{LT} + \frac{1}{COP_{MT}}}}
\]

\[
COP_{comp,tot,LR,corr} = \frac{1}{LR + \frac{1}{LR \cdot COP_{comp,LT} + \frac{1}{COP_{comp,MT}}}}
\]
5.4 Compressor data

Performance data is available from the compressor manufacturers for different values of evaporation and condensing temperatures. These data include the amount of superheat and subcooling, the power consumption ($E$), cooling capacity ($Q_{evap}$) and volumetric flow rate ($V$). The high and low pressures are known from the evaporation and condensing temperatures and the enthalpies can then be calculated with Refprop before and after the evaporator ($h_{evap,in}$ and $h_{evap,out}$) and at the compressor inlet ($h_{comp,in}$). The mass flow is either already available as a measured value, or calculated using equation 5.51.

$$\dot{m} = \frac{Q_{evap} }{h_{evap,out} - h_{evap,in}} \quad 5.51$$

The specific volume ($v_{comp,in}$) and the entropy at the compressor inlet are calculated from pressure and temperature and the isentropic enthalpy ($h_{comp,out,is}$) at the compressor outlet is calculated with Refprop using the constant entropy for isentropic expansion and the high pressure. The real compressor outlet enthalpy is then calculated with equation 5.52.

$$h_{comp,out} = h_{comp,in} + \frac{E}{\dot{m}} \quad 5.52$$

The volumetric efficiency is calculated with equation 5.53 and the total efficiency with equation 5.54.

$$\eta_s = \frac{\dot{m} \cdot v_{comp,in}}{V} \quad 5.53$$

$$\eta_{tot} = \frac{h_{comp,out,is} - h_{comp,in}}{h_{comp,out} - h_{comp,in}} \quad 5.54$$

Figure 5.2 shows the total efficiency of the different compressors as functions of pressure ratio based on compressor manufacturer data. The R404A scroll compressors of CC2 (Copeland ZB220KCE-TWM) can clearly operate with higher pressure ratios than the compressors working with CO2. Compressor data for the two MT compressors of TR4 was not obtained from Bitzer but supplied by the company that built the entire refrigeration system, based on their own performance measurements. In figure 5.2, one of these compressors showed the highest total efficiency followed by the MT compressors of TR5. The LT compressors of TR4 and TR5 had almost identical total efficiencies and these were also the lowest of all compressors. The volumetric efficiencies for all compressors except for unit KS4 are shown in figure 5.3.

The volumetric efficiency for the CO2 compressors can vary somewhere between 0.62 and 0.92 for pressure ratios between 1.6 and about 4. The highest volumetric efficiencies are found for the R404A compressors which also have a greater range for pressure ratios than the CO2 compressors.
Figure 5.2: Total compressor efficiencies as functions of pressure ratios based on manufacturer data.

Figure 5.3: Compressor volumetric efficiencies as functions of pressure ratios based on manufacturer data.
6 Results

This chapter contains the results of the analysis for each system. The main parameters that have been investigated to determine the efficiency of the systems are cooling capacity, power consumption and coefficient of performance. For the trans-critical CO2-systems, the operation of heat recovery systems has also been investigated. Emphasis has also been placed on evaluating the influence of different parasites in the systems such as pumps and fans. Most of the results are displayed as monthly average values but some contain data over a shorter time period for instance to show how the values of certain parameters vary between day and night.

6.1 Cascade System CC2

The cascade system, having the most complex structure of the three systems, will require a more thorough analysis of its different parts before evaluating the entire system. The heat recovery of this system has not been included in this investigation due to the lack of measurement points at the system location.

6.1.1 Individual units

Figure 6.1 shows monthly averages of compressor power consumption for the three R404A-units and the CO2-unit in supermarket CC2 and the outdoor temperature. As expected, the power consumption peaks during July since it is the month with the highest average outdoor temperature, and drops during August and September due to the decrease in temperature.

![Figure 6.1: Monthly averages of outdoor temperature and compressor power consumption for the units of CC2.](image)
It is apparent that the ambient conditions have a much larger impact on the power consumption of the VKA medium temperature units than that of the low temperature CO2-unit KS4 which remains rather constant over the time period. Unit VKA2 has very low values of compressor power consumption compared to the other medium temperature units since it is rarely in operation. The values for VKA3 are about twice as large as the values for VKA1 as it also supplies the low temperature unit. The shape of the graphs for VKA1 and VKA3 are similar but the very low values in June and the high values in July can be questioned. This is even more obvious when looking at the cooling capacities in Figure 6.2. For VKA1, the cooling capacity increases from 35 kW in June to 55 kW in July. Such a large increase does not seem reasonable when comparing with the temperature difference. The monthly average of the outdoor temperature is about the same for June and September, but the cooling capacities have completely different values. This also affects the calculations on the basis of the total system and this is further investigated in chapter 7.

In Figure 6.2, VKA3 has the highest cooling capacity of about 60 kW followed by VKA1 at between 35-55 kW. VKA2 contribute with about 11 kW in July and 6 kW in August but almost nothing in June and September. The low temperature unit KS4 is very stable at about 23 kW cooling capacity and does not change much with the ambient temperature. All the VKA-units show the highest values in July when the ambient temperature is at its highest. The increase in cooling capacity and compressor power consumption in July is mainly due to the increase in relative humidity of the indoor and outdoor air when the ambient temperature increases. The reason why the low temperature unit is not affected the same way is mainly due to the presence of glass doors on the freezers and the fact that horizontal cabinets contain the cold better than vertical cabinets.

Figure 6.2: Monthly averages of outdoor temperature and cooling capacity for the different units of CC2.
In Figure 6.3, the coefficient of performance is shown for the different units of CC2 together with outdoor temperature based on monthly averages. It is the compressor COP that is shown, calculated by using compressor power consumption and excluding parasites such as dry cooler fans and pumps.

As expected, the CO2 unit KS4 has the highest value of COP which is rather constant, about 4. Unit VKA1 has a COP of about 3.6 while VKA3 has much lower values of about 2.2. One reason for these low values is the low evaporation temperature required in unit VKA3 in order to absorb heat from the KS4 condenser as described in chapter 4. The high values for VKA2 in June and September may be because of the fact that VKA2 is almost never in operation during these two months. The very short periods of operation for VKA2 result in faulty values in the data series and even though these faults can, for the most part, be eliminated by filtering the data, some parameters will be over- or underestimated which has an impact on the results for series with few data points. Looking at July and August, when VKA2 is running more frequently, it is clear that it reaches similar values of COP as VKA1. All systems, including KS4, show an increase in COP with falling ambient temperature.

6.1.2 Analysis of VKA3

The figures 6.1 and 6.2 depict how the different units of CC2 operate and it is clear that VKA3 is the main unit that drives the system. The large contribution of cooling capacity merits a closer look at unit VKA3. Figure 6.4 shows the ratios between the MT cooling capacities that each unit produces to the total cooling capacity used for medium temperature applications. From this diagram, it is clear that the average contribution from VKA3 is between 35 and 45 percent of the total MT cooling capacity. The rest is supplied mainly by VKA1 but VKA2 also contributes with about 11 and 7 percent during the two warmest
months, July and August respectively. This shows that the VKA2 unit is not only used as a backup-unit in case VKA1 fails, but that it actually is in operation when the outdoor temperature reaches a certain point and the cooling load increases.

While producing a large portion of the cooling capacity for the medium temperature cabinets, VKA3 also drives the low temperature CO2-cycle by being connected to its condenser as described in chapter 4. Figure 6.5 shows the ratio between the VKA3 cooling capacity used for low temperature applications and the total cooling capacity produced by VKA3. Evidently, about 50 percent of the total VKA3 cooling capacity is used to supply the low temperature unit KS4. During the warmest month, July, this value drops in favor of the MT cooling capacity as the medium temperature cooling cabinets are more sensitive to an increase in ambient temperature.

