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Master Thesis

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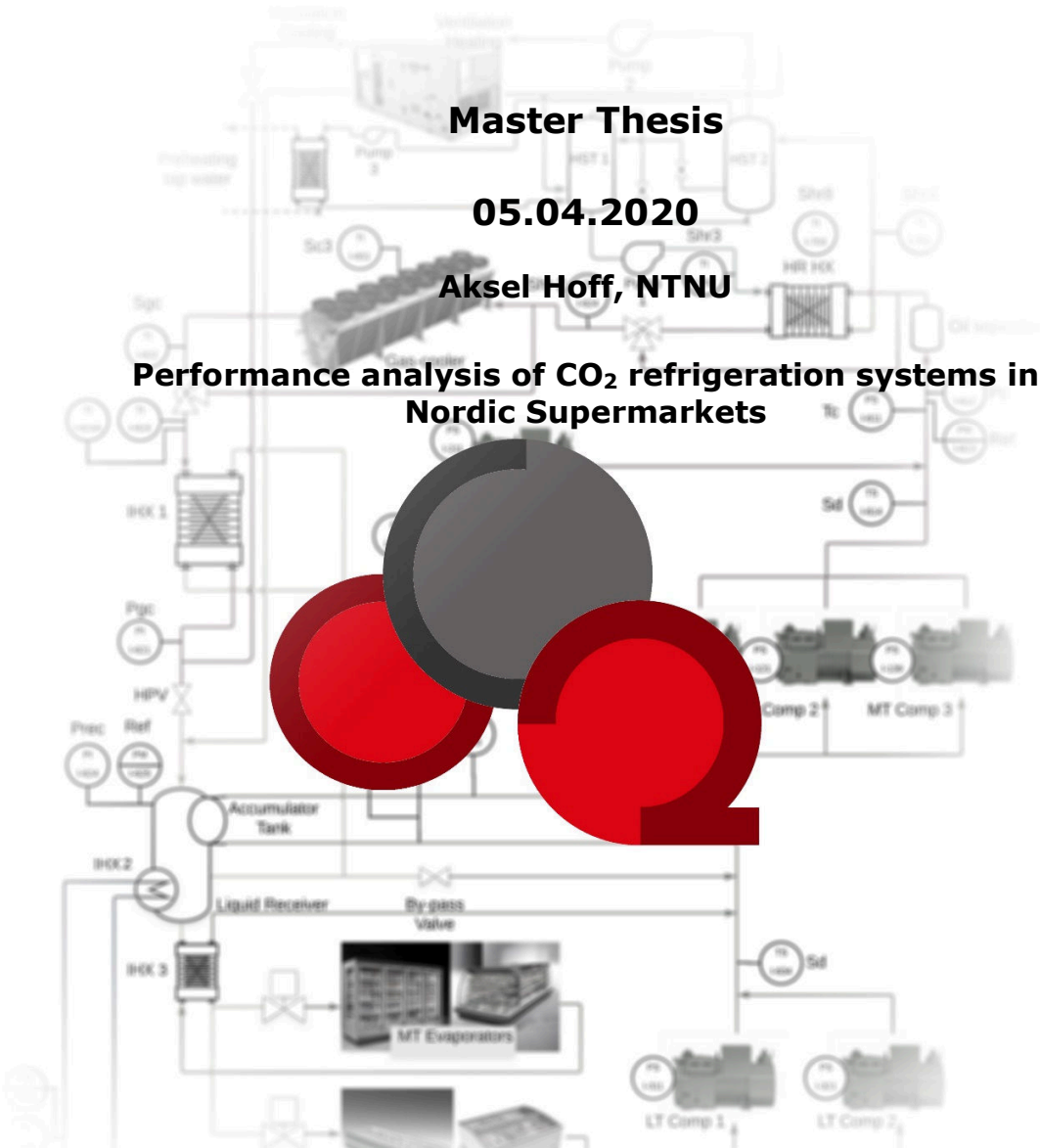
Performance analysis of CO_2 refrigeration systems in Nordic Supermarkets



Study course: Energy and Process Engineering
Academic supervisor: Prof. Armin Hafner (NTNU)

1 Preface

The current master thesis is carried out at the Department of Energy and Process Engineering at the Norwegian University of Science and Technology. Due to the prohibition of the common chlorine and fluorine containing refrigerants, the food retail industry uses natural refrigerants such as CO_2.This thesis is a contribution to a more thorough documentation of the performance of CO_2 as a refrigerant in a Nordic climate.

First and foremost, I will thank my supervisor, Professor Armin Hafner, for his patient guidance and help during the master studies. His comprehensive knowledge concerning refrigeration technology has been of great support. Additional thanks to Erik Hoksrød for his helpful guidance regarding the explanation of the web-monitoring systems.

I am grateful to my parents for continuous support and encouragement throughout the master period. A special thanks to my brother, Karsten Wiik, for his artistic contribution to the front page.

Aksel Hoff
Stavanger, 2020

Abstract

Hydrofluorocarbons are being replaced by natural refrigerants in the food retail industry. Research on replacing non-natural refrigerants with CO_2 is ongoing and more and more supermarkets in Northern-Europe have refrigeration systems running on CO_2.

The consensus is that CO_2 is a promising natural refrigerant, and research shows that the CO_2-applying refrigeration systems have competitive energy performance during lower ambient temperatures, 15°C.

This thesis is intended to perform a comprehensive evaluation of three transcritical CO_2 booster refrigeration systems in Nordic climate using real data obtained from web-monitor systems. The data is collected and structured using Microsoft Windows Excel. The calculations are based on thermodynamic theory, and an add-inn provided by the institute of Energy and Process Engineering for finding theoretical values of dependent variables such as pressure and specific enthalpy.

The main finding of the evaluation of the supermarkets are that they perform energy efficient during colder ambient temperatures, while the most advanced technical system Spar Snarøya has the best overall performance. The performances of the systems were as expected lower during warmer climate, the systems with parallel compression performs best. For CO_2 to be a leading refrigerant on the market it is vital that the refrigeration systems have efficient ways of utilizing the excess heat of high-pressure CO_2.

Sammendrag

Hydrofluorkarboner erstattes med naturlige kjølemedier i matvareindustrien. Forskning på å erstatte ikke-naturlige kjølemedier med CO_2 pågår. Stadig flere supermarkeder i Nord-Europa har kjølesystemer basert på CO_2.

Faglig konsensus er at CO_2 er et lovende naturlig kjølemiddel, og forskning viser at kjølesystemer som anvender CO_2, har konkurransedyktig energiytelse når omgivelsestemperaturen er lavere enn 15 ^{0}C.

Formålet med denne oppgaven er å utføre en grundig evaluering av tre transkritiske CO_2 «booster»-kjølesystemer i nordisk klima ved å bruke virkelige data som er hentet ut av web-baserte loggesystemer. Dataene er samlet og strukturert ved å anvende Microsoft Windows Excel. Beregningene er basert på termodynamisk teori samt en tilleggsmodul skaffet av Institutt for energi og prosessteknikk for å finne teoretiske verdier som f.eks. trykk og spesifikk entalpi.

Hovedfunnet i vurdering og sammenligning av kjølesystemene i de tre supermarkedene, er at de opererer energieffektivt gjennom perioder med relativt lav utetemperatur, mens det avanserte tekniske systemet, Spar Snarøya, har den beste totale ytelsen. Det var forventet at ytelsen til systemene er lavere gjennom varmere perioder. Systemene med parallell kompresjon yter best ved forholdsvis varme utetemperaturer, over 15 ^{0}C. For at CO_2 skal være et foretrukket kjølemedium på markedet, er det avgjørende at kjølesystemene har effektive måter å bruke overskuddsvarmen fra CO_2 under høyt trykk.

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Abbreviations

AC	Air Conditioning
AHU	Air Handling Unit
BPV	By-pass Valve
CFC	Chlorofluorocarbon
COP	Coefficient of Performance
DC	Dry Cooler
DX	Direct Expansion
EM	Energy Meter
EVA	Evaporator
GC	Gas Cooler
GWP	Global Warming Potential
HCFC	Hydrochlorofluorocarbon
HFC	Hydrofluorocarbon
HFO	Hydrofluoroolefin
HPV	High-pressure Valve
HR HX	Heat Recovery Heat Exchanger
HPS	High Pressure Section
HRC	Heat Recovery Cycle
HSC	Side 25 xx
HVAC	Heat Ventilation and Cooling
HST	Heat Storage Tank
IHX	Internal Heat Exchanger
IPS	Intermediate Pressure Section
ITS	Intermediate Temperature Section
LT	Low Temperature
LTS	Low Temperature Section
LPS	Low Pressure Section
LR	Liquid Receiver
MT	Medium Temperature
MTS	Medium Temperature Section
ODP	Ozone Depletion Potential
PC	Parallel Compressor
PHE	Plate Heat Exchanger
PI	Piping and Instrumentation
TEWI	Total Equivalent Warming Impact
VCB	Ventilation Cooling Battery
VHB	Ventilation Heating Battery
VSD	Variable Speed Drive

1 Introduction

An ongoing change in the food retail sector is reflecting a change going on in the global energy industry; replacing harmful climate and environmental damaging substances with environmentally friendly and renewable resources. While replacing CO_2-emitting processes in the oil and gas-industry with greener alternatives is the key on a global energy industry scale, utilizing CO_2 for refrigeration is contributing to leave smaller carbon footprints in the food retail industry.

A rapid growing number of supermarkets in Europe are using CO_2-only refrigeration systems. The main reason for replacing the common chlorine and fluorine containing refrigerant is the F-gas regulation. Most countries have ratified a protocol for replacing conventional hydrochlorofluorocarbons (HCFCs) and chlorofluorocarbons (CFCs) with natural refrigerant. Research on applying CO_2, or R744 which is its name within refrigeration industry terminology, in cascade and transcritical refrigeration systems have been performed since the late 1980s.

A worldwide ban on ozone-depleting substances creates a need for a refrigerant substitute in heating, cooling, and freezing appliances. Known substitute working fluids are natural- and synthetic refrigerants. Natural refrigerants include CO_2 (R744), ammonia (NH_3; R717), isobutane (methylpropane; R600a) and propane (R290). Synthetic refrigerants include hydrofluorocarbons (HFC) and hydrofluoroolefins (HFO). Some of the synthetic refrigerants have a very high global warming potential (GWP). As an example, CHF_3 (HFC23) has an GWP value of 14800 compared to the GWP value of 1 for CO_2.

CO_2 stands out as a refrigerant with low toxicity, does not harm the ozone layer and has low GWP. The concern of CO_2 as a greenhouse gas is due to the massive emission from other sources. As refrigerant CO_2 can be processed from air and does not contribute to the greenhouse effect. In addition, the direct effect of CO_2 emissions form refrigeration systems and the indirect effect CO_2 emissions form the generation of energy to run the refrigeration system, combined have a low Total Equivalent Warming Impact (TEWI).

Sustainable development can be defined as "development that meets the need of the present without compromising the ability of future generations to meet their own needs" [2]. Sustainable use of energy is part of the report. Sustainable development has during the 21st century become a well-known term together with climate change. The terms are linked as electrical energy production and transport are partly based on combustion and petroleum (i.e. oil and gas) and thereby generate CO_2-emissions, which cause climate change.

Devices for heating, cooling and freezing use energy for their operation. In that sense, it is important that the devices work as efficient as possible. It turned out that the previous working fluid for refrigeration in these systems were ozone-depleting substance such as hydrochlorofluorocarbons (HCFCs) and

chlorofluorocarbons (CFCs) and halons. Thus, these refrigerants had an environmental cost or negative impact that would be harmful to human health if the protective stratospheric ozone layer become too thin or even disappeared. The Montreal Protocol (1987) [3] laid down principles to phase down production and use of the ozone-depleting substances. In 2016 the Kigali amendment was adopted to phase down production and consumption of hydrofluorocarbons (HFCs) worldwide. HFCs are widely used alternatives to ozone depleting substance.

There are, however, some drawbacks by using CO_2 as refrigerant. One issue is the inherent high working pressure compared to alternative natural and synthetic refrigerants. Higher pressure is more hazardous and components including compressors must be designed to handle this pressure. Higher pressure increases the leak potential. A second issue is the ambient temperature of the refrigeration system and how the ambient temperature relates to COP of the systems. Until 2010 it was assumed that CO_2 refrigeration systems was more efficient than HFC-systems at yearly average mean ambient temperature below 15°C.

Customers do not pay much attention to the energy consumption in supermarkets. It is taken for granted that the premises are comfortable for shopping. Goods to be kept cold refrigerated and frozen good are kept frozen. On general level, it is expected that the operations are carried out as environmentally friendly and sustainable as possible, without any more detailed knowledge about how to do it.

This thesis is a contribution to an increased understanding in how to manage energy more efficient in supermarkets by using CO_2 as refrigerant.

2 Literature review

There are a lot of talk about carbon dioxide, CO_2. Scientist conclude that increased CO_2 in the atmosphere due to burning of fossil fuels and coals leads with great probability to an anthropogenic global warming. Natural occurring CO_2 in the atmosphere is on the other hand certainly vital for life. The roles of CO_2 come in many shapes, and CO_2 could even be utilized as a natural refrigerant to reduce the carbon footprint of an entire industry; the food retail industry. The supermarkets national electricity consumption are approximately 3-5% in Scandinavian countries [4]. The refrigeration systems in the supermarkets consume typically 47% of this energy [5]. By increasing the energy efficiency of refrigeration systems and abide the latest protocol for refrigerant use, the food retail industry moves in an eco-friendly direction. This chapter provides a review of the technological development of the refrigeration systems in the food retail industry and how CO_2 is emerging as a preferable modern refrigerant.

Since Jacob Perkins took out a patent on a vapour-compression refrigeration system using ethyl ether in a closed circuit in 1834, there has been a search for the ideal refrigerant [6]. An ideal refrigerant must be a substance having numerous qualities; stable, non-toxic, non-flammable, miscible with lubricating oil, operate above atmospheric pressure, highly insulating to electricity and easy compressible. The generation of refrigerants replacing the first generation of refrigerants such as H_2O (water), SO_2 (sulphur dioxide), CH_3Cl (methyl chloride), NH_3 (ammonia), hydrocarbons and CO_2, were replaced with compounds meeting the criteria set for a refrigerant for commercializing domestic refrigeration. Various types of halogenic compounds emerged as near ideal refrigerants. Thomas Midgeley selected on a mission from General Motors dichlorodifluoromethane (CCl_2F_2), named R12, as a qualified refrigerant[6]. The refrigerants got names with "R" representing "Refrigerant" and a prefix indicating the order of when the compounds were accepted by ASHRAE[1]. Domestic refrigeration got commercialized eventually and the refrigeration industry grew. R-12 was used in domestic and industrial refrigeration, and other refrigerants such as R-11 (CCl_3F) and R-13 ($CClF_3$) followed into use in different applications. R-22 ($CHClF_2$) replaced R-12 which was later replaced with the azeotrope R-502 and for many years halocarbons covered the refrigerant demand in all ranges of refrigeration and air conditioning. The reason why the halocarbons eventually got banned were the emission of the chlorine containing substance. Chlorine reacts with ozone when being emitted into the atmosphere and causes ozone depletion in the stratosphere. An international set of rules, known as the Montreal Protocol, were established for controlling the use of harmful substances. The halocarbons and other substances with high ozone-depletion potential were phased out and for transitional period replaced with another set of synthetic refrigerants; hydrochlorofluorocarbons (HCFCs). Especially

[1] ASHRAE was formed as the American Society of Heating, Refrigeration and Air-Conditioning Engineers. The society's designation of refrigerants is globally common in refrigeration science

R-134a ($CH_2F.CHF_3$), which has similar properties as R-12, was used in broad extent. The Montreal Protocol stated that a 90% production and consumption reduction should be met in 2015 for all countries, including developing countries[7]. Later, by virtue of the Montreal Amendment, the schedule for complete phase-out of HCFCs was by 2020 in developed countries, while developing countries are following a stepwise reduction until a complete phase-out by 2030 [8].

An energy efficient supermarket in colder climates exploits the excess heat from the refrigeration plant. The heat can be transferred to sub-systems such as heat, ventilation and air conditioning (HVAC), floor heating system and heat storage tanks for multiple means. Such an arrangement where sub-systems exchanges energy with the refrigeration plant is an energy integrated concept. The core of the arrangement is the refrigeration plant itself. The main tasks in a supermarket refrigeration system are to maintain the desired temperatures at the cooling and freezing cabinets. The ratio between the cooling provided and the compression work required is the coefficient of performance (COP) of the refrigeration cycle. Higher COP results in lower running costs of the refrigeration plant. A refrigeration plant which only task is to deliver cooling loads to the cooling and freezing cabinets, is designed to deliver the requested load of cooling at highest COP possible. However, a refrigeration plant in an integrated energy concept also provides the supermarket with heat for internal heating. The inclusion of using excess heat from the refrigeration system changes the prerequisites for the refrigeration system's design point. An essential prerequisite is the refrigerant's heat transferring capacity at temperatures above the desired indoor temperature and floor temperature.

Commercial refrigeration systems in supermarkets use mostly R134a and R404A when hydrofluorocarbons (HFCs) are preferred as refrigerant [9], with R404A the most common refrigerant in Europe (2010) [10]. R404A has a high global warming potential (GWP), and natural refrigerants has been established in the commercial refrigeration industry to use more environmentally friendly substances. The natural refrigerants of interest are R744 and R717, respectively CO_2 and ammonia. Also, hydrocarbons are represented in modern commercial refrigeration systems with R290, propane. All these substances represent different properties and consequently different conditions for the design of a refrigeration plant in a supermarket, with or without heat reclaim. CO_2 is of particular interest due to its low global warming potential (GWP), no ozone depletion potential (ODP) and characteristics which are promising for use as refrigerant. One thermodynamic property that distinguish R744 from the other relevant substances in particular is the critical temperature. R744 has a critical temperature of 31.1°C, which is significantly lower compared with other refrigerants. Such a low critical temperature implies that the refrigeration cycle is running transcritical when the ambient temperature at the heat rejection is close to 30°C or the excess heat is used to heat up domestic water. Because the COP are lower at transcritical operation, refrigerant systems with R744 solely is not suitable in warmer climates. In colder climates however, systems with R744 solely has been proved to be competitive

with other systems [11, 12]. The most relevant refrigeration system for supermarkets with R744 solely are the R744 Transcritical Booster System [10, 13, 14].

The supermarkets investigated in this thesis are such systems. Other systems where R744 are used are indirect arrangements and cascade systems [13]. R744 works as a secondary refrigerant in both of these systems. Systems using HFC for both the low and temperature (LT) and medium temperature (MT), the secondary and primary circuit respectively, are centralised direct expansion systems. A very similar system is the distributed direct expansion system, where the only difference is the location of the compressors. The following sectors are based on the writer's project thesis of 2014.

2.1 Indirect arrangements for refrigeration systems using CO_2

By using indirect arrangements, CO_2 operates as the refrigerant at the secondary circuit at reasonable pressures and at conditions well beneath the critical point. In indirect arrangements with CO_2 as secondary fluid, a pump circulates the CO_2 through an evaporator, before it is condensed. Heat is transferred to the primary refrigerant circuit in the condenser, which works as an evaporator in the primary circuit. An accumulation tank collects the CO_2 downstream or upstream of the condenser/evaporator. CO_2 has higher heat transfer coefficient at vapour state than at liquid state, hence the placement of the accumulation tank will affect the heat transfer rate in the evaporator/condenser. Figure 1 shows simple schematics for the two alternative indirect arrangements. The arrangement in Figure 2 (A) is expected to have lower performance than the arrangement in Figure 2 (B) due to the placement of the accumulation tank. The pressure level at the display case/evaporator depends on the temperature level. Cooling cabinets usually provide temperatures around 3°C, and by assuming a 7°C difference between the display temperature and the refrigerant inlet temperature, the pressure level will be reasonable (31 bar). The primary refrigerant evaporates due to energy transfer from low temperate CO_2. A pump circulates the CO_2 in liquid state from the accumulation tank to the evaporator. The evaporator is kept wept at in all circulation states (ratio between liquid and gas phase in the accumulation tank). The temperature difference across the evaporator is small and stable temperature at the refrigerant side in evaporators results in uniform frost formation, and this minimize the defrosting time [13].

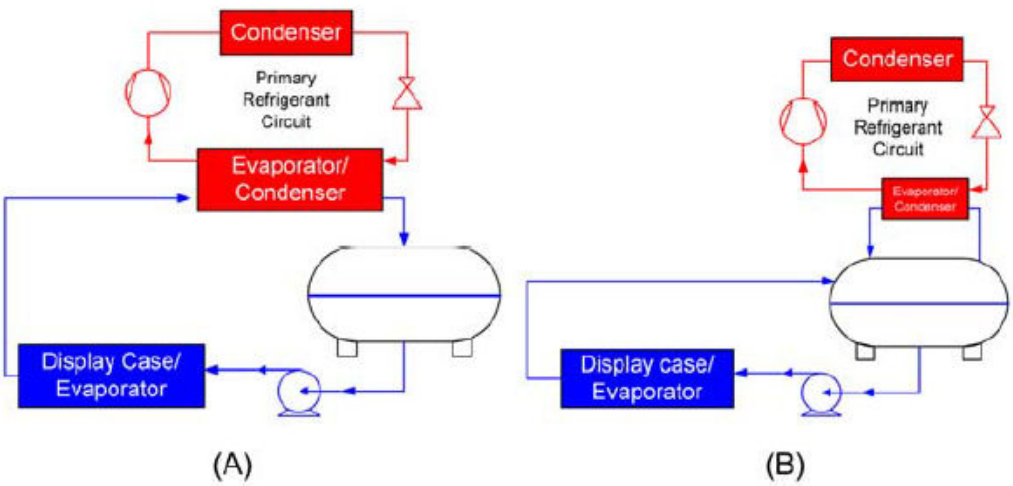


Figure 1: Simple schematic of two indirect refrigeration system arrangements [13].

2.1.1 Cascade refrigeration systems using CO_2

Cascade refrigeration systems consist of a low-temperature, a medium-temperature and a high-temperature refrigeration circuit. CO_2-cascade arrangements have a high-temperature circuit with propane, NH_3 or R404A, and CO_2 in the lower temperature circuits, which provides freezing (-20°C) and cooling (4°C). There are multiple alternatives for the cascade joint (condenser that connects the primary refrigerant and the CO_2) and the evaporator arrangement (the evaporator in the low-temperature CO_2 circuit). Figure 2-9 shows a simple schematic of a cascade system with alternative arrangements. The heat exchanger is a condensing unit for the CO_2-circuit and an evaporating unit for the high-temperature circuit refrigerant. Only one heat exchanger is necessary in the cascade system. This is due to CO_2 being in two-phase in the cooling process, and a receiver tank in each of the lower temperature levels distribute liquid and gas to the systems components (pumps, compressor and display cabinets). The evaporator/condenser is typically a plate, shell-and-plate or shell-and-tube heat exchanger. For cascade systems with single phase secondary refrigerant in the lower temperature circuits, the primary refrigerant can arrange to expand in two separate heat exchangers in order provide cooling at the medium temperature circuit and freezing at the lower temperature circuit.

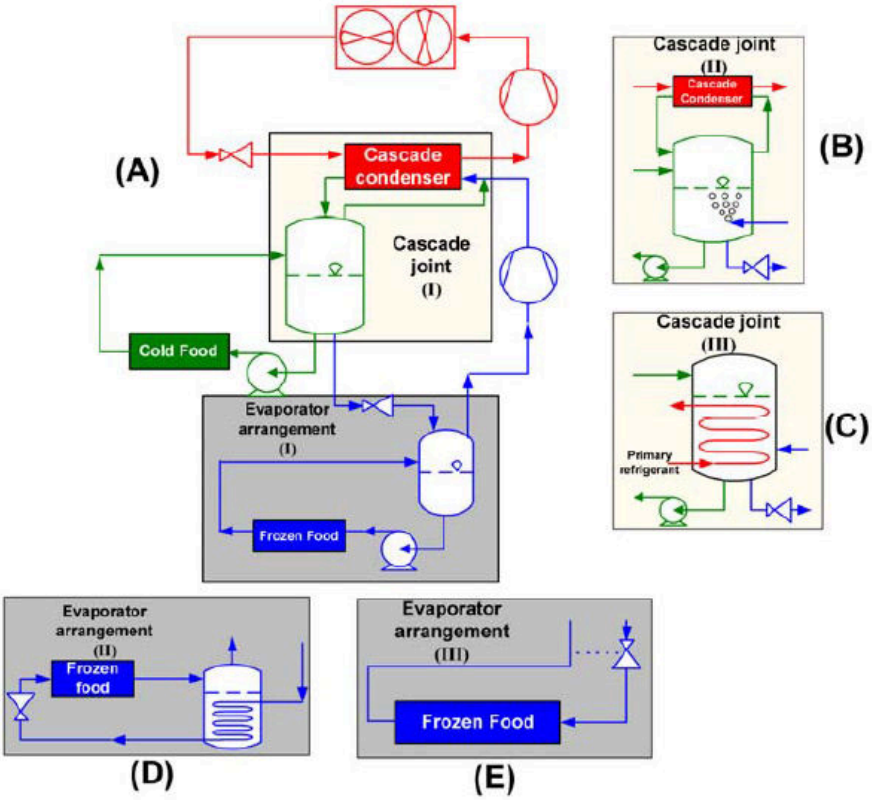


Figure 2: Cascade system with CO_2 at the medium and low-temperature levels. B and C are alternative arrangements for the cascade joint, while D and E are alternative arrangements for the evaporator arrangement [13]

For the cascade system with CO_2 at lower temperature and pressure levels, the COP of CO_2 refrigeration cycles is not far behind the COP of the equivalent refrigeration cycles (same evaporation temperature and condenser temperature) of R134a, NH_3 and R404A. Figure 2-10 shows the correlation between the COP of these refrigerants and CO_2 at T_{evap}=-10°C and T_{evap}=-35°C with assumptions of an ideal cycle. At T_{cond} above the critical temperature of CO_2, the COP is obtained at optimum pressure at the high-pressure side. Low pressure and temperature losses in the systems components will in real life contribute to higher COP of CO_2-systems compared to the other systems. The refrigerants in the cascade systems operates within the system's boundaries, and can an achieve high COP [13]. The cascade systems offer a way to use CO_2 in refrigeration systems without the high compressor discharge pressure CO_2-only refrigerant systems tends to have.