Figure 6.4: Ratios of cooling capacity that VKA1, VKA2 and VKA3 each supply to the medium temperature cabinets based on monthly averages.

Figure 6.5 also shows the relation between LT and MT use of cooling capacity when considering the total production of system CC2. About 74-81 percent is used for MT-applications. This makes sense because of the very large number of medium temperature cabinets that are used in the store-area of supermarket CC2. The highest percentage of cooling capacity for MT use is found in July when the cooling load is at its highest.
The power consumption of VKA3 including parasites is shown in Figure 6.6. The power used for supporting the KS4 unit is about 12.8-14.6 kW. The power used for MT applications is about the same, around 13 kW but increases to 17 kW in July.

6.1.3 Overall system performance CC2

In the previous chapters, the results of the analysis for the individual units of CC2 have been shown. These results do not include parasites like pumps or the dry coolers but are only based on compressor power consumption. In order to make a valid comparison of the systems, the total power consumption must be taken into account. Figure 6.7 shows the power consumption and cooling capacity for LT and MT applications for and for the total system. The LT cooling capacity for the freezers is supplied by the CO2 unit KS4 and is about 22 kW. The power consumption for LT use is the sum of the compressor power for KS4 and the part of the power consumption for VKA3 that is used for LT applications. In Figure 6.1, the LT
compressor power was about 6 kW and in Figure 6.6, the LT-power consumption for VKA3 is about 12.8-14.6 kW and together, the total power consumption for the LT-stage of the system is about 20 kW. The power consumption of the brine pump P2 is included in this number.

For the MT-stage, the cooling capacity is the sum of the cooling capacities for VKA1 and VKA2 and the MT-cooling capacity from VKA3. This amounts to about 80 kW but the value reaches 100 kW for July. As mentioned before, such a large increase in cooling capacity in only one month may not be reasonable and will be treated further in chapter 7. The power consumption is about 30-50 kW for the MT-stage. This includes compressor power consumption for VKA1 and VKA2 and the MT-power consumption for VKA3. The power consumption of pump P1, P3, P5, P6, P7 and the dry cooler fans are also included in this value. The total cooling capacity for the entire system is the sum of LT and MT cooling capacities and is between 90 and 125 kW. The total power consumption for CC2 is between 50 and 70 kW. The cooling capacity and power consumption for the freezers remains very constant for time period but for the chillers, and thereby also the total system, the values are more affected by the ambient conditions. Figure 6.8 shows the COP for the LT and MT stages, the total system COP and the total COP using load ratio correction, with and without the power consumption of pumps and fans included in the calculations. The influence of parasites on system performance is discussed further in chapter 6.6.

![Figure 6.7: Monthly averages of LT, MT and total cooling capacities and power consumption for CC2.](image)

The COP for the low temperature stage is about 1.2 compared to 2.1 for the medium temperature stage (four month averages). This results in a total COP of about 1.8 (four month average). It is a comparatively low value of COP and it is mainly because the VKA3 unit has to supply the KS4 unit with about 50 percent of the total cooling capacity it produces as shown in Figure 6.5. When excluding the influence of parasites, the COP is about 1.2 for the LT stage, 2.6-3.0 for the MT stage and 2.0-2.2 for the total system.
6.1.4 Cabinets

The brine that flows to the medium temperature cabinets in the store and back to the compressor room must have a sufficiently low temperature in order to absorb heat from the cabinets. The monthly averages for brine inlet and outlet temperatures to the medium temperature cabinets are shown in Figure 6.9. The inlet temperature has values between -2.4 and -3.4°C which are very high for a system of this size because of the long distribution lines and subsequent external heating of the refrigerant. The return temperature is between 0.4 and -0.4°C and this means that the temperature difference over all MT cabinets is only about 3°C.
Because of the fairly high inlet brine temperature to the MT-cabinets and the long distance the refrigerant has to be transported through the store area, there has been some concern that the temperatures inside the cabinets will be too high. Table 6-1 shows outlet temperatures for a selected group of MT cabinets from different parts of the store area. This data is previous to the time frame of this thesis but it can serve as an indication of what kind of temperature levels that can be expected. Despite of the high brine temperature, the cabinets appear unaffected and maintain appropriate temperature levels.

Table 6-1: Temperature samples in the MT cabinets of CC2 for three months.

<table>
<thead>
<tr>
<th>Cabinet No.</th>
<th>Dec</th>
<th>Jan</th>
<th>Feb</th>
</tr>
</thead>
<tbody>
<tr>
<td>K10A</td>
<td>0.5449</td>
<td>0.4250</td>
<td>0.3210</td>
</tr>
<tr>
<td>K10B</td>
<td>1.7889</td>
<td>1.6245</td>
<td>1.5958</td>
</tr>
<tr>
<td>K10C</td>
<td>1.7894</td>
<td>1.6273</td>
<td>1.6282</td>
</tr>
<tr>
<td>K14A</td>
<td>3.2302</td>
<td>3.1616</td>
<td>3.1957</td>
</tr>
<tr>
<td>K14B</td>
<td>3.1936</td>
<td>3.1563</td>
<td>3.1008</td>
</tr>
<tr>
<td>K14C</td>
<td>3.2042</td>
<td>3.1564</td>
<td>3.1694</td>
</tr>
<tr>
<td>K14D</td>
<td>3.2445</td>
<td>3.2059</td>
<td>3.2021</td>
</tr>
<tr>
<td>K14E</td>
<td>3.1837</td>
<td>3.1569</td>
<td>3.1301</td>
</tr>
<tr>
<td>K18A</td>
<td>5.7342</td>
<td>5.6962</td>
<td>5.7314</td>
</tr>
<tr>
<td>K18B</td>
<td>5.3074</td>
<td>5.2470</td>
<td>5.2418</td>
</tr>
<tr>
<td>K18C</td>
<td>6.1743</td>
<td>6.3661</td>
<td>6.0473</td>
</tr>
<tr>
<td>K22</td>
<td>3.0532</td>
<td>3.0507</td>
<td>3.0384</td>
</tr>
</tbody>
</table>
6.2 System TR4

This chapter contains some of the results for the smallest of the three systems, TR4. A separate analysis has been made for this system just as for the cascade system and the main parameters shown here are cooling capacity, energy consumption and coefficient of performance. Results have been obtained for the low and medium temperature application of these parameters and also on a total basis.

6.2.1 System performance

Figure 6.10 shows monthly averages of power consumption and cooling capacity used for LT and MT applications and for the total system as well as the ambient temperature and condensing temperature. The measured values of power consumption are used for the MT-stage since these values also contain the power consumption of the gas cooler fans. However, because of the large tolerances of the measured values, the calculated compressor power for the LT-compressors is used as the power consumption for the LT stage.

![Figure 6.10: Monthly averages of ambient and condensing temperatures, LT, MT and total power consumption and cooling capacity, for TR4.](image)

The LT compressor power and cooling capacity are not much affected by the change in ambient temperature with rather constant values of about 1.5 and 6 kW respectively. The MT cooling capacity amounts to about 21 kW and is clearly more dependent on the ambient temperature. So is the MT compressor power with about 7-11 kW, reaching its peak in July. The total cooling capacity for TR4 is about 26-29 kW and the total power consumption 9-12 kW. The difference between condensing temperature and ambient temperature seems to be as much as 10°C. Figure 6.11 shows the LT, MT and total COP without the influence of
system parasites. The figure also includes ambient, evaporation and condensing temperatures and the total COP with LR=3. The LT COP is around 1.5-1.7 and the MT COP is between 3.1 and 3.9. This results in a total COP for TR4 between 2.5-3.0. The same plot is displayed in figure 6.12 but with the influence of system parasites included. For this plot, the LT COP is about 1.5-1.7, the MT COP about 2.9-3.8 and the total COP about 2.4-2.9.