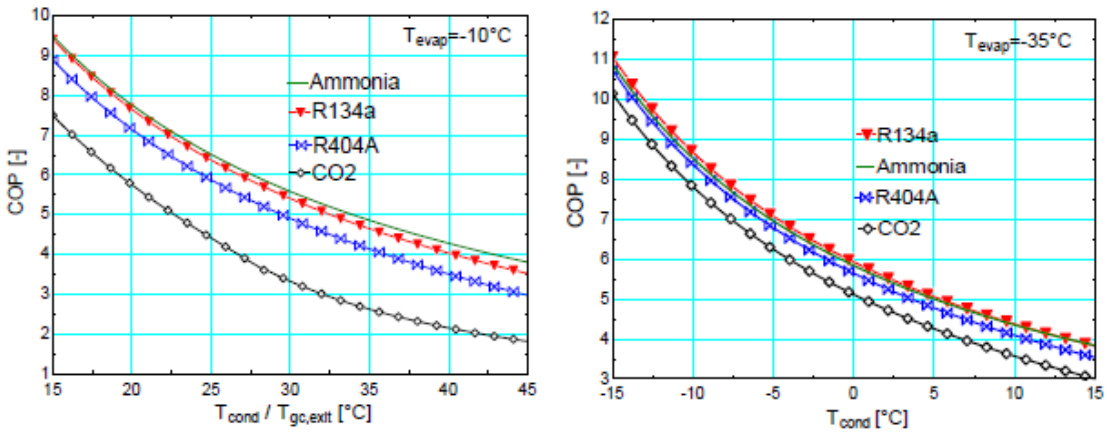


Figure 3: COP of CO_2, NH_3, R-134a and R-404A in ideal high stage (-10°C) and ideal low stage (-35°C) operation

2.1.2 Transcritical CO_2 booster system

An advantage of refrigeration system using only CO_2 is the absence of the evaporator/condenser, which connects the primary and the secondary circuits in the indirect arrangements using CO_2. A disadvantage is the high operating pressure at the heat rejection. High ambient temperatures at the heat rejection gives a transcritical refrigeration cycle for CO_2-only systems. Because of the thermodynamic properties of CO_2, controlling the compressor discharge pressure becomes important to obtain the best possible refrigeration performance. At cold climates, such as in Norway during most parts of the year, the ambient temperatures are sufficiently low enough for CO_2-only refrigeration systems to operate subcritical. If the heat from the gas cooler (condensing unit at supercritical operation) is utilized for hot water production, or other heating productions, the overall efficiency may be high for CO_2-only systems, despite transcritical operation and correlated low COP for the refrigeration system.

There are three main alternative system solutions for CO_2-only refrigerant systems: parallel, centralized and booster [15]. This section will emphasize on the booster arrangement due to the refrigerant system described in chapter 4 is a booster arrangement.

The parallel system differs from the other two arrangements due to separate cooling circuits. Direct expansion valves lower the temperature and the pressure downstream of the condenser in each of the circuits. Single stage compression is used in the medium temperature circuit and two-stage compression is used in the low temperature circuit. The two-stage compression in the low temperature circuit decreases the discharge temperature, minimize the compression losses and reduce the compression work compared with a single stage compression with the same heat sink and heat source temperatures [15].

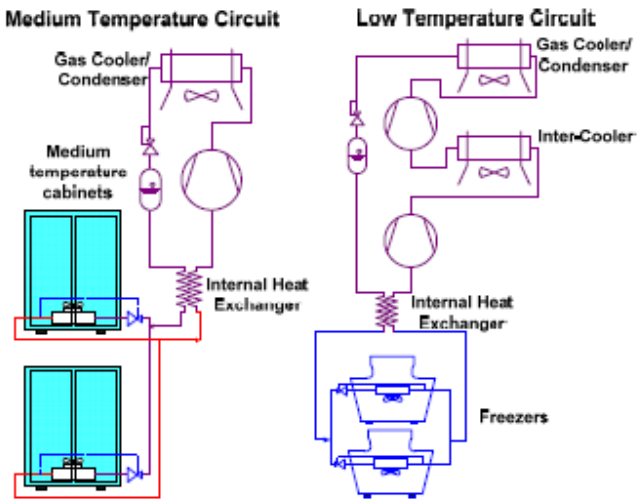


Figure 4: Parallel arrangement for CO_2-only refrigeration system.

The cooling and freezing cabinets are connected by an accumulation tank in centralized arrangements for CO_2-only refrigeration systems. The medium temperature cabinets are flooded with CO_2. A pump circulates the CO_2 to the medium temperature cabinets, and evaporated CO_2 returns to the accumulation tank. Direct expansion valves lower the temperature and pressure of the CO_2 flowing from the accumulation tank to the low temperature cabinets. The evaporated CO_2 is compressed and returns to the accumulation tank. The pressure and temperature of the CO_2 in the tank is regulated by a conventional refrigeration cycle with an internal heat exchanger (Figure 4).

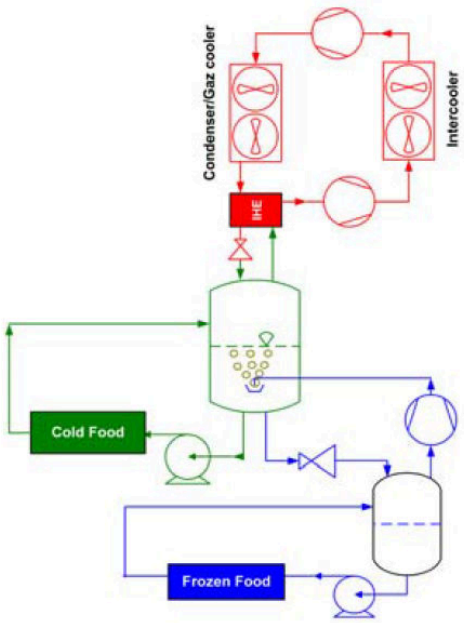


Figure 5: Centralized arrangement for CO_2-only refrigeration system [15]

The booster system is a two-stage compression cycle. Figure 6 shows a piping and instrumentation diagram of a transcritical CO_2 booster system, and important equipment are marked with numbers. A high-pressure control valve (4) reduces the pressure of the cooled CO_2 from the gas cooler/condenser (2). The flow is divided into gas and liquid in a receiver tank (5). The gas flows to the suction line of the high-pressure compressor (1) through a bypass valve (6). This "boost" the load going into the compressor and hence the heating load going into the gas cooler/condenser. Expansion valves (7 and 8) controls the pressure and temperature of the liquid CO_2 flowing from the receiver to the medium temperature (9) and low temperature (10) evaporators. The evaporators ensure cooling (-10°C) and freezing (-30°C) to the cabinets storing the food. The gas from the low temperature evaporator is compressed in a low temperature compressor (11). The gas from the medium temperature evaporator mixes with the compressed CO_2 from the low temperature compressor. The mix enters the suction line of the high-pressure compressors, along with the bypass-gas from the receiver. The CO_2 is finally compressed in the high-pressure compressor and completes the cycle.

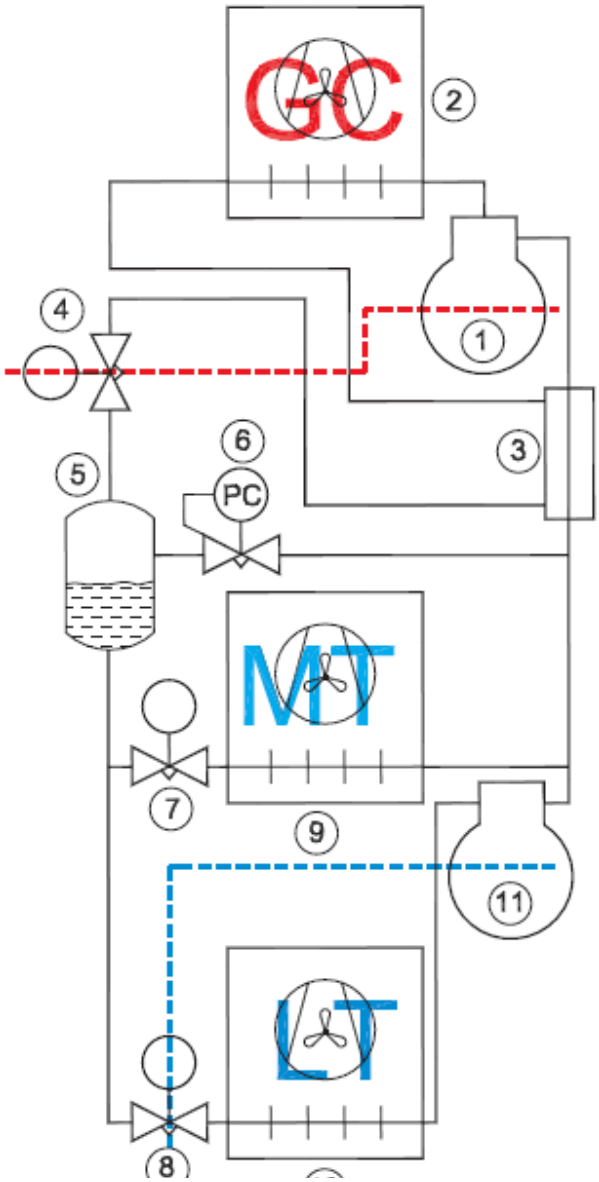


Figure 6: PI diagram of Transcritical CO2 booster refrigeration system

In order to increase the performance of refrigeration cycle, different solutions for internal heat exchanging exist. The internal heat exchanger (3) is typically placed between the high-temperature compressor and the receiver tank. The heat sink is the refrigerant flowing to the compressor, and the heat source is the refrigerant being cooled in the condenser/gas cooler. The refrigerant is subcooled, and the temperature-rise of the inlet fluid in the compressor ensures less risk of droplets contained in the fluid.

For 32°C in design condition for ambient temperature (which means the system is running

transcritical), the pressure in the gas cooler is approximately 90 bar for optimum COP. The high-pressure valve controlling the pressure in the receiver is a constant backpressure valve. The pressure must be higher than the highest evaporator pressure and lower than the design pressure for the specific receiver. The pressure in the receiver should also minimize the amount of liquid in the gas bypass. The pressure is independent of the ambient temperature, but the flow ratio of gas and liquid varies the gas cooler pressure and the outlet temperature in the gas cooler. The flow in the evaporators is a function of the cooling capacity, and the flow varies with a factor of 2, depending on the ambient temperature. This variation offers challenges in terms of the design of the pipes and the oil return system in the compressors.

3 Case supermarkets

Three case supermarkets are evaluated in terms of energy system performance in chapter 3. The energy systems of the supermarkets are described in this section to give a profound understanding of the interaction between the systems' components. The way the components are controlled and how they interact on each other determines the performances of the systems. The way the systems operate are not only determined by internally operation modes and limitations of capacities, but also by external factors. These factors are temperature, air pressure, humidity, precipitation, sunshine, cloudiness and winds; the local climate. The local climate affects especially the operation mode of transcritical CO_2 refrigeration systems, as described in chapter 2.1. The warmest and coldest days during 2019 at each of the supermarkets of interest were chosen as days for evaluation and analysis for the systems. Two of the supermarkets, Kiwi Olsvik and Kiwi Tertnes, are located in Bergen in Western Norway. The third supermarket is in Bærum, close to Oslo (see Figure 7). Bærum is within the humid continental climate as defined by the Köppen climate classification. This means that the mean monthly temperature of the coldest month is below -3°C and there are four months whose mean monthly temperatures are minimum 10°C. Winters are typically cold and summers are relative cool. There is typically no dry season, as the precipitation is dispersed evenly throughout the year. The climate at the Western coast is mostly oceanic climate. The monthly mean temperature is below 22°C in the warmest month, and above 0°C in the coldest month.

Table 1 presents key temperature data for Bergen and Bærum from January to September in 2019. The weather has met the climate classification definitions regarding monthly mean temperatures. The hottest and coldest periods have noteworthy fallen into the same periods, as seen in table 1 and from Figure 8. The warmest and coldest day in Bergen was at both occasions the day before the warmest and coldest day in Bærum. The set-dates for evaluation and analyses of the supermarkets during cold climate in Bergen and Bærum is January 28 and January 29 respectively. Likewise, the set-dates for evaluation of the supermarkets during warm climate are July 27 for both Bergen and Bærum. The warmest day in Bærum was July 28, a Sunday. Only days when the supermarkets operate normally are considered, meaning Sundays and certain holidays are not included. This means that the single and three-days maximum and minimum average temperatures presented in the table below are for normal operating days only.

Table 1: Noticeable temperatures 2019 (°C) in Bergen and Bærum

	Bergen	Bærum
Hottest day	24.6[a]	23.6[b]
Coldest day	-2.7[c]	-9.7[d]
Hottest three-day avg. streak	23.73[e]	22.46[f]
Coldest three-day avg. streak	-0.7[g]	-8.67[h]
Average (January-September)	10.05	8.22
January average	2.63	-3.04
February average	5.33	-0.23
March average	5.33	-0.23
April average	4.58	1.58
May average	9.83	7.46
June average	14.10	14.55
July average	16.16	16.95
August average	16.45	15.89
September average	11.63	10.74

[a] July 27, 2019; [b] July 27, 2019; [c] January 28, 2019; [d] January 29, 2019
[e] July 25-27, 2019; [f] July 25-27, [g] January 28-30, 2019; [h] January 29-31, 2019

Figure 7: Geographical location of the three supermarkets in Bergen and Oslo, Norway

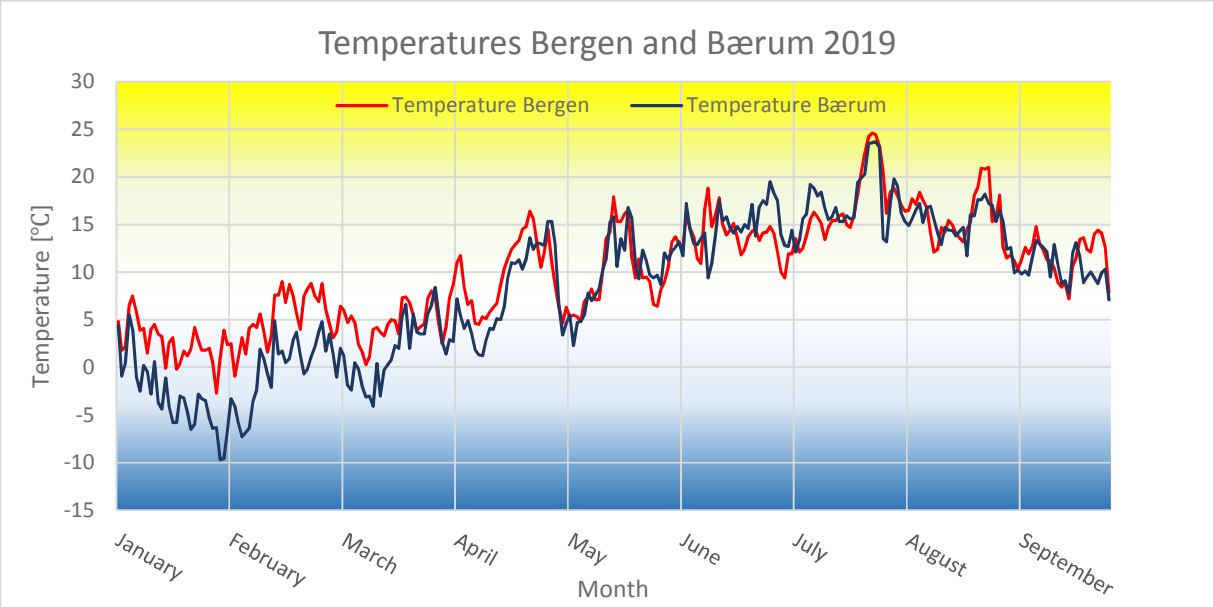


Figure 8: Temperature development graph for Bergen and Bærum January to September 2019

Figure 9: The façade of Kiwi Tertnes (top left), Kiwi Olsvik (top right) and Spar Snarøya (bottom)

3.1 The common features of three transcritical CO_2 refrigeration systems

The three refrigeration systems of the supermarkets SPAR Snarøya, Kiwi Tertnes and Kiwi Olsvik have CO_2 as the working fluid. The systems share many of the same features as they are all transcritical CO_2 refrigeration systems, albeit the technical approach differs slightly. The main differences are in the heat rejection process at the high stage pressure section. This section gives a short description of the similarities of the three case refrigeration systems. Each of the refrigeration systems has its own section in this chapter where the characteristic features are described.

There are four pressure sections in a transcritical refrigeration booster system. The pressure in the separator is typically 40 bar, while the pressure over the evaporators is typically 30 bar at the medium temperature evaporators (MT evaporators) and around 15 bar at the low temperature evaporators (LT evaporators). These two sections are labelled after their temperature levels, namely the LT section (LTS) and the MT section (MTS). The pressure section from the high-pressure valve (HPV) to the expansion valves prior to the evaporators and to the by-pass valve (BPV) is labelled the intermediate temperature section (ITS). Two of the refrigeration systems (Kiwi Tertnes and Spar Snarøya) have a parallel compressor (PC, also known as auxiliary compressor), and the section upstream of the parallel compressor is part of the ITS. Figures 14, 15 and 17 present PI and sensor diagrams for each system, and the pressure sections are easily recognized for their colours. The LTS and MTS are blue and green respectively, and the ITS is yellow. The fourth pressure section is labelled the high pressure section (HPS) and is coloured red in the PI and sensor diagrams. This section starts with the MT compressors (and the parallel compressor in the cases of Kiwi Olsvik and Spar Snarøya) and continues through the heat rejection units and is completed at the HPV.

The supermarkets possess a various amount of chilling and freezing cabinets, as shown in table 2. A similarity between all three systems is the type of expansion valves in the storage rooms. The expansion valves are electric expansion control valves from Danfoss. The cabinets are equipped with integrated expansion valves, and the types of cabinets in the three supermarkets are presented in table 3. The evaporation process in the LT cabinets- and rooms is in the temperature region of -23°C to -18°C. Similarly, the MT cabinets- and rooms have an evaporation temperature range from -17°C to 4°C.

Table 2: Number of installed chilling and freezing appliances

	Freezers	Freezer rooms	Coolers	Cold rooms	Fruit and vegetables cabinets
SPAR Snarøya	6	1	15	6	4
KIWI Tertnes	8	1	10	4	3
KIWI Olsvik	6	1	6	4	5

The evaporators in the cold and freezer rooms are Güntner- GACCs and GASCs, except for the evaporator (*LU-VE F27HC*) in the freezer room at Kiwi Olsvik. The evaporators are dry evaporators, and in contrast to wet evaporators, the CO_2 is completely evaporated and superheated before entering the low stage compressors. The transcritical booster system has a separator downstream of the high-pressure expansion valve, thus share some of the same aspects of components and technical arrangement as a flooded evaporator (i.e. wet evaporator) system. The systems of Kiwi Olsvik and Kiwi Tertnes are similar for LPS, while the system of Spar Snarøya has more components dealing with potential slugging in the low stage compressors, see chapter 3.4. The LT compressor unit controls the pressure over the LT evaporators. When the freezing demand increases, which happens at instances where more goods are to be held frozen or the temperature around the freezing cabinets increases, the LT compressor unit will increase its capacity. More on compressors and the compressor control mechanisms below table 3.

Table 3: Overview of the types of cabinets and evaporators in relation to cold rooms and freezer rooms

Label	Model	Evaporator
FM 60-61	Colonia 3 BT [-22/25°C]	
FM 62-63	Colonia 3 BT [-22/25°C]	
FM 64	Astana 2C H216	
FM 65	Astana 2C H216	
FR 76	Wearhouse	F27HC 70 E7
KM 01	Berlino 3 FCW 120/216 [+4/+6°C]	
KM 05-06	Berlino 3 FCW 120/216 [+4/+6°C]	
KM 11-12	Berlino 3 FCW 120/216 [+4/+6°C]	
KM 20-21	Santiago LF 105/216 [-1/+1°C]	
KM 22-23-24	Santiago LF 105/216 [-1/+1°C]	
KM 35-36	Santiago LF 105/216 [-1/+1°C]	
KR 45	Cold room – dairies	Guntner GASC CX 031.1
KR 46	Cold room – fruits	Guntner GASC CX 031.1
KR 47	Cold room – garbage	Guntner GASC CX 031.1
F60-61	Colonia 3 BT [-22/25°C]	
F62-64	Colonia 3 BT [-22/25°C]	
F65-66	Astana 2C 216 EverClear [-22°C/-25°C]	
F76	Wearhouse	Guntner GACC CX 031.1
K01	Berlino 3 FCW 120/216 [+2°C /+4°C]	
K05-06	Berlino 3 FCW 120/216 [+4/+6°C]	
K11-12	Berlino 3 FCW 120/216 [+4/+6°C]	
K20-23	Santiago LF 105/216 [-1/+1°C]	

K24-25	Santiago LF 105/216 [-1/+1°C]	
K35-36	Santiago LF 95/150 [-1/+1°C]	
K45-46	Cold room – dairies	Guntner GASC CX 031.1
K47	Cold room – fruits	Guntner GASC CX 031.1
K48	Cold room - garbage	Guntner GASC CX 031.1
F01-03	Toronto G3	
F04-06	Toronto G3	
F07	Freezer room	Guntner GACC CX 031.1
K01	Osaka 3 90/216	
K02-03	Osaka 90/216	
K04	Osaka 90/216	
K05	Osaka 90/216	
K06	Osaka 90/216	
K08	Victoria VCB [0/+2°C]	
K09	Victoria VCB [+2/+4°C] fish	
K10-13	Osaka 90/150	
K14-16	Osaka 90/216	
K17-20	Berlino 3 FCW 120/216 [+4/+6°C]	
K21	Cold room - dairies	Guntner GACC CX 031.1
K22	Cold room – packaged food	Guntner GACC CX 031.1
K23	Cold room – unpackaged food	Guntner GACC CX 031.1
K24	Cold room – fruit	Guntner GACC CX 031.1
K25	Cold room - Fish	Guntner GACC CX 031.1
K26	Cold room - Garbage	Guntner GACC CX 031.1

3.1.1 Compressor units

A technical aspect all three of the systems share is the type of compressors used. All compressors are from the compressor fabricant Bitzer, and an overview of the types is listed in table 4.

The compressors are designed for specific pressure and condensing temperatures. The maximum pressure level limit is 160 bar for the MT compressors, and 100 bar for the LT compressors. They are semi-hermetic reciprocating compressors. A semi-hermetic compressor has the motor and the compressor bolted together, and unlike the hermetic compressor, it is possible to separate the motor and the compressor for inspection and repairing. The compressor and the rotor are assembled on the same crankshaft. The rotor and the compressor are inside a sealed housing, and the electric motor stator is outside of the housing around the rotor. This arrangement makes it possible to repair and replace the stator windings, as well as other benefits which increase the longevity of the compressors [16].

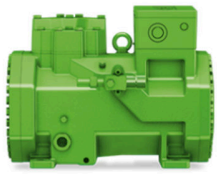


Figure 10: Illustation of a Bitzer compressor used for transcritical CO_2 refrigeration systems [1]

Capacity control of the compressors are important in terms of energy efficiency. A standard feature for transcritical CO_2 booster refrigeration systems are one variable speed driven (VSD) compressor in each compressor unit [17]. The lower limit for speed reduction in compressors is 50% [16]. The VSD compressor ensures energy saving and fatigue reduction of the compressors, as it prevents on and off operation at lower loads. At an operation mode where the compressor capacity is greater than the capacity of compressor number one, the VSD compressor runs with a capacity adjusted to the total load demand and the capacity of one of the other or both compressors in the specific compressor unit. The maximum displacement volume of a compressor unit is when all compressors are running at 100 percent. The minimum displacement volume occurs when the VSD compressor runs on minimum frequency [18].