![Figure 6.11](image1.png)

**Figure 6.11:** Monthly averages of LT, MT and total COP (excluding parasites), ambient temperature, evaporation (in brackets) and condensing temperatures and total COP corrected for a load ratio of 3 for TR4.

![Figure 6.12](image2.png)

**Figure 6.12:** Monthly averages of LT, MT and total COP (including parasites), ambient temperature, evaporation (in brackets) and condensing temperatures and total COP corrected for a load ratio of 3 for TR4.
6.3 System TR5

The following section contains the results for system TR5. The main parameters shown here include cooling capacity, power consumption and coefficient of performance. Values of these parameters are shown for the low and medium temperature stages and for the total system. All values are based on monthly averages.

6.3.1 System performance

The power consumption and cooling capacity for LT and MT use and for the total system is shown in Figure 6.13 together with the ambient and condensing temperature. The LT power consumption is about 6 kW compared to about 20-27 kW for the MT stage. The values for the MT stage are based on measured data and include gas cooler fans while the LT-data is calculated due to the large tolerances of the measured values. The total power consumption for TR5 is between 25 and 34 kW. The LT cooling capacity is about 22 kW and the MT cooling capacity somewhere between 51 and 61 kW. This yields a total cooling capacity of about 74-83 kW for TR5. The MT cooling capacity and power consumption depend on the ambient temperature, showing the highest values in July when the temperature is at its peak. For LT results, the values don’t vary a lot with the ambient conditions. The temperature difference between condensing temperature and ambient temperature is about 8°C for TR5.

The LT, MT, total and load ratio corrected total COP is shown in figure 6.14 (excluding parasites) and in figure 6.15 (including parasites). The ambient, condensing and evaporation temperatures are also included. In figure 6.14, the COP is about 1.6-1.9 for the LT stage and 3.8-4.8 for the MT stage, resulting in a total COP of 2.8-3.2. When the power consumption of...
gas cooler fans is included in the calculations, the COP is about 1.5-1.8 for the LT stage, 3.2-4.1 for the MT stage and 2.5-2.9 for the total system as shown in figure 6.15. The values of total COP for TR5 are similar to those of TR4.

Figure 6.14: Monthly averages of LT, MT and total COP (excluding parasites), ambient temperature, evaporation (in brackets) and condensing temperatures and total COP corrected for a load ratio of 3 for TR5.

Figure 6.15: Monthly averages of LT, MT and total COP (including parasites), ambient temperature, evaporation (in brackets) and condensing temperatures and total COP corrected for a load ratio of 3 for TR5.
6.4 System comparison

The three systems are first compared on the basis of the actual measured and calculated data to investigate how well the systems are operating under their respective conditions. Second, the comparison is made using the equation for load ratio correction to equalize the operating conditions of the systems.

6.4.1 Comparison of Measured Data

Figure 6.16 shows the average condensing temperatures per month for all three systems including the separate units of CC2. Unit VKA3 has the highest condensing temperature, about 34°C, followed by VKA1 with about 26-31°C. All systems and units except the LT-unit KS4 display the highest values during the warmest months. The condensing temperature of KS4 is dependent on the brine temperature and not on the ambient. Although similar in construction, TR4 has a higher condensing temperature than TR5 and the difference is about 3°C.

Looking at the levels of the condensing temperature compared to the ambient temperature for TR4 and TR5 in Figure 6.12 and 6.13, the condensing temperature does not follow the ambient temperature exactly. In July and most of all in September, the condensing temperature is kept at a higher level when the ambient temperature drops. This is because the need for heat recovery has increased and the compressor discharge pressure is raised to meet this demand. This results in higher condensing temperatures and lower COP. If this investigation included the winter months as well, the difference between ambient and condensing temperature would be even greater.
Figure 6.17 shows the monthly average evaporation temperatures for TR4, TR5 and the different units of CC2. System TR4 has an evaporation temperature of about -31°C and TR5 is slightly lower at -32°C for the LT cabinets. For the MT cabinets, the evaporation temperature is very similar for both systems, about -7°C. For CC2, the CO2-unit KS4 has the lowest evaporation temperature at -35°C. This means that there is a difference of 3-4 degrees between the reference system CC2 and the trans-critical systems regarding the evaporation temperature for LT cabinets.

![Evaporation temperatures for TR4, TR5 and the units in CC2 based on monthly averages.](image)

---

The units VKA1 and VKA2 have the same evaporation temperature since they are connected. VKA3 has the lowest evaporation temperature of the MT-units of CC2 at about -17°C. This is twice as low as for the other VKA units but it is necessary to achieve a low brine temperature at the inlet to the condenser of unit KS4. All evaporation temperatures appear very stable over time and show no signs of being affected by changes in ambient temperature.

### 6.4.2 Comparison of Main Parameters

The main results from the individual system analysis have been put together to make comparisons. The cooling capacity and power consumption for the LT stage is plotted for the different systems in Figure 6.18. The cooling capacities for the freezers are at similar levels for TR5 and CC2. The values for CC2 are slightly higher but this system also has a few more cabinets than TR5. At the same time, the power consumption differs a lot between the two systems. For TR5, the LT power consumption is about 6 kW but for CC2, it has much higher values at about 20 kW, mainly because of the higher cooling capacity required. For TR4, the LT power consumption is about 1.5 kW and the LT cooling capacity about 6 kW.
In Figure 6.19, the cooling capacities and power consumption for MT applications is shown for the three systems. CC2 has the highest MT cooling capacity with an average of about 80 kW over the four months followed by TR5 with about 51-61 kW. The systems CC2 and TR5 are roughly the same size but CC2 has a few more MT cabinets. The MT power consumption for CC2 has an average value of about 37 kW over the four months compared to 23 kW for TR5. The MT cooling capacity for TR4 is around 21 kW and the MT power consumption around 7-11 kW.

The values of total cooling capacity and total power consumption are shown in Figure 6.20. CC2 has the largest total cooling capacity with 90-125 kW followed by TR5 at about 78 kW, both peaking in July. The total power consumption of CC2 is somewhere between 50 and 70 kW and for TR5 only between 25 and 34 kW. For TR4, the total cooling capacity is about 26-29 kW and the total power consumption 9-12 kW.
Figure 6.19: Monthly averages of MT cooling capacity and power consumption for CC2, TR4 and TR5.

Figure 6.20: Monthly averages of total cooling capacity and power consumption for CC2, TR4 and TR5.
The corresponding coefficients of performance for the LT and MT stages and for the total systems are shown in figure 6.21. In the middle of the summer (July and August), the ambient temperatures are higher than for June and September and in order for the systems to continue removing heat in the gas and dry coolers, the condensing temperatures have to increase as well. As seen previously in Figure 6.16 and 6.17, the evaporation temperatures are not very dependent on the ambient conditions compared to the condensing temperatures. This decrease in COP during the warmer months July and August, shown in Figure 6.21, is therefore mainly due to the increase in condensing temperature. This requires a higher discharge pressure and a higher compressor power consumption which results in a lower value of total COP. The increased cooling loads on the cabinets due to higher humidity and is also a reason for lower values of COP.

TR4 and TR5 have very similar total COP for all months, starting at about 2.9 in June, dropping to 2.4-2.5 in July and gradually increasing to 2.9 – 3 in September. It makes sense that the level of the total COP should be very similar in June and September since the average ambient temperatures are also very similar (about 1.3 °C difference for TR4 and 1.9 °C for TR5). Based on four month averages, the total COP for CC2 is about 1.8 compared to 2.7 for TR4 and 2.8 for TR5. For the MT stages, TR5 has a four month average COP of 3.8 compared to 3.4 for TR4. The COP for the medium temperature units of CC2 is about 2.1 (four month average). For the LT stages, system TR4 and TR5 are very similar with values of about 1.6 and 1.7 respectively (four month averages). This is also much higher than CC2 which has a four month average LT COP around 1.2 for low temperature COP. The total COP of the cascade system appears to be more constant than those of the CO2 systems possibly due to the rather constant condensing temperature of unit VKA3 as seen in figure 6.22. The decrease in COP for warmer ambient temperatures is because the condensation temperature/pressure has to increase which means that more compressor power is needed.
At the same time, higher ambient temperature and humidity increases the cooling load and the frost accumulation in the cabinets. This reduces the system efficiency since a layer of ice on the evaporator lowers the heat transfer from the cabinets to the refrigerant.

Figure 6.22 shows the COP for the low and medium temperature stages and the total COP for TR4 and TR5 as well as the compressor COP for each of the R404A units in system CC2. Clearly, the main units of the cascade system, VKA1 and VKA3, operate with higher condensing temperatures than the trans-critical CO2 systems. VKA3 has the highest condensing temperatures with 33-35°C which is one of the reasons for the low coefficient of performance for this unit compared to VKA1. The condensing temperature is also more stable for this system as shown by the small spread of the points in figure 6.22. The low average condensing temperatures of VKA2 is a result of the fact that it is rarely in operation. All values of COP increase with decreasing condensing temperatures which is reasonable since less compressor work is needed to reach the corresponding discharge pressure.

The condensing temperatures for TR4, TR5 and the VKA units of CC2 are shown in figure 6.23 as functions of the ambient. The inclination of the graphs gives an indication of how the systems depend on the ambient. Unit VKA3 seems less dependent on the outdoor temperature while VKA1 appears to be more dependent. TR5 also has a stronger inclination than the other systems. During the time period of the study, the heat recovery in TR4 and TR5 has not been in operation very frequently and the influence of the heat recovery on condensing temperature is therefore not so obvious. When the heat recovery is used, the high stage discharge pressure is raised and this affects the level of the condensing temperature but the influence will be more clearly visible during the colder winter months when heat recovery is used frequently. The heat recovery is discussed further in chapter 6.5.
Figure 6.23: Condensation temperature as a function of ambient temperature for CC2, TR4 and TR5 based on monthly averages.

6.4.3 Dry- and gas coolers.

So far, the system analysis has focused on cooling capacity, power consumption and the coefficient of performance but in this chapter, the operation of the dry- and gas coolers has been included as well for the sake of comparing the different systems.

The heat removed in the gas cooler of TR4 and TR5 and in the dry coolers of CC2 has been plotted in Figure 6.24. The figure also shows how the gas coolers are controlled by displaying the percentage of full capacity. These values do not include the heat removed in de-superheaters. It is clear that the gas cooler fans are operating at almost 100% capacity in July and that this value drops with the ambient temperature. The gas coolers of TR4 are operating at about 10 percent lower capacity than in TR5. The shape of the graphs for gas cooler fan operation is identical with the graphs for condenser power. TR4 has values of about 30 kW and TR5 has values of about 80-105 kW. No signal for the dry cooler fan capacity has been obtained for this investigation but the heat removed in the dry coolers is between 130-170 kW according to the calculations. The energy consumption of the dry cooler shown in chapter 6.6 also gives some indication of how the fans operate.

The amount of sub-cooling in the gas bypass heat exchangers and in the condensers/gas coolers (only for sub-critical operation) of TR4 and TR5 are shown in figure 6.25. The sub-cooling in the gas coolers is about 4K for TR5 and three degrees for TR4. The sub-cooling in the gas bypass heat exchangers is very small, only 1K or less for both systems.
Figure 6.24: Heat rejected in the dry- and gas coolers for CC2, TR4 and TR5 and the gas cooler fan operation for TR4 and TR5 in percent of full capacity, based on monthly averages.

Figure 6.25: Amount of sub-cooling in condensers (only sub-critical operation) and in gas bypass line heat exchangers for TR4 & TR5 based on monthly averages.

6.4.4 Cabinet comparison

One of the main components in a refrigeration system has not yet been treated in this report, the cabinets. The storage of chilled and frozen food products by using cooling cabinets and freezers is in fact the main reason for having a refrigeration system in a supermarket and therefore, the operation of the cabinets is investigated in this chapter.
Figure 6.26 show the average time for defrosting for the freezers and chillers of TR4 and TR5. The total time of defrosting has been calculated for each cabinet during each month and the values shown in the diagram are averages for all LT and MT cabinets for each month divided by the number of days that data has been available for each month. This way, the influence of gaps in the data series, as previously discussed in chapter 5, is eliminated. It is clear that the chillers require 3-4 times more defrosting time than the freezers. The reason for this could be that the medium temperature cabinets don’t use electric defrost but instead, the refrigerant flow is stopped and air with room temperature is allowed to flow across the evaporator, thereby warming it and removing the frost. Higher temperature and humidity during the summer increases the frost accumulation in the cabinets and more defrosting is needed. However, the differences are very small for the four month period and since the values are the result of averaging all cabinets, these minor differences may not be of any importance. The MT cabinets show an average defrosting time of about 2 and 2.5 hours per day for the MT cabinets of TR4 and TR5 respectively. The same values for the LT cabinets are much smaller, at about 0.7 and 0.4 hours per day for TR4 and TR5 respectively.

The amount of internal superheat was compared for the LT and MT cabinets of TR4 and TR5 and for the LT cabinets of CC2. Figure 6.27 shows the average values of internal superheat for these cabinets per month. The internal superheat for the MT cabinets of CC2 was not used for the calculations but the temperature difference of the brine inlet and outlet was included in the figure.

The LT cabinets of CC2 showed the highest values of internal superheat at about 13.5K followed by the LT cabinets of TR5 at 9-10K. For TR4, the LT and MT internal superheat was rather similar at about 6-7K and this is also the level of the MT internal superheat of TR5.
The brine temperature difference between the inlet and outlet to the MT cabinets was very stable at about 3K. No obvious relation between ambient temperature and the level of internal superheat was found but since the values shown in the figure are based on averages for all LT and MT cabinets of each system, it may be difficult to observe the influence of the ambient.

![Chart showing brine temperature difference and internal superheat](image)

**Figure 6.27**: Internal superheat for LT cabinets of CC2, TR4 and TR5, internal superheat for MT cabinets of TR4 and TR5 and brine temperature difference over MT cabinets for CC2 based on monthly averages.

### 6.4.5 Comparison with Load Ratio Correction

The load ratio is the relation between cooling capacities for medium and low temperature applications, i.e. the ratio of MT-cooling capacity over LT-cooling capacity. This value is different for different systems and will change with ambient conditions. When the ambient temperature and humidity increase, so does the load ratio, mostly because of the increase in MT cooling capacity since MT cabinets are more dependent on the ambient conditions than LT cabinets. For a typical European supermarket it is common with a value of 3 for the load ratio [FRE09]. Figure 6.28 show the load ratios and the outdoor temperature for each of the systems. It is clear that the systems CC2 and TR5 have load ratios that are mostly higher than 3 while TR4 is well below this value. The effect of the high value of medium temperature cooling capacity in July and the low value in June for CC2 are clearly visible in the graph. The value starts at below three in June and increases to over 4 in July. All systems show the highest value in July when the ambient temperature is at its maximum at the dependence on ambient temperature is clear.
When comparing the total COP of different refrigeration systems, we are only looking at how well the systems are performing in their respective environments, under their respective conditions. To make a valid comparison of different systems, the same load ratio must be used in order to normalize the conditions. An equation for load ratio correction, Equation 5.49, is used for each system to calculate the total COP with a load ratio of 3. Figure 6.29 shows the actual total COP and the total COP when using this load ratio correction.