Table 4: Bitzer compressors used in the case supermarkets

Supermarket	Pressure section	Bitzer compressor type	Capacity [kW]
Kiwi Olsvik	MT	1 x 60 Hz 4KTC-10K	17.9
Kiwi Olsvik	MT	2 x 50 Hz 4HTC-20K	23.1
Kiwi Olsvik	LT	1 x 70 Hz 2KME-1K	4.8
Kiwi Olsvik	LT	1 x 50 Hz 2JME-2K	6.2
Kiwi Tertnes	MT	1 x 70 Hz 4PTC-7K	10.8
Kiwi Tertnes	MT	1 x 50 Hz 4MTC-10K	17.3
Kiwi Tertnes	MT	1 x 50 Hz 4KTC-10K	25.5
Kiwi Tertnes	IT	1 x 70 Hz 4PTC-7K	15.9
Kiwi Tertnes	LT	1 x 70 Hz 2KME-1K	4.4
Kiwi Tertnes	LT	1 x 50 Hz 2JME-2K	5.6
Spar Snarøya	MT	1 x 70 Hz 4PTC-7K	9.6
Spar Snarøya	MT	1 x 50 Hz 4MTC-10K	15.4
Spar Snarøya	MT	1 x 50 Hz 4KTC-10K	22.8
Spar Snarøya	IT	1 x 70 Hz 4JTC-15K	31.2
Spar Snarøya	LT	1 x 70 Hz 2KME-1K	4.8
Spar Snarøya	LT	1 x 50 Hz 2KME-1K	4.8

3.1.2 Liquid receiver

In a transcritical CO_2 booster refrigeration system the workload of the compressors, gas cooler and evaporators will vary with the cooling demand, and the heating demand if a heat recovery system is integrated to the refrigeration system is in place. This means that the volume of CO_2 circulating changes rapidly and needs to be controlled. The receiver, or the separator, in all three systems are a liquid receiver with suction accumulator and heat exchanger delivered by Frigomec Klimal. The liquid receiver contains saturated CO_2 at the surface of the liquid CO_2, liquid at the bottom and gas at the top. Subcooled CO_2 flows from the bottom of the tank toward the expansion valves prior to each of the LT- and MT evaporators, which are constituting a dry expansion evaporator arrangement. The receiver tank works as a storage and buffer tank for the saturated liquid CO_2 in the refrigeration system. The design pressure in the receiver is 60 bar. The pressure is controlled by a CCM 20 by-pass valve from Danfoss. At the refrigeration systems where an auxiliary compressor is present, the pressure is controlled in cooperation between the by-pass valve (BPV) and the capacity of the auxiliary compressor. It is desirable that the auxiliary compressor runs when there is a heat demand for the heat recovery cycle (HRC), due to the direct compression of the CO_2 [18]. The lowest capacity of the PC in part load operation decides the minimum flow rate of flash gas entering the compressor. The vapour content in the separator should cover the minimum flow rate for actuating parallel compression.

3.2 Transcritical CO2 refrigeration system – Kiwi Olsvik

The main difference between the transcritical CO_2 booster system at Kiwi Olsvik and the other two systems evaluated in this thesis is the technical arrangement of the HPS. Whereas the other two systems include a parallel compressor (auxiliary compressor) fed on flash gas from the separator, the by-pass arrangement in Kiwi Olsvik only serves the MT compressor suction line (see Figure 12).

The three systems have similar technical arrangement in the LPS (Spar Snarøya stands out with additional equipment, see chapter 3.4), with the variation being the numbers of installed freezing- and chilling cabinets and freezing- and cold rooms. The refrigeration system at Kiwi Olsvik serves six freezing cabinets, six chilling cabinets, one freezing storage room, four cold storage rooms and five chilling cabinets for fruits and vegetables (see table 2). The total cooling capacity at Kiwi Olsvik are 13.4 kW and 47.7 kW for freezing and chilling respectively.

Figure 12 presents a PI diagram with a map of the sensors monitoring the system. The pipes of the intermediate pressure section (IPS) are marked yellow and green. The low- and high-pressure sections (LPS and HPS) are correspondingly marked blue and red respectively. Table 6 in chapter 4.1 gives a description of the sensors for all the refrigeration systems evaluated.

The transition between LT and MT occurs in the LT compressors. At Kiwi Olsvik there are two Bitzer compressors operating within the design pressure of 46 bar each. The discharge pressure is normally between 30 and 35 bar. The compressors used in the refrigeration systems is listed in table 4. One compressor in each rack are VSD, allowing the compressor capacity to be adjusted to the refrigeration workload. The CO_2 is superheated in DX evaporators and is further superheated in the integrated heat exchanger (IHX 2) inside the separator before reaching the LT compressors. A *Ss* (Suction sensor) registrates the temperature of the suction CO_2. The compressed gas from the LT compressor(s) is mixed with the superheated CO_2 coming from the MT evaporators. The *Sd* (Discharge sensor) registrate the temperature of the LT discharge gas. The gas enters the suction line together with gas from the by-pass line when the by-pass valve is open. The by-pass valve is an electrically operated valve and provides back-pressure regulations in the receiver during subcritical application. The gas mixture enters the integrated accumulator tank inside the liquid receiver before flowing into the high temperature compressor(s) (HT Compressor(s)). The lubrication oil used in the compressors accumulates in the bottom of the accumulator tank, thus being separated from the CO_2. There are three HT compressors, and one is a VSD. The design pressure at each of the compressors is 130 bar, and the discharge pressure is normally between 65 to 90 bar. See table 4 for an overview of the compressors. As for the LT compression process, there are a Ss sensor register the MT suction temperature and a Sd register MT discharge temperature. The discharged CO_2 is cooled in a

SWEP heat exchanger working as a water-cooled gas-cooler, as described in chapter 3.2.1. Two sensors, *Sgc* (Sensor gas cooler) and *Pgc* (Pressure gas cooler) ensures that the temperature and the pressure at the HPS are tracked.

A high-pressure Danfoss CCMT electrical regulation valve controls the gas cooler pressure and expands the CO_2 downstream of the plate heat exchanger (PHE) and completes the refrigeration cycle. There are two periods evaluated for each of the refrigeration systems. Figure 11 shows the refrigeration cycle for Kiwi Olsvik during the warmest and coldest period in 2019. The sensors in the HSC is *S7* (sensor 7) and various sensors such as sensors tracking the operation mode of the flow switch, the capacity of the heating battery, and the inlet and outlet temperatures of the heating battery.

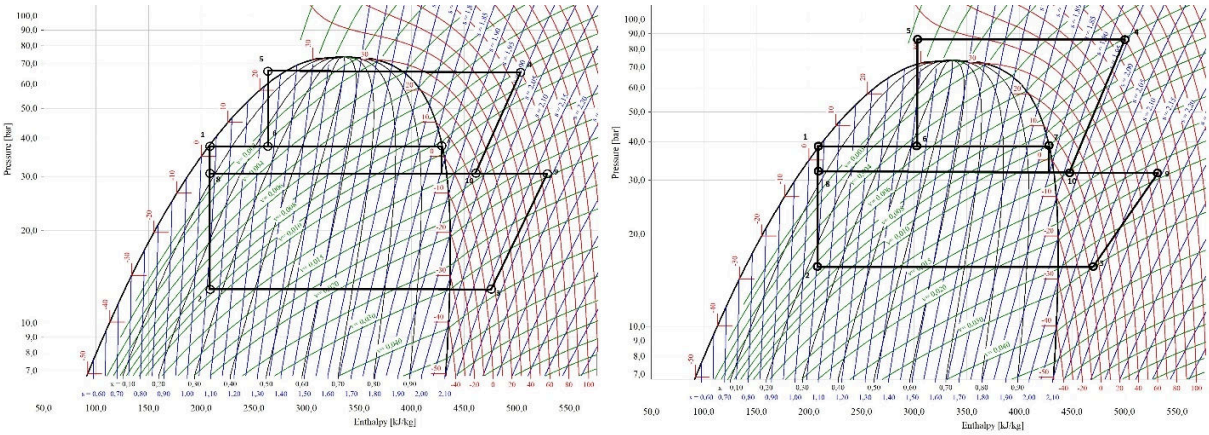


Figure 11: log P-H diagram of the refrigeration cycle of Kiwi Olsvik during the warmest (left) and coldest (right) continuous normal operationally three-day period in 2019

3.2.1 3.2.1 Heat Recovery – Kiwi Olsvik

The HR HX in the system of Kiwi Olsvik is, unlike the HR HXs in the other two refrigeration systems, the gas-cooler (condenser) of the refrigeration cycle. Brine, a mixture of water and 35% ethylene glycol, is circulated by the power of two pumps operating in parallel and by an extra pump ensuring flow to the heat battery at the air handling unit (AHU). An expansion tub with 18 L volume regulated by a jacking pump deals with thermal expansion of the working fluid.

The heat removal capacity in the HR HX is dependent on the capacity of the dry-cooler. The dry-cooler has two 4-fans columns and the maximum capacity is 130 kW. The fan power is dependent of the outlet temperature of the dry-cooler. The fans are not operating when the compressors are not running, and they slow down when the outlet temperature of dry-cooler is lower than the set-point temperature. The pressure in the HR HX is a function of the outlet temperature of the dry-cooler (Sc3 in Figure 12) in transcritical operation, where the goal is to maintain as high COP as possible.

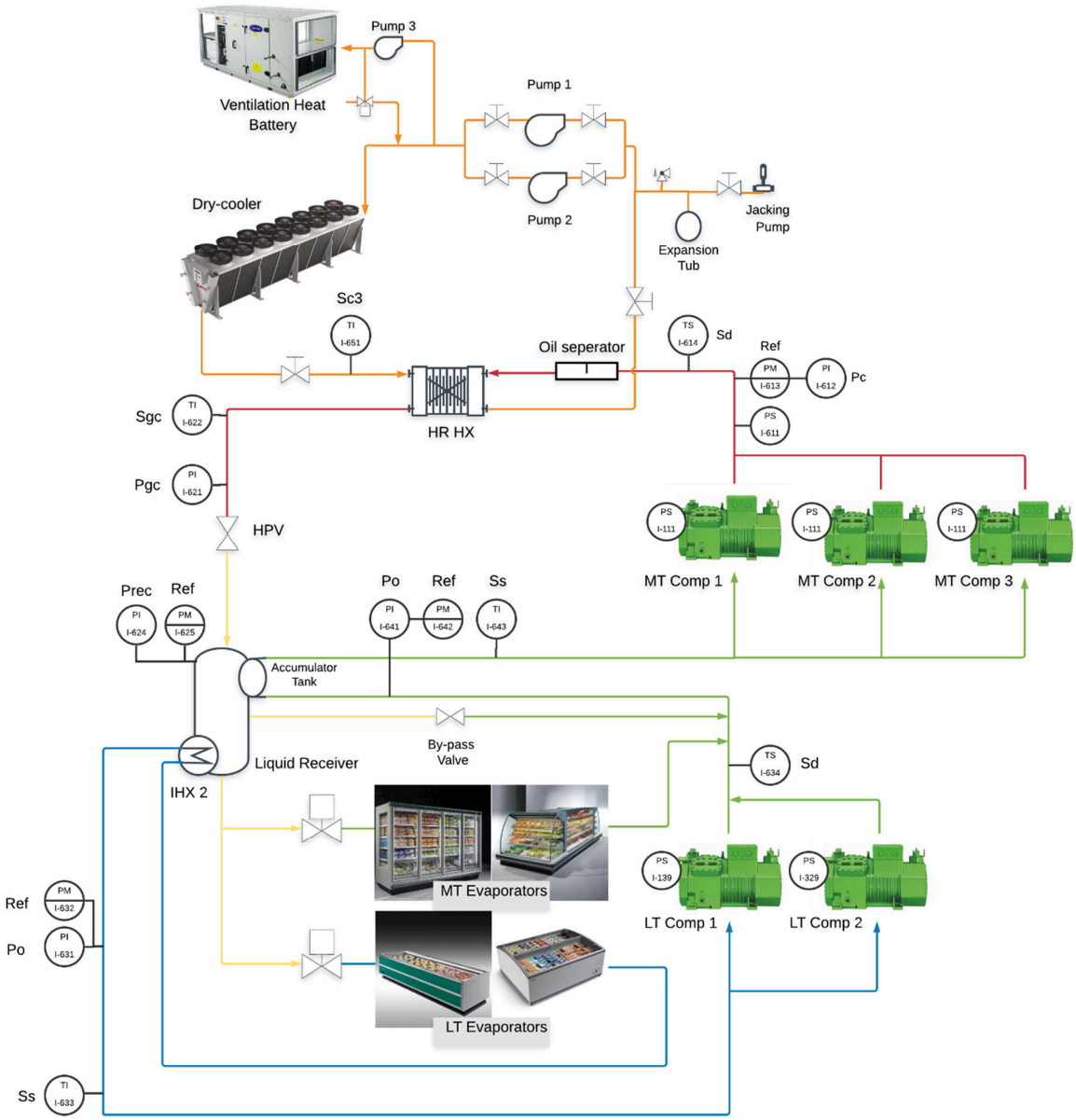


Figure 12: PI and sensor diagram Transcritical CO_2 Booster Refrigeration System Kiwi Olsvik

3.3 Transcritical refrigeration system – Kiwi Tertnes

The refrigeration systems of Kiwi Olsvik and Kiwi Tertnes are close to identical in the LPS of the systems, the difference being the numbers of evaporators (see Table 2 and 3). Kiwi Tertnes is a slightly bigger supermarket in terms of the number of installed chilling- (13 vs 11) and freezing cabinets (8 vs 6). The installed cooling- and freezing capacity at Kiwi Tertnes is 54 kW and 11.4 kW. The LT and MT evaporators are DX evaporators. The cooling and freezer cabinets are manufactured with expansion valves, while the cold- and the freezer room have electrically operated expansion valves. The cooling load in the evaporators is determined by the ambient temperature in the supermarket and the volume of the content in the storage compartment and frequency of change of content. Each evaporator has a reference value for the storage temperature, and a high surrounding temperature will consequently cause higher cooling load due to the amount of heat needed to be absorbed by the refrigerant. Higher cooling load leads to higher volume flow of the refrigerant and enforces the system to use LT compressor number 2 when the capacity of LT compressor number 1 is surpassed. LT Compressor 1 is a VSD and the total compressor work can therefore be adjusted precisely according to the cooling demand. A VSD Compressor is especially gently to the system during night load to avoid start/stopping operation of the compressor. The pressure in the LT evaporators are determined by the capacity control of the LT compressor. The CO_2 is further superheated in an integrated heat exchanger inside the receiver, typically by 6-8K. As for all the refrigeration systems, a Ss- and a Sd sensor are placed upstream and downstream of the LT compressor unit respectively.