**Figure 6.28:** Load ratios and ambient temperatures for CC2, TR4 and TR5 based on monthly averages.

**Figure 6.29:** Monthly averages of total COP with and without load ratio correction (LR=3) for CC2, TR4 & TR5.
For TR5 the total COP increases slightly when using the corrected load ratio. As seen in Figure 6.28, the real load ratio was below 3 for this system. For TR4, the values were higher than 3 and the total COP decreased when using load ratio correction. System CC2 has also decreased its COP slightly when using a load ratio of 3. This system has the lowest total COP of about 1.70 – 1.76 and seems to be more stable than the trans-critical system over time. TR4 and TR5 have bigger change in COP for load ratio correction than CC2. Figure 6.30 shows the total COP with load ratio correction as a function of condensing temperature.

![Figure 6.30: Total COP for LR=3 as a function of condensation temperature based on monthly averages (the condensation temperatures for CC2 are average values for VKA1 and VKA3).](image)

The values of condensing temperature for CC2 in figure 6.30 are taken as averages for the units VKA1 and VKA3. This is not an optimal solution since there is no single condensing temperature for CC2 but it is as close as possible. The total COP for CC2 remains fairly constant with changing condensing temperature and also with changing ambient temperature as shown in figure 6.31. The inclination of the graphs for TR4 and TR5 are very similar and all systems show decreasing total COP with increasing ambient and condensing temperatures. One possible reason why the total COP of CC2 is more constant than those of the trans-critical CO2 systems is that there is an extra temperature difference between the low temperature stage and the medium temperature stage. However, the main reason is likely to be the rather constant condensing temperatures used in the different units of CC2, especially in the important unit VKA3 where the condensing temperature is very stable. The trans-critical systems TR4 and TR5 require the condensing temperature to be raised to a high level in order to have heat recovery but this is not the case with the more stable cascade system CC2.
Figure 6.31: Total COP for LR=3 as a function of ambient temperature for TR4-5 and CC2 (monthly averages).

Figure 6.32 shows the total COP, excluding the influence of system parasites, with and without load ratio correction as a function of condensing temperature for the three systems (based on monthly averages). The LT and MT evaporation temperatures are also included in the figure.
6.5 Heat recovery

The use of heat recovery can be a very effective way of reducing the energy consumption of a refrigeration system. Even though this increases the efficiency of the entire refrigeration plant, the efficiency of the refrigeration cycle itself actually decreases due to the high condensing temperatures needed for heat recovery. The three refrigeration plants in this study all have heat recovery systems installed and in this chapter, an attempt is made at investigating how these systems are controlled.

Figure 6.33 shows outdoor temperature and outlet temperatures for the high stage compressors, heat recovery system and gas cooler as well as the discharge pressure during six days for TR4 at the end of August. The temperature variation between day and night is clearly visible and the drop in heat recovery outlet temperature (on the refrigerant side) coincides with an increase of the discharge pressure to trans-critical levels. Low temperatures at the heat recovery outlet (gas cooler inlet) show that the heat recovery system is in operation and the pressure increase shows how the system is controlled. The sharp increase in pressure appears to start in the morning at about seven a clock when the outdoor temperature has reached its lowest point and the demand for heat recovery is at its peak. The pressure is raised to about 75-76 bar and kept at this level for a certain period of time before decreasing to sub-critical levels. This decrease seems to continue until about three a clock in the morning when it stabilizes. There seems to be a lower limit for the discharge pressure as well at about 50 bars. This level is visible three times in Figure 6.33 and coincides with the lowest values of ambient temperature (about 10°C). The compressor discharge temperature (heat recovery inlet temperature) is a function of discharge pressure in the calculations and therefore follows the pressure graph almost exactly. There is a small temperature difference between ambient and gas cooler outlet temperature of a few degrees.

![Figure 6.33: Outdoor temperature, high stage discharge pressure and outlet temperature from high stage compressors, heat recovery system (on the refrigerant side) and gas cooler for TR4.](image-url)
Figure 6.34 shows the same diagram for TR5 but for the entire month. When the system is operating in trans-critical mode at the end of the month, the difference between the compressor discharge temperature (heat recovery inlet temperature) and the heat recovery outlet temperature is about 20 degrees (just like for TR4 in figure 6.35) which indicates that heat is being recovered. However, for the beginning of the month during sub-critical conditions, the temperature difference is very small. This is true for TR5 for the entire test period but for TR4, a constant temperature difference of about 20-25 °C has been observed.

It is not clear what the reason for the temperature difference in TR4 is. It could be due to insufficient insulation on the pipes between the high stage compressors and the heat recovery unit resulting in heat loss to the ambient. It could also be the result of a constant flow in the water circuit of the heat recovery system even when no heat recovery is needed.

Even though the pressure gives a good indication about the use of the heat recovery system, other parameters such as the gas cooler outlet temperature can also be useful for the analysis. Figure 6.36 shows the condensing temperatures and gas cooler outlet temperature as functions of high stage discharge pressure for both sub- and trans-critical operation for each month. When the pressure exceeds 73.84 bar the system is in trans-critical mode and all values obtained will be above the critical point. This means that temperature and pressure will be independent of each other and that no values of condensing temperature are available in the data from LDS. For this reason, condensing temperatures after 73 bars have been extrapolated by using the Excel plot option to view the equations for the graphs until 73 bars in the diagrams. The extrapolated values were then used in figure 6.36, figure 6.37 and for all plots containing condensing temperatures for TR4 and TR5.
From Figure 6.36, it can be seen that the gas cooler outlet temperature is slightly lower than the condensing temperature for sub-critical operation but when the system becomes trans-critical, the gas cooler outlet temperatures start to drop indicating heat recovery operation. It is noticeable how July differs from the other months as the gas cooler outlet temperature to a large extent continues to follow the extrapolated condensing temperature. This demonstrates that the need for heat recovery was small in July and the lowest value for gas cooler outlet temperature in trans-critical mode is 15°C. In September, there are even values below 10 degrees and the measured temperatures have clearly moved away from the condensing temperature. Some of the heat recovery takes place during sub-critical conditions for each month except July. The pressure also reaches a higher level for July indicating an increase in cooling load due to higher ambient temperature.
Figure 6.36: Condensation temperature and gas cooler outlet temperature for sub- and trans-critical operation as function of discharge pressure for TR4 from June to September.

Figure 6.37 shows the equivalent plots for TR5 for all months. Some differences between TR4 and TR5 can be observed immediately. There seems to be no trans-critical operation in June and the gas cooler outlet temperature is close to the condensing temperature which suggests that there is no heat recovery is sub-critical mode either.

Figure 6.37: Condensation temperature and gas cooler outlet temperature for sub- and trans-critical operation as function of discharge pressure for TR5 from June to September.
For July, the trans-critical operation has not reduced the gas cooler outlet temperature even though there seems to be some temperature drop in sub-critical mode, indicating no or very little heat recovery. For August and September, there appears to be a substantial amount of heat recovery and this starts already at very low pressures.

The fact that TR5 starts getting low values of gas cooler outlet temperature at low pressures and that these temperatures are fairly constant around 10 degrees as the pressure increases would suggest that there is a difference in how the two trans-critical systems are controlled. When heat recovery is required in TR4, the pressure is raised and the heat recovery system starts working when the pressure is close to trans-critical levels. For TR5 however, the heat recovery can start working already at low pressures and the gas cooler outlet temperature will have a much narrower span than TR4, about ± 5, centered around 10°C.

Figure 6.38 shows the average temperature difference per month over the heat recovery system for CC2, TR4 and TR5 and the corresponding ambient temperatures. As figure 6.34 and 6.35 showed, the temperature difference over the heat recovery is higher for TR4 than for TR5. The constant temperature difference of about 20-25 °C for TR4 can also be seen in Figure 6.37. The temperature difference is highest during June and September and lowest in August, showing that there is more heat recovery when the ambient temperatures are low. There is almost no temperature difference at all for the coolant inlet and outlet to the heat recovery of CC2, indicating that heat recovery is not used during the four months of the study.

![Figure 6.38: Temperature difference over the heat recovery systems (on the refrigerant side) for CC2, TR4 and TR5 and corresponding ambient temperatures based on monthly averages.](image-url)
6.6 Parasites

The influence of parasites in a refrigeration system can be substantial, especially for large systems that require pumps for refrigerant circulation. The CO2 cycles TR4 and TR5 have no need for pumps which would suggest that they are less affected. Dry- and gas coolers will often require even more energy than the pumps of a system. This section aims to investigate how much of the total system energy consumption that is lost to parasites and what kind of influence these parasites have on system performance.

6.6.1 Parasite energy consumption

Figure 6.39a&b shows the distribution of energy consumption for the different system components of CC2 for August 2010. In the left diagram, only energy consumption of parasites and compressors are included but in the diagram to the right, the energy requirement of the low and medium temperature cabinets due to lighting, defrosting etc has also been included. Evidently this energy amounts to as much as 45 % of the total. However, cabinet energy measurements has not been available for TR4 and TR5 in this study and therefore, the left diagram will be used for comparisons between cycles.

![Pie charts showing energy consumption distribution](image)

Figure 6.39-a&b: Distribution of energy consumption for CC2 with and without cabinet energy consumption in August 2010.

It is clear from Figure 6.39a that the MT-compressors have the largest energy consumption at 47 % of the total followed by LT-compressors at 34 % and then the dry cooler fans at nine percent. The main brine pump P1 (Figure 4.2) required five percent of the total energy consumption of the system in August while the other brine pumps P2 (P2_A & P2_B) reached a demand of one percent. The pumps P3 and P4 that create the split flow in the coolant loop reached similar energy demands of about two percent each. The pump P4 that
provides the flow of coolant to the heat recovery system is not included in the calculations of MT-energy consumption and coefficient of performance but has still been included in this analysis. The fairly high energy consumption of two percent of the total for this pump gives an indication of the operation of the CC2 heat recovery system. Table 6-2 contains the same values that are displayed in figure 6.39a but also includes the other three months of the study period as well as averages for this period. Figure 6.40 shows the LT- and MT compressor power consumption and the energy consumption of the gas coolers for TR4 and TR5. It can be seen that TR4 has a higher percentage of energy consumption for MT compressors than TR5 and a lower percentage for LT compressor energy consumption. The TR5 gas cooler fans have higher percentile energy consumption than those of TR4.

![Figure 6.40](image)

**Figure 6.40:** Distribution of energy consumption over LT and MT compressors and gas cooler fans for TR4 and TR5 in August 2010.

According to table 6-2, the energy consumption of the parasites in system CC2 is between 16 and 20 percent of the total. For TR5, it is between seven and nine percent and even less, one percent, for TR4. The negative value for the gas cooler of TR4 implies that the calculation method used to obtain the values is not very accurate and that the results may vary a few percent from the actual value. Therefore, the calculated percentages should be considered only as guideline values, showing the differences between the systems rather than exact values.
Table 6-2: Distribution of energy consumption for June to September for CC2, TR4 and TR5.

<table>
<thead>
<tr>
<th>System</th>
<th>component</th>
<th>June [%]</th>
<th>July [%]</th>
<th>August [%]</th>
<th>September [%]</th>
<th>Average:</th>
</tr>
</thead>
<tbody>
<tr>
<td>CC2:</td>
<td>LT_compr</td>
<td>38</td>
<td>30</td>
<td>34</td>
<td>34</td>
<td>34</td>
</tr>
<tr>
<td></td>
<td>MT_compr</td>
<td>46</td>
<td>55</td>
<td>47</td>
<td>46</td>
<td>48,5</td>
</tr>
<tr>
<td></td>
<td>Pump P1</td>
<td>6</td>
<td>4</td>
<td>5</td>
<td>5</td>
<td>5</td>
</tr>
<tr>
<td></td>
<td>Pump P2</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td></td>
<td>Pump P3</td>
<td>2</td>
<td>2</td>
<td>2</td>
<td>3</td>
<td>2,25</td>
</tr>
<tr>
<td></td>
<td>Pump P4</td>
<td>2</td>
<td>1</td>
<td>2</td>
<td>2</td>
<td>1,75</td>
</tr>
<tr>
<td></td>
<td>Pump P5</td>
<td>&lt;0,1</td>
<td>&lt;0,1</td>
<td>&lt;0,1</td>
<td>&lt;0,1</td>
<td>&lt;0,1</td>
</tr>
<tr>
<td></td>
<td>Pump P6</td>
<td>&lt;0,1</td>
<td>&lt;0,1</td>
<td>&lt;0,1</td>
<td>&lt;0,1</td>
<td>&lt;0,1</td>
</tr>
<tr>
<td></td>
<td>Pump P7</td>
<td>&lt;0,1</td>
<td>0,1</td>
<td>&lt;0,2</td>
<td>0,1</td>
<td>&lt;0,125</td>
</tr>
<tr>
<td></td>
<td>Pump P8</td>
<td>&lt;0,1</td>
<td>&lt;0,1</td>
<td>&lt;0,2</td>
<td>&lt;0,1</td>
<td>&lt;0,125</td>
</tr>
<tr>
<td></td>
<td>DC_fans</td>
<td>5</td>
<td>7</td>
<td>9</td>
<td>9</td>
<td>7,5</td>
</tr>
<tr>
<td></td>
<td>Sum_par.:</td>
<td>&lt;16,4</td>
<td>&lt;15,4</td>
<td>&lt;19,6</td>
<td>&lt;20,4</td>
<td>&lt;17,95</td>
</tr>
<tr>
<td>TR4:</td>
<td>LT_compr</td>
<td>16</td>
<td>13</td>
<td>15</td>
<td>17</td>
<td>15,25</td>
</tr>
<tr>
<td></td>
<td>MT_compr</td>
<td>83</td>
<td>86</td>
<td>84</td>
<td>82</td>
<td>83,75</td>
</tr>
<tr>
<td></td>
<td>GC_fans</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>-1</td>
<td>0,5</td>
</tr>
<tr>
<td></td>
<td>Sum_par:</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>-1</td>
<td>0,5</td>
</tr>
<tr>
<td>TR5:</td>
<td>LT_compr</td>
<td>21</td>
<td>18</td>
<td>21</td>
<td>23</td>
<td>20,75</td>
</tr>
<tr>
<td></td>
<td>MT_compr</td>
<td>70</td>
<td>75</td>
<td>72</td>
<td>68</td>
<td>71,25</td>
</tr>
<tr>
<td></td>
<td>GC_fans</td>
<td>9</td>
<td>7</td>
<td>7</td>
<td>9</td>
<td>8</td>
</tr>
<tr>
<td></td>
<td>Sum_par:</td>
<td>9</td>
<td>7</td>
<td>7</td>
<td>9</td>
<td>8</td>
</tr>
</tbody>
</table>

6.6.2 Parasite influence on system performance

The energy consumption with and without parasites included will result in different values of total COP as displayed in Figure 6.41. This diagram shows how the performance of the different systems is affected by the presence of parasites. It clearly shows that for TR4, the smallest system, the parasites will have the least impact on system performance. For TR5, the parasite influence is more noticeable which can also be seen in table 6-2 where the gas cooler requires about 8 percent of the total energy consumption. For both TR4 and TR5, the only parasites are the gas coolers and as you might expect, CC2, with all the pumps and the dry cooler fans is subject to the largest parasite influence on system performance.