The pressure in the MT Evaporators is determined the same way as the pressure in the LT Evaporators. The MT compressor capacity control is the determinative variable, not only for the pressure in the MT evaporators, but also for the by-pass valve exit pressure. The CO_2 entering the MT compressors is a mixture of CO_2 from the LT evaporators, MT evaporators and flash gas from the separator. An integrated accumulator tank in the top of the receiver ensures separation of oil from the CO_2 flow before it enters the suction line. There is a Ss sensors in the suction line of the MT compressors and a Sd sensor at the high-pressure line downstream of the MT compressors. The sensors continuously measure the pressure levels, which are used in this thesis to do calculations on the system.

The pressure in the receiver is controlled by the flash gas by-pass valve, and the parallel compressor. The flash gas is directly compressed in the parallel compressor; thus, the compression work is lower than for the MT compressors due to lower pressure ratio over the compressor. The high side pressure may be either subcritical or supercritical. At subcritical operation, the system acts as conventional refrigeration system. The ambient temperature serves as the limiting parameter for the subcooling of the high stage CO_2. The high side pressure is determined by the condenser exit temperature. At transcritical operation, however, the optimum high side pressure is determined by the outlet temperatures of the heat rejecting devices.

3.3.1 Heat Recovery – Kiwi Tertnes

The HR HX serves as a de-superheater in the transcritical CO_2 booster refrigeration system. The discharge CO_2 from the MT compressors is in an external heat recovery cycle like the one at Kiwi Olsvik. The cold working fluid is a mixture of water and 35% ethylene glycol. The working fluid transports heat absorbed from the CO_2, and the heat is exploited to heat the supply air in the AHU. The heat exchanger in the AHU is also a Swep B18 PHE. Unlike the system of Kiwi Olsvik, there is no dry cooler attached to this circuit. Instead, the heat removal process occurs in the gas cooler. Twelve fans give the gas-cooler a capacity of 104 kW with the following conditions; an air inlet temperature of 30°C, air velocity of 1.2 m/s, inlet CO_2 temperature of 97.1°C and a volume flow of 11.06 m^3/h. The CO_2 is cooled by over 64°C at these conditions and at a high-side pressure of 84.8 bar. A log P-H diagram for operation during the coldest and warmest period in Bergen is presented in Fig. 13 below.

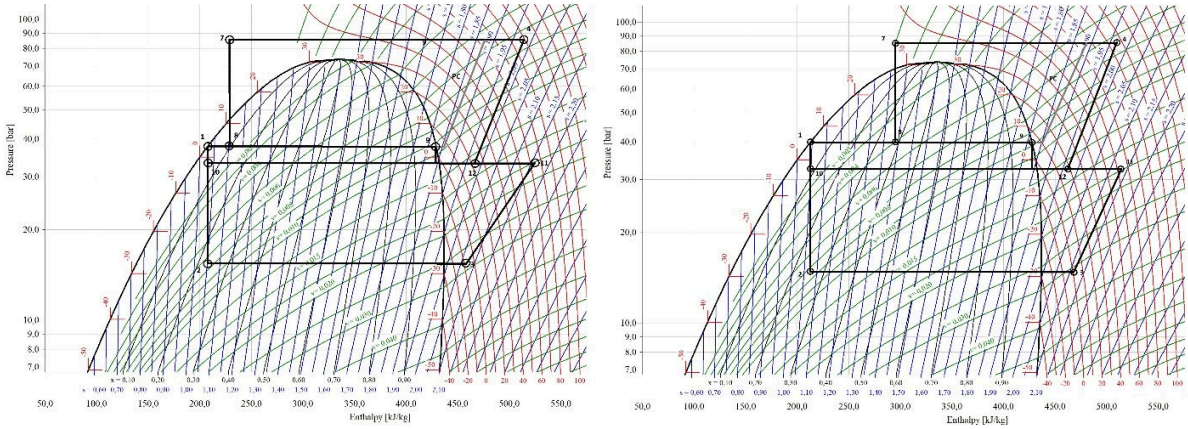


Figure 13: log P-H diagram of the refrigeration cycle of Kiwi Tertnes during the warmest (left) and coldest (right) continuous normal operationally three-day period in 2019

Another PHE is located downstream of the gas-cooler. It is shown as IHX 1 in the PI and senor diagram in Fig. 14. The heat exchanger ensures superheating of the CO_2 before entering the suction line of the IT compressor. In this process the CO_2 at the HPS is also further cooled down and is beneficial in times where ventilation cooling battery (VCB) is utilizing cold CO_2 for cooling of the supply air in the AHU.

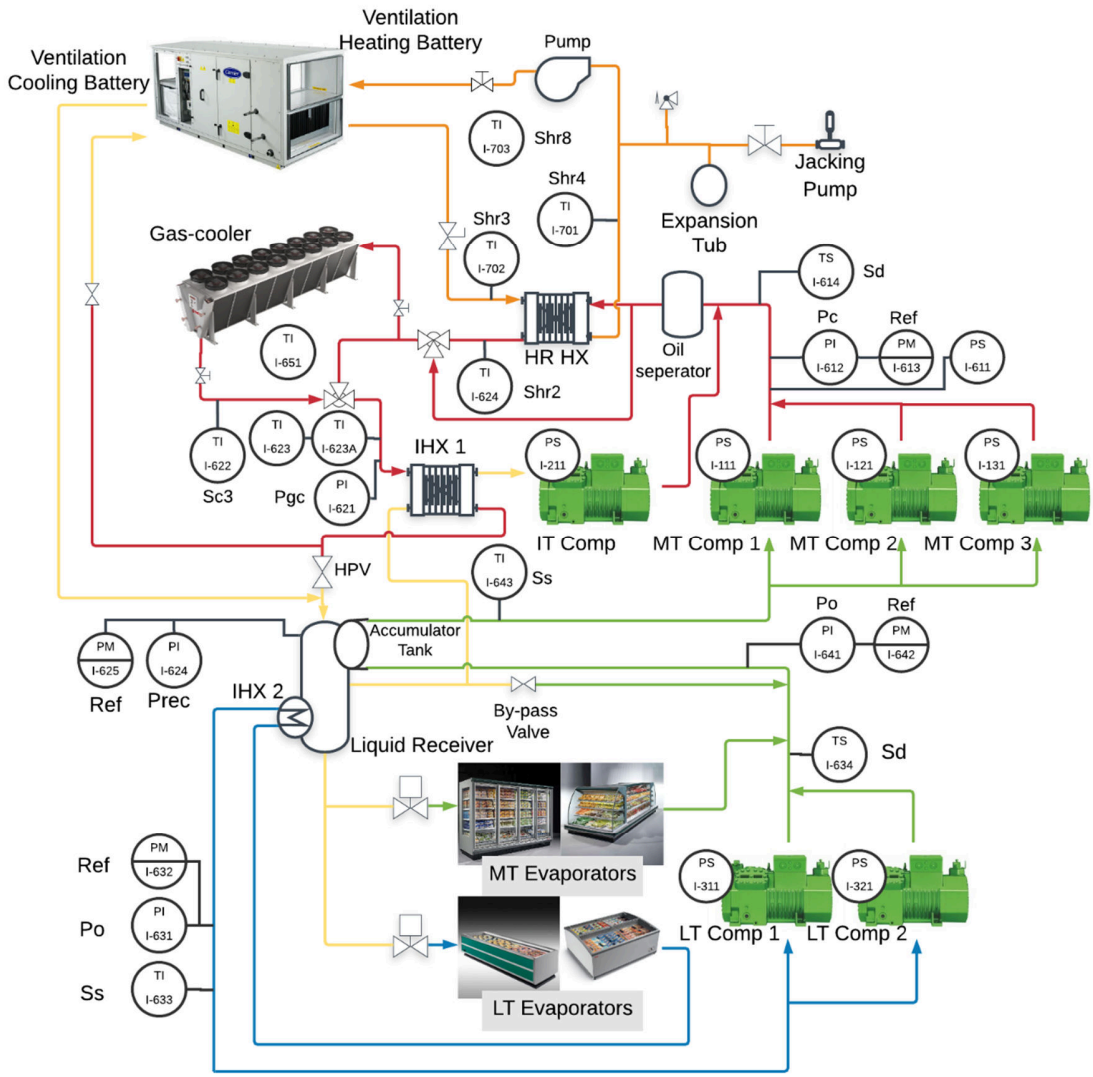


Figure 14: PI and sensor diagram Transcritical CO2 Booster Refrigeration System Kiwi Tertnes

3.4 Transcritical refrigeration system – Spar Snarøya

Spar Snarøya is in Bærum, just outside Oslo. Like the CO_2 refrigeration system of Kiwi Tertnes, the refrigeration system of Spar Snarøya has an auxiliary compressor. What mainly distinguish these two systems are the technical approach for heat recovering and the addition of an extra internal heat exchanger at the LPS of Spar Snarøya.

The different pressure levels are indicated with the colours blue, green, yellow and red for LT-, MT-, IT- and HP-pressure level respectively. As seen in Figure 15, the additional internal heat exchanger in the LPS is labelled *IHX 2*. The function of IHX 2 is allowing heat being transferred from the CO_2 liquid flow coming from the separator to the colder superheated MT-evaporator outlet flow. This heat exchanging process enhances the refrigeration capacity at LT- and MT level.

The control mechanisms are as described in chapter 3.2 and 3.3. The refrigeration system of Spar Snarøya has a different web monitoring database, but the project planner behind the systems is the same. There are same type of sensor gathering temperature and pressure data at the suction lines and high-pressure pipes, and sensors logging temperature data downstream of the heat rejecting devices. An additional sensor compared to the system of Kiwi Tertnes is the placement of the Pgc sensor. This sensor is located downstream of IHX 1 rather than upstream.

The two cooling cabinets used for fresh food operate at a different pressure level than the other cooling compartments. These MT evaporators have a lower suction pressure than the other MT evaporators due to a demand for lower storing temperatures for perishable food. The pressure of the CO_2 is throttled by a Danfoss CCM expansion valve downstream of the two MT evaporators, hence lowering the temperature to meet the temperature range of LT CO_2. The placement of the

cooling and freezing compartments in the web monitor interface system IWMAC can be seen in Figure 15 below.