The difference in condensing temperatures between the systems can also be noted in Figure 6.41. TR4 has higher a few degrees higher condensing temperatures than TR5. Since the units of system CC2 all have different condensing temperatures, the values used in the diagram for CC2 are based on averages for the most important medium temperature units, VKA1 and VKA3.
Based on four month averages, the total COP is 2.1 for CC2, 2.8 for TR4 and 3.0 for TR5 when excluding the influence of parasites. This can be compared to the actual values of 1.8, 2.7 and 2.8 respectively for CC2, TR4 and TR5. When using load ratio correction (LR=3) and excluding parasites, the values of total COP did not change much except for TR5 where the total COP increased from 3.0 to 3.2.
7 Validation of specific parameters

When comparing the different systems, it is important to be aware of the differences in the basic calculations and the method of data collection. The two trans-critical CO2 systems are based on the same method for data acquisition, but the cascade system CC2 is not. A number of assumptions, simplifications and compromises have been used in order to obtain results and to make comparisons. This chapter takes into account the differences in calculations and measurements and makes an attempt to validate the results.

7.1 Differences in data acquisition.

Two different programs are used for the data acquisition, IWMAC and LDS. The data collected using LDS has a constant time interval of 2 minutes which is than filtered to every 10 minutes. These data are measured at the exact time that they correspond to in the data series except for energy and power measurements which are based on averages for the time intervals. This makes the input data from LDS very reliable and the results as well.

For the cascade system, IWMAC is used for the data collection. A big problem with using this program is that data for a parameter is only available when there is a change in that parameter. For example, a data series for a compressor signal can have as few as two values, one and zero. This means that the compressor is on until a certain time when it turns off and remains off for the rest of the month. This results in a major problem with the calculation since there is no constant time interval for the input data. This problem was solved by using software called Python to make 10 minute averages of the input data that have time gaps of less than 10 minutes. When the time gap between two sets of data is larger than 10 minutes, like in the case with the compressor signal, the previous value is used every 10 minutes. This is an important difference between the CO2 systems and the cascade system and one should keep in mind that in one case, it is actual measurements that are shown and in the other, it is averages.

The problems with the data acquisition due to malfunctioning logging computers have resulted in gaps in the data series. Sometimes, the gaps are only a few minutes and sometimes several days, and this will have some effect on the results. The impact is much smaller when using monthly averages for the comparison, especially if the gaps are somewhat evenly spread over time, but some parameters such as outdoor temperature and condensing temperature will still be affected. For example, data is missing for the second half of September for TR5 which means that the average condensing and outdoor temperatures will be higher than the real values. This problem occurs for CC2 as well, since data is only available for the first eight days of September due to a problem with one of the system signals.

7.2 Measured and Calculated Energy

There are differences in how the results for power consumption, and thereby also the COP, have been obtained for the different systems. For TR4 and TR5, the power consumption for the LT stage is equal to the compressor power consumption. This is calculated using temperature measurements, signals from the compressors and compressor manufacturer data. The same method of calculation is used for the MT stage. The power consumption in kW for LT and MT is also measured directly and available in LDS with a fifteen minute interval. However, these measurements are most likely based on the measurement of energy
consumption for each of the fifteen minute periods and therefore only show the average value in kW for each period. To get more realistic values, the energy consumption for the entire month could be used and divided by the number of hours that data is available but this presents a problem since the number of hours missing due to gaps in the data collection must be counted by hand. The smaller gaps of just a few minutes will be difficult to spot and they will eventually add up to numbers that will influence the results. Therefore, the measured power consumption for the trans-critical systems is not very accurate which is why the calculated values have been used whenever possible, for example for the LT stage results. For the MT stage, the measured values of power consumption must be used since they contain the power consumption of the gas coolers. Figure 7.1 shows the measured and calculated values of power consumption for TR4.

![Figure 7.1: Measured and calculated power consumption for TR4 based on monthly averages.](image)

The very small difference between the calculated and measured values for MT is the power consumption of the gas coolers but for the LT-stage, there is a bigger difference. The calculated values are approximately 20 percent larger than the measured and this difference also influence the total result. Figure 7.2 shows the measured and calculated values for TR5. The calculated power consumption for the LT stage is about 7 percent higher than the measured values. The difference in the MT-stage shows that a larger part of the energy consumption is used in the gas coolers for TR5 than for TR4. This is also described in the parasite analysis in chapter 6.6.
The comparison of calculated and measured LT power consumption for TR4 and TR5 has shown that the calculated values were about 20 percent and 7 percent higher than the measured values respectively. Although the measured data is not extremely accurate, the differences suggest that there is an overestimation of the compressor power in the calculations. This could possibly relate to differences between the compressor manufacturer data and the actual performance of the compressors. Unfortunately, a similar comparison is not possible to make on the MT-side since the gas cooler fans are included in the measured values of power consumption. Installation of a separate energy measurement of the gas coolers would solve this problem.

For the cascade system, a combination of measured and calculated values is also used to obtain results. The power consumption for the LT stage is the sum of the measured compressor power for KS4 and for pump P2, and the calculated value of compressor power consumption in VKA3 that is used to support unit KS4. For the MT-stage, compressor power is calculated for the VKA-units based on measurements of temperature, compressor speed and on compressor manufacturer data. This is then added to the measured power consumption of the dry cooler and the pumps.

One of the main reasons for using calculated values for the different systems, even when measured values are available, is that it allows comparisons to be made with other supermarkets included in the larger, overall project where this is the standard method of evaluating power consumption.
7.3 Refrigerant liquid temperature after gas bypass heat exchanger

For the trans-critical CO2 systems, the temperature of the refrigerant after the heat exchanger in the liquid line (the temperature $T_i$ in figure 4.5) was assumed in the calculations even though the measured values were available. With a rather constant pressure of 34 bars in the receiver, a high value of this temperature will end up very close to the saturation temperature. This resulted in some problems with the calculations using Refprop and in order to avoid this, a value of $-1^\circ C$ was chosen for all calculations. Figure 7.3 show the monthly averages for the measured values of this temperature.

As seen in the figure, the values for TR4 are lower than -1. However, by looking in a p-h diagram, it is clear that a difference of one or two degrees for this temperature will not have a significant impact on the results. In Figure 4.6, point “i” denotes the temperature of the refrigerant liquid after the heat exchanger. If this temperature was one degree lower, this point would move slightly to the left, resulting in a minor increase in enthalpy difference over the evaporator, and in turn a very small increase in cooling capacity. The main purpose of the heat exchanger is not to increase the cooling capacity this way, but to ensure that the temperature is low enough to avoid vapor bubbles in the evaporator. This means that the assumption of a temperature of $-1^\circ C$ is valid for the analysis.

![Temperature of the refrigerant liquid exiting the heat exchanger for TR4 and TR5 based on monthly averages.](image-url)
7.4 High cooling capacity for CC2.

In the overall system comparison, the results were shown for low and medium temperature stages and for the total system. The value of MT cooling capacity was very high for July compared to June, which also affects the value for total cooling capacity of the system. This large increase in cooling capacity in just one month did not seem reasonable. Looking at Figure 6.2, it is clear that the high value of MT cooling capacity is a result of the high cooling capacity for VKA1. In order to justify the high levels, VKA1 was looked at further in detail. By examining the main parameters used in the calculations; enthalpy difference over the evaporator and mass flow, it was clear that the average mass flow was higher in July than for the other months while the enthalpy difference remained rather constant as shown in Figure 7.4.

The mass flow depends on how the compressors are controlled and this is shown by the average compressor signal. If a compressor is on all the time, the signal will have a value of one, and if it is off, a value of zero. It is clear that compressor 1 has values around 0.65-0.75 and is rather stable while compressor 2 is almost never in operation in June but show a much higher value in July. This is the reason for the large increase in MT and total cooling capacity shown in the results. It does not explain why there is a sudden need for this increase but since the compressor signals are measured values, the high values of cooling capacity can be accepted.
8 Discussion

Field measurements have been carried out on three supermarket refrigeration systems, two trans-critical DX CO2 systems and one R404A/C02 cascade system. The results have shown that the cascade system CC2 has a significantly lower COP compared to the trans-critical CO2 systems. There are several reasons for the low values of COP for this system. The condensing temperatures for the main units VKA1 and VKA3 in CC2 are kept at a much higher level than those of TR4 and TR5. For instance, the difference in condensing temperature between TR5 and VKA3 is more than 10°C. VKA3 has the highest condensation temperature since it is the unit located closest to the dry-coolers. At the same time, the evaporation temperature of VKA3 is twice as low compared to VKA1 and VKA2 which are at the same level as the MT-stage evaporation temperatures of TR4 and TR5. System CC2 also has the lowest LT-stage evaporation temperature (for unit KS4) at about -35°C compared to about -32°C for the trans-critical CO2 systems. High condensing temperature and low evaporation temperature require more compressor work which puts CC2 at a disadvantage compared to the other systems.

The low evaporation temperature of VKA3 is mainly a result of the design and control of the system. It is necessary to have a very low evaporation temperature because VKA3 must remove heat from the condenser in KS4. The inlet temperature to the KS4 condenser must be low enough to ensure that the temperature at the condenser outlet is not too high, since this flow will mix with the flow from the other VKA-units before being supplied to the MT cabinets. The inlet temperature to the chillers was found to be rather high for a supermarket the size of CC2 and even though the temperature levels in the cabinets seemed sufficiently low, there may not be much room for improving the system performance by increasing the evaporation temperature of VKA3.

The results have shown that as much as half of the produced cooling capacity in unit VKA3 is used for removing heat from the condenser of unit KS4. VKA3 also consumes twice the amount of energy compared to VKA1. At the same time, unit VKA2 is almost never in operation except during the two warmest months July and August when it contributes with about 11 and 7 percent of the total cooling capacity respectively. Otherwise, it seems that the only purpose of this unit is to serve as a backup for VKA1.

System CC2 had a four month average COP of 1.2 for the low temperature side and about 2.1 for the medium temperature side, resulting in an average total COP of about 1.8. For system TR4, the average COP for the LT stage was about 1.6 for the four months of the study and for the MT stage, the average COP was about 3.4 which resulted in a total COP of about 2.7. For system TR5, the LT-side had an average COP of about 1.7 compared to about 3.8 for the MT-side for the four months. The average total COP for this system was about 2.8, which is also the best value for the three systems. The maximum total COP for TR5 was about 2.9 but TR4 had the highest value of total COP at about 3.0 in September. When using the equation for load ratio correction to equalize the operating conditions of the three systems, CC2 had a four month average total COP of 1.7 while TR4 and TR5 had values of 2.7 and 2.9 respectively.

The average defrosting time per month was compared for the LT and MT cabinets of system TR4 and TR5 in order to investigate the influence of frost accumulation due to increased humidity. Even though no obvious relation was found between defrosting time and ambient temperature and humidity, it was clear that the medium temperature cabinets require 3-4
times more defrosting time than the low temperature cabinets. One possible reason for this
difference could be that the medium temperature cabinets don’t use electric defrost but
rather the warmth of the air in the store area to heat the evaporator when the refrigerant flow
is stopped.

The amount of internal superheat for the cabinets was also investigated for the three
systems. The highest values were observed for the LT cabinets of system CC2 at about 13K
followed by the LT cabinets of TR5 at 9-10K. The other LT and MT cabinets had an internal
superheat of about 6-7K. No clear connection between ambient temperature and internal
superheat could be observed when using average values but in order to draw any
conclusions about how the cabinets are controlled, a more thorough investigation of the
individual cabinets is needed.

The different parasites in the systems were investigated in terms of energy consumption and
influence on system performance. The result showed that for CC2, about 18 percent of the
total energy consumption was lost to parasites like pumps and dry cooler fans. This number
was about eight percent for TR5 and one percent for TR4. This clearly shows one of the
biggest advantages with CO2 systems over cascade solutions; the only parasites are the gas
cooler fans. The pressure drop in the pipes is very small when using CO2 which eliminates
the need for pumps to circulate the refrigerant. The coefficient of performance was calculated
with and without parasites and when comparing the values, the cascade system showed the
largest decrease in efficiency followed by TR5 while the difference was much less for TR4.
The total COP without parasites was found to be 2.1 for CC2, 2.8 for TR4 and 3.0 for TR5.
When using load ratio correction (LR=3), these values did not change much except for TR5
where the total COP reached 3.2.

The operation of the heat recovery for the two trans-critical CO2 systems was investigated by
comparing temperature measurements. When comparing the condenser/gas cooler outlet
temperature during sub- and trans-critical operation and the condensing temperature
(extrapolated values for trans-critical state), some differences were found between TR4 and
TR5. For TR4, the gas cooler outlet temperature followed the condensing temperature until
the pressure started to get above 70 bar where it dropped; indicating the use of heat
recovery. For TR5, low values of condenser/gas cooler outlet temperature were found
already at low compressor discharge pressures, suggesting that the heat recovery is
operating for a wider refrigerant temperature range. However, when looking at the gas cooler
fans of TR4 and TR5, it was clear that TR5 was operating between about 90-100 % of full
capacity which is about 10 % more than for TR4 and this could also explain the lower gas
cooler outlet temperatures at low pressures.

A constant refrigerant temperature difference of about 20K was found over the heat recovery
of TR4, much higher than for TR5 but the reason for this is not known. It could be because of
poor insulation of the pipes in TR4 or because there is a constant flow on the water side of
the heat recovery system. For CC2, the temperature difference of the coolant over the heat
recovery was found to be almost 0K, indicating that there was no heat recovery for the time
period of the study. However, when investigating the parasites, it was found that the pump
P4 required as much as 1-2 percent of the total energy consumption. Pump P4 is not a very
large pump and the large power consumption indicates that there is a substantial flow of
coolant to the heat recovery system, even if heat is not being recovered. In order to safely
draw conclusions about the use of the heat recovery in the three systems, this study should
be extended to also cover the winter months when heat recovery in needed.
The inherent disadvantages with cascade systems are the additional temperature levels due to the brine- and coolant circuits and the need for circulation pumps. For every temperature level that is added to a system, there will be losses when transporting heat between levels, resulting in a lower efficiency of the system. In the trans-critical CO2 systems, no pumps are needed and the use of floating condensation to take advantage of low ambient temperature makes these systems much more favorable, especially in cold climates.
9 Conclusions

The use of CO2 as a refrigerant can be very beneficial because of its good thermodynamic properties and low environmental impact. Three supermarket refrigeration systems were investigated by collecting measured data and making calculations of cooling capacity and coefficient of performance.

It was found that the cascade system CC2 had much lower values of COP than the transcritical systems. This is mainly because of the structure of the system where unit VKA3 has to support unit KS4 and part of the MT-stage while unit VKA2 is rarely in operation, and because of the extra temperature levels resulting from the use of the brine- and coolant circuits. The great number of pumps used in the system also reduces the COP as shown in the investigation of parasites. The higher values of COP for the trans-critical CO2-systems can be attributed to the use of floating condensation and to the fact that no pumps are required for the circulation.

The use of a heat recovery system reduces the COP of the refrigeration system. This is very obvious for trans-critical CO2 systems where the condensation temperature/pressure must be raised to a level that is appropriate for heat recovery and this increases the power consumption of the compressors, reducing the COP. Since only the summer period was included in this thesis, it is difficult to draw conclusions about the heat recovery.

Suggestions for future improvements may include putting energy meters on the gas cooler fans for TR4 and TR5 in order to separate fan power from the power consumption of the high stage compressors. Putting glass doors and lids on the display cabinets of TR4 is also a good way of saving energy and increasing the system performance. For system CC2, if possible, it would be very advantageous to let VKA3 support the low temperature stage in unit KS4 only, while letting unit VKA1 and VKA2 support the medium temperature stage.

In order to see how the three systems operate in colder ambient temperatures, this study should be extended to also cover the winter months.
10 References


[IWM10] IWMAC homepage (2010), Information and access to Iwmac operation center with field measurements, available at www.iwmac.se, last accessed 2011-02-10