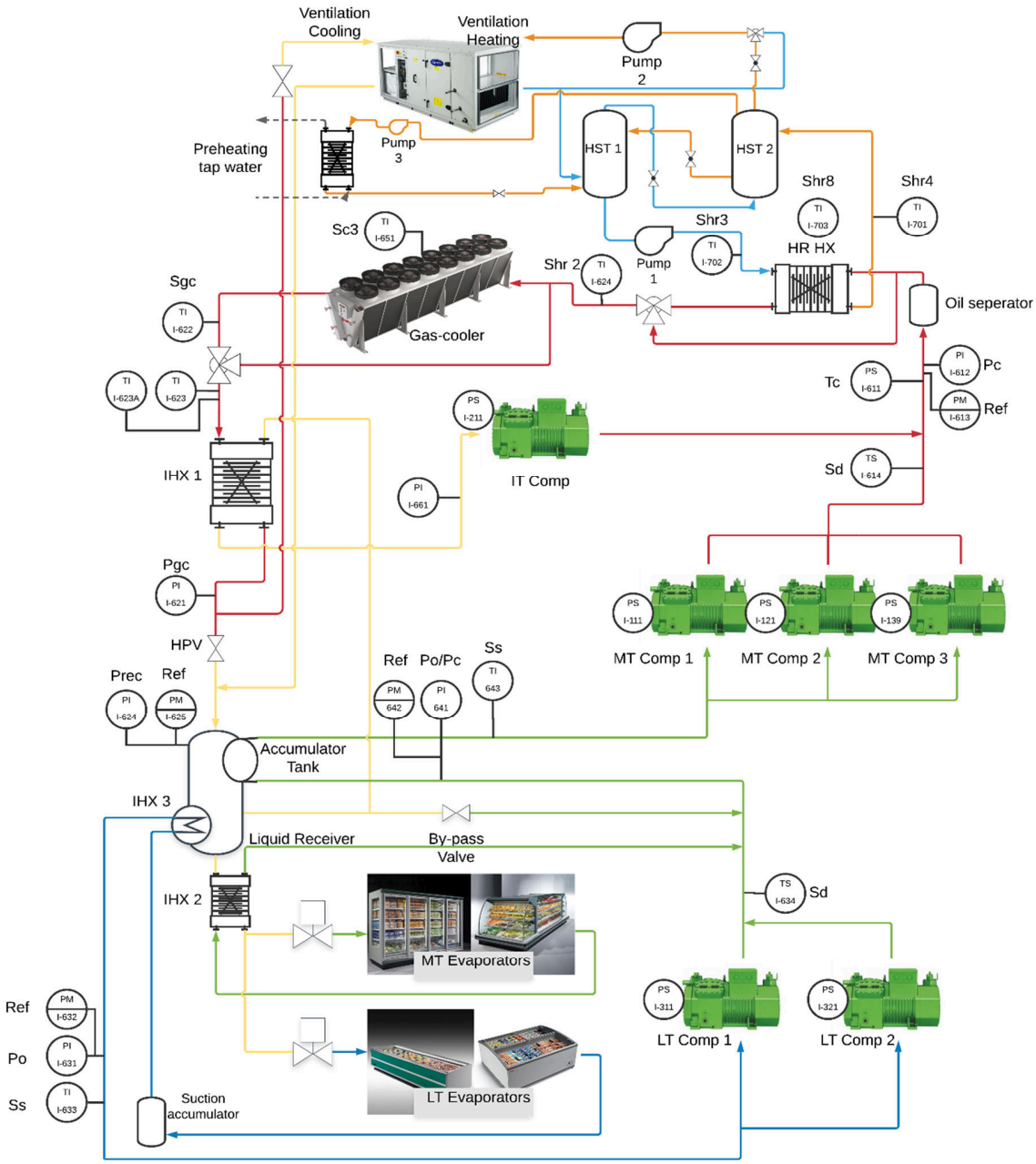


Figure 15: PI and sensor diagram Transcritical CO2 Booster Refrigeration System Spar Snarøya

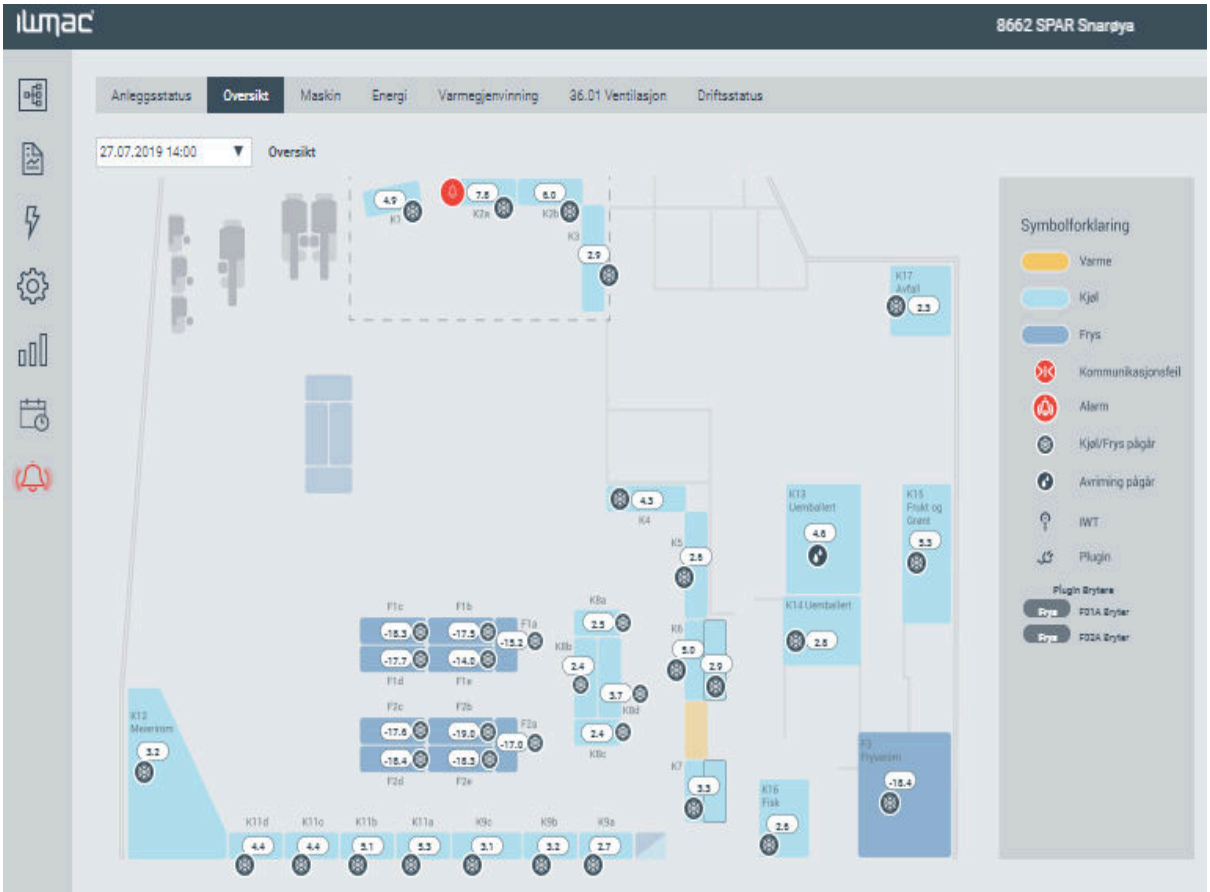


Figure 16: Screenshot picture of IWMAC>Overview cooling and freezing compartments [19]

3.4.1 Heat recovery – Spar Snarøya

The heat recovery (HR) system of Spar Snarøya is the only one of the three refrigeration systems with capability to store heat. The core of the heat recovery system is two heat storage tanks with a volume of 500 l each. Pure water circulates between the tanks and a heat ventilation battery at the air-handling unit and a heat exchanger transferring heat for pre-heating of tap water. Such a system has, in addition to internal heating, the opportunity to sell excess heat to the district heating utility network (DHU). The HR system is controlled by a regulator (Danfoss ECL Comfort 310), which offers the operator to monitor, adjust and control the operating conditions.

A sensor checks the temperature in HST 2 and sends a signal to the 3-way valve to open and enabling heat accumulated in the tanks to reach the heat battery at the AHU. Pump 1 is regulated by the head temperature, which is adjusted for ventilation heat demand. When the temperature in the tanks reaches the set-point temperature or when no more heat can be accumulated, a sensor in HST 1 sends a signal to M2 to close. When there is abundance of heat available, pump 2 will

activate and enable heat transfer for pre-heating of tap water (or selling heat to DHU if that is an option). When the heat demand exceeds the demand at normal operation, a 0-10 V signal is generated from the HST1 to AK-PC 781A which raises the pressure in the gas cooler. The step by step measures in the gas cooler for additional heat production is stopping the fans, enabling by-passing and finally having a false load operate as an evaporator.

The heat demand for at the AHU is very low or zero outside opening hours. The heat accumulates in the HSTs at these hours and contributes for an independent refrigeration system in terms of heat demand. This allows the compressors to run more stable as the high stage discharge pressure is mainly controlled by the cooling demand. Other benefits are a less mechanical fatigue on the compressors and lower noise level.

3.5 Heat recovery system overview

There are multiple solutions for heat recovery integrated in refrigeration systems, as described in chapter 2. As for the three refrigeration systems discussed in the previous sub-chapters, they have their own distinctive heat recovery set-up. A significant difference between the system of Kiwi Olsvik and the other two is the condensing unit. Whereas the more complex high stage pressure sections of Kiwi Tertnes and SPAR Snarøya includes a gas-cooler, Kiwi Olsvik's refrigeration system has a dry-cooler as a part of its heat recovery cycle. The heat exchanger model upstream of the compressor units is the same in each of the refrigeration systems (Swep B18), but unlike for the other two systems, it works as condenser rather than a de-superheater in the system of Kiwi Olsvik. Further description of the distinctive heat recovery cycles follows in the next chapters. Table 5 presents a technical summary for the heat recovery systems.

Table 5: Overview of technical installments in the transcritical CO_2 refrigeration booster systems

	Kiwi Olsvik	Kiwi Tertnes	Spar Snarøya
Number of heat exchangers	1	2	2
HR HX	SWEP B18Hx110 130 bar 35% ethylenglycol	SWEP B18Hx72 130 bar 35% ethylenglycol	SWEP B18Hx110 130 pure water
PC HX	-	SWEP B18Hx26	SWEP B18Hx26
Ventilation heating	Yes	Yes	Yes
DHW	No	No	Yes
Ventilation Cooling	No	Yes	Yes

4 Methods

Temperature and pressure sensors are installed in key areas of each refrigeration system. The measurements of the sensors are logged in web monitoring systems, allowing the user to inspect data from each sensor at any given time and look up previous measurements. Two different web monitoring systems are applied in collecting data for the three refrigeration systems evaluated in this thesis:

- ENVO; energy surveillance for Kiwi Olsvik and Kiwi Tertnes [20]
- IWMAC; energy surveillance for Spar Snarøya [19]

The systems share multiple similarities; an interface showing a PI-diagram of the machine where key sensors are visible, an interface of the ventilation system and an interface of the floor plan. Both systems have interfaces of the energy consumption, with slightly different visible parameters. All systems have the same technical entrepreneur, hence the technical similarity between all the refrigeration systems. The data are collected and processed in Microsoft Office Excel, see next chapter.

4.1 Sensors and data collection

The sensors in the refrigeration systems are mainly pressure and temperature sensors. There are three types of pressure sensors: pressure transmitter (PI), manometer (PM) and pressure switch (PS). The pressure transmitters and the manometers are connected. The saturation pressure and the reference pressure of the CO_2 are collected by these two sensors. The PS sensor acts as safety installment by measuring the pressure in the compressors and allowing the compressors to be shut down when the pressure exceed the pressure limit. Each compressor's running capacity is continuously logged, as are other non-pressure or non-temperature data such as the opening degree of the flash gas by-pass valve and the high-pressure valve. The temperature sensors are either of the type AKS 11 (TI) or AKS 21A (TS). The suction and the discharge gas temperatures are measured by one or the other depending on which temperature range the sensor is operating in. All temperature, pressure and valve sensors in the refrigeration systems are listed in Table 6.

Table 6: An overview over temperature, pressure and valve sensors present in the three case supermarkets evaluated in this thesis

Sensor name	Purpose	PID tag number Olsvik	PID tag number Tertnes	PID tag number Snarøya	Technical description	Operation range [°C/bar]
Ss - LT	Suction gas temperature LT compressors	TI I-633	TI I-633	TI I-633	Temperature sensor, AKS 11	-50 – 100
Po – LT	Saturation pressure corresponding to the LT suction pressure	PI I-631	PI I-631	PI I-631	Pressure transmitter, MBS 8250	-1 – 159
Pref – LT	Reference LT saturation pressure	PM I-632	PM I-632	PM I-632	Manometer	70
Sd - LT	Discharge temperature LT compressors	TS I-634	TS I-634	TS I-634	Temperature sensor, AKS 11	-50 – 100
Cap Comp 1	Pressure switch LT comp. 1	PS I-319	PS I-311	PS I-311	Pressure switch, Emerson CS3-WPS 46 bar	46
Cap Comp 2	Pressure switch LT comp. 2	PS I-329	PS I-321	PS I-321	Pressure switch, Emerson CS3-WPS 46 bar	46
Po/Pc – MT	Saturation pressure corresponding to the MT suction pressure	PI I-641	PI I-641	PI I-641	Pressure transmitter, MBS 8250	-1 – 159
Pref – MT	Reference MT saturation pressure	PM I-642	PM I-642	PM I-642	Manometer	70
Ss - MT	Suction gas temperature MT compressors	TI I-643	TI I-643	TI I-643	Temperature sensor, AKS 11	-50 – 100
Cap Comp 1	Pressure switch MT comp. 1	PS I-119	PS I-111	PS I-111	Pressure switch, Emerson CS3-S8S 130 bar	130
Cap Comp 2	Pressure switch MT comp. 2	PS I-129	PS I-121	PS I-121	Pressure switch, Emerson CS3-S8S 130 bar	130
Cap Comp 3	Pressure switch MT comp. 3	PS I-139	PS I-131	PS I-139	Pressure switch, Emerson CS3-S8S 130 bar	130
Sd - MT	Discharge temperature MT compressors	TS I-614	TS I-614	TS I-614	Temperature sensor, AKS 21A	0-150
Vrec OD	By-pass valve opening degree	V-630	V-630	V-630	Gas by-pass valve CCM 20	90
Cap PC	Pressure switch PC	-	PS I-211	PS I-211	Pressure switch, Emerson CS3-S8S 130 bar	130
P-limiter	Pressure switch	PS I-611	PS I-611	PS I-611	Pressure switch, Emerson CS3-WQS	108
Pc – MT	Discharge pressure MT compressors	PS I-612	PS I-612	PI I-612	Pressure transmitter, MBS 8250	-1 – 159
Pref - MT	Discharge reference pressure MT compressors	PM I-613	PM I-613	PM I-613	Manometer	160

Pgc	Pressure gas cooler	PI I-621	PI I-621	PI I-621	Pressure transmitter, MBS 8250	-1 – 159
P - Rec	Pressure receiver	PI I-624	PI I-624	PI I-624	Pressure transmitter, MBS 8250	-1 – 159
Pref - Rec	Reference pressure receiver	PM I-625	PM I-625	PM I-625	Manometer	100
Vhp OD	High pressure valve opening degree	V-622A	V-622A	V-622A	High pressure control valve, CCMT 8.	140 bar - 40/+60°C

The sensors in the heat recovery circuits are mainly TI-sensors. The sensors measuring the inlet and outlet temperature at the HR HX are the key sensors. Another important device is the sensor keeping track of the on/off -status of the three-way valve at heat recovery heat exchanger at the refrigeration systems of Kiwi Tertnes and Spar Snarøya. The sensors in the heat recovery circuits are listed in table 7 below. Other important logged data which are not included with sensor names, such as the gas cooler running capacity, pump flow switch and pump speed, are also included in table 7.

Table 7: Sensors in each of the case supermarket's heat recovery circuit

Heat Recovery								
Kiwi Olsvik			**Kiwi Tertnes**			**Spar Snarøya**		
Sensor name	PID tag	Description	Sensor name	PID tag	Description	Sensor name	PID tag	Description
Sgc	TI I-622	Temp. outlet HRHX	V3hr	V-619	Three-way valve	V3hr	V-619	Three-way valve
Sc3/S7	TI I-651	Temp. inlet HR HX (cold side)	Shr2	TI I-624	Temp. outlet HR HX (hot side)	Shr2	TI I-624	Temp. outlet HR HX (hot side)
Tur VGV	-	Temp. inlet Heating Battery	Sgc	TI I-622	Temp. outlet GC	Sgc	TI I-622	Temp. outlet GC
Retur VGV	-	Temp. outlet Heating Battery	Sc3	TI I-651	Temp. GC	Sc3	TI I-651	Temp. GC
Tot. Kap %	-	Drycooler capacity	Shr3	TI I-702	Temp. inlet HR HX (cold side)	Shr3	TI I-702	Temp. inlet HR HX (cold side)
			Shr4	TI I-701	Temp. outlet HR HX (cold side)	Shr4	TI I-701	Temp. outlet HR HX (cold side)
			Shr8	TI I-703	Extra sensor for brine	Shr8	TI I-703	Extra sensor for brine
			HR Pump. Speed	-	HR Pump energy use	Run Cap %	-	Running capacity Condenser/Gas Cooler
			Cond. Running Cap	-	Running capacity Condenser/Gas cooler	V3gc%	-	Valve opening three-way valve Gas Cooler
			Akv	-	Opening degree of Cooling Battery Valve			
			Te	-	Exit temperature Cooling Battery			

The data in the web monitoring system of ENVO is collected for every fifth minute during the span of the relevant time of evaluation. The corresponding data of IWMAC is recorded when the value of a parameter changes. This implies that the point of time when each parameter is logged will vary, and for very stable parameters the time between each logging can be substantially different compared with unstable parameters. The data are collected and processed in Microsoft Office Excel. Extensive work of coding in Microsoft Excel Visual Basic Applications (VBA) was performed during period of the data evaluation in order to synchronize the parameters. The modules for arranging the data are attached in Appendix 1. In order to use built-in function in Excel, the timestamp for each parameter is first categorized and secondly averaged for fixed time intervals. The third step is organising the data to which RnLib-functions can efficiently perform thermodynamic calculations in order to evaluate each system in terms of energy efficiency.

The collected data are from the coldest and warmest period in 2019, as described in Chapter 3. The minimum and maximum three-days period will highlight the operation mode differences between the systems, which is the main topic to evaluate in this thesis. The average of each parameter of interest is calculated for 20 minutes intervals. This means a set of 216 points of data for each parameter for each three-days minimum and maximum mean temperature period. A combined PI and sensor diagram of the three refrigeration systems are presented in Fig. 17. For a complete PI and sensor diagram for each refrigeration system see Fig. 14, 15 and 17 in Sections 3.1-3.3 respectively. The evaporator outlet temperature sensors of the cooling and freezing compartments are not included. More installed sensors could ideally track and collect important non-present data, as discussed in Further Work in Chapter 7. The method of dealing with lack of data together with the method of calculating the refrigeration systems' performances is presented in the next chapter. An excerpt of the excel calculation is found in Appendix 4.

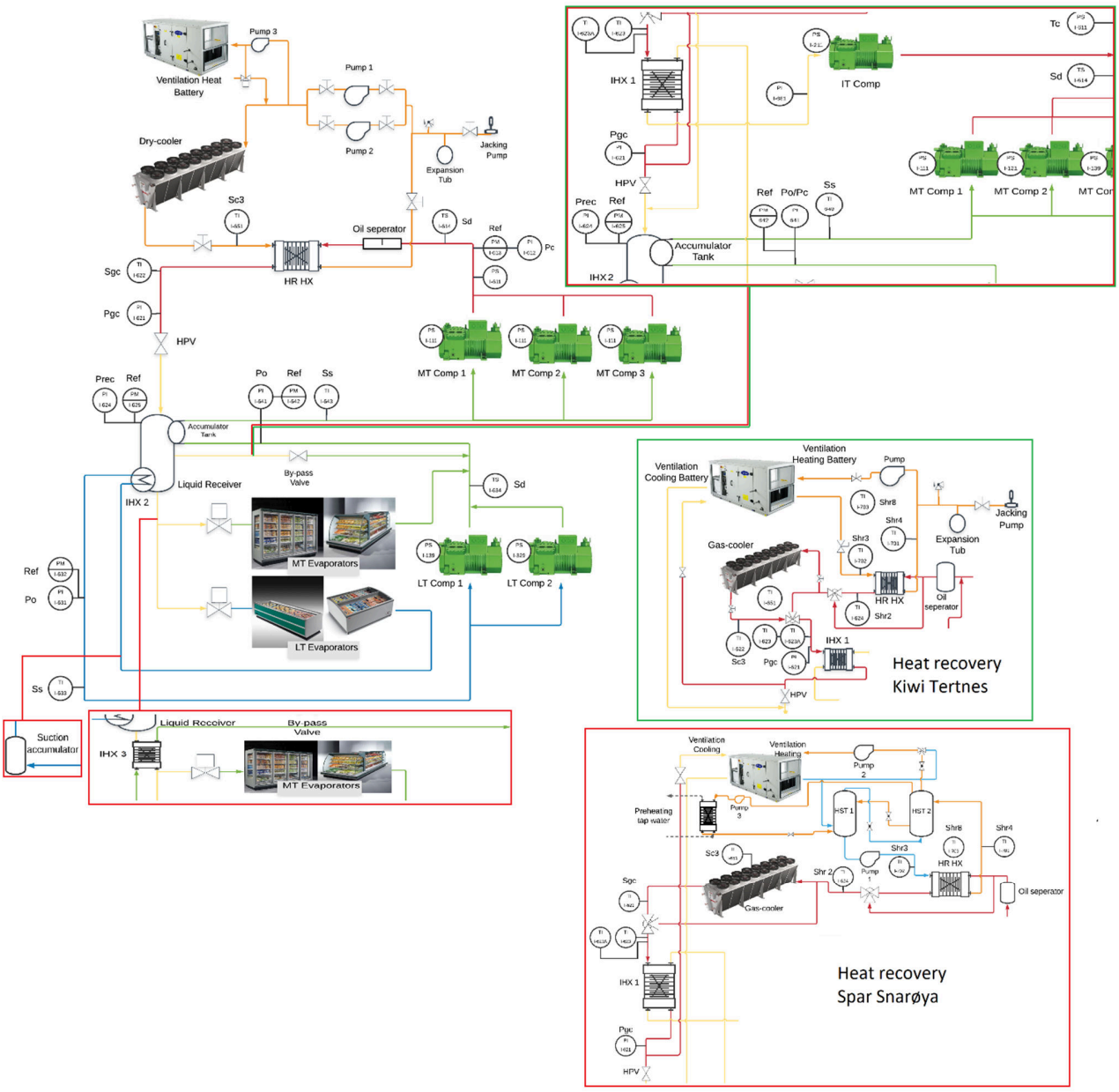


Figure 17: Combined PI and sensor diagram for Kiwi Olsvik, Kiwi Tertnes and Spar Snarøya. The refrigeration system of Kiwi Olsvik, Kiwi Tertnes (framed in green) and Spar Snarøya (framed in red)

4.2 Energy efficiency calculations

The focus of the analysis is comparing the energy performances of the systems. The coefficients of performances (COP) of any refrigeration cycle is the ratio of the refrigeration effect to the net work input required to achieve that effect [21]. To put it simply, COP tells how energy efficient a refrigeration system is. The total refrigeration COP_{ref} is the ratio of total refrigeration effect to the total electricity used for providing the refrigeration:

$$COP_{ref} = \frac{\dot{Q}_{LT} + \dot{Q}_{MT}}{\dot{E}_{tot,refrigeration}} \tag{1}$$

The electrical power $\dot{E}$ is continuously logged in the web monitoring interfaces. The cooling effects are obtained by deduction of the control volume energy rate balance:

$$\frac{dE_{cv}}{dt} = \dot{Q}_{cv} - \dot{W}_{cv} + \dot{m}\left(u_t + pv + \frac{V^2}{2} + gz\right)_{in} - \dot{m}\left(u_t + pv + \frac{V^2}{2} + gz\right)_{out} \tag{2}$$

The subscript "cv" means each term is evaluated over the boundary of a control volume. As the system is at steady state, i.e. all properties remain the same in time, the total amount of energy present remains constant; $dE_{\mathrm{cv}}/dt = 0$. The subscript "t" of the internal energy, u, means that u_{t} refers to the thermal energy of the compound evaluated. Thermal energy in study of heat transfer is referred to as the sum of the sensible and latent internal energy, denoted as u_{sens} and u_{lat} respectively. Specific enthalpy, h, is defined as:

$$h = u + pv \tag{3}$$

The term enthalpy is brought in solely as a convenience and is very useful in heat transfer calculations [21]. The product of the specific pressure p and the specific volume v is, in relation with the mass flow $\dot{m}$ across the inlet and outlet of the control volume, the flow work. The remainder of the net rate of work is $\dot{W}_{cv}$. By assuming no additional work effects on the evaporators other than the flow work, and neglectable changes in potential and kinetic energy, Equation 2 reduces to:

$$\dot{Q}_{cv} = \dot{m} \cdot (h_{out} - h_{in}) \tag{4}$$

$\dot{Q}_{cv}$ is the net rate of energy transfer by heat across the boundary of the control volume. For evaporators that means the heat transfer from the refrigerated space. As the throttling process downstream of the separator tank is isenthalpic due to the assumptions made, the specific enthalpies of the CO_2 at the LT and the MT evaporator inlets are known for the systems with DX evaporation without an internal heat exchanger prior to the evaporation process. The refrigeration system at Spar Snarøya is equipped with such a heat exchanger. The liquid is throttled to the MT and LT evaporation pressure after transferring heat to the colder low-pressure superheated CO_2 coming from the MT evaporators. There are no temperature sensors between the MT evaporators and IHX 3 or between IHX 3 and

the suction line for the MT compressors, see the red framed square second from the left at the bottom in Fig. 17. An indirect approach to estimate the enthalpy difference across the IHX 3 is needed to calculate the cooling load at the LT and MT evaporators in the refrigeration system of Spar Snarøya.

4.2.1 Internal superheating and the enthalpy change across the IHX 3 estimate

The outlet temperature of the LT and MT evaporators are continuously measured and acquired from IWMAC. The inlet temperatures are however not. By assuming no heat loss to the surroundings of the energy transfer in IHX 3, the enthalpy changes at the hot (saturated liquid CO_2 from receiver) and cold (superheated CO_2 from the MT evaporators) side are to be considered as linearly dependent to the mass flow[b]. The state of the refrigerant is known both for the cold and hot inlet flow, but this is insufficient for calculating the rate of heat transfer in the IHX 3.

The liquid flow exiting the separator is subcooled in IHX 3 before being throttled to the LT and MT pressure level respectively. The implementation of an extra internal heat exchanger adds to the cooling capacity of the system.

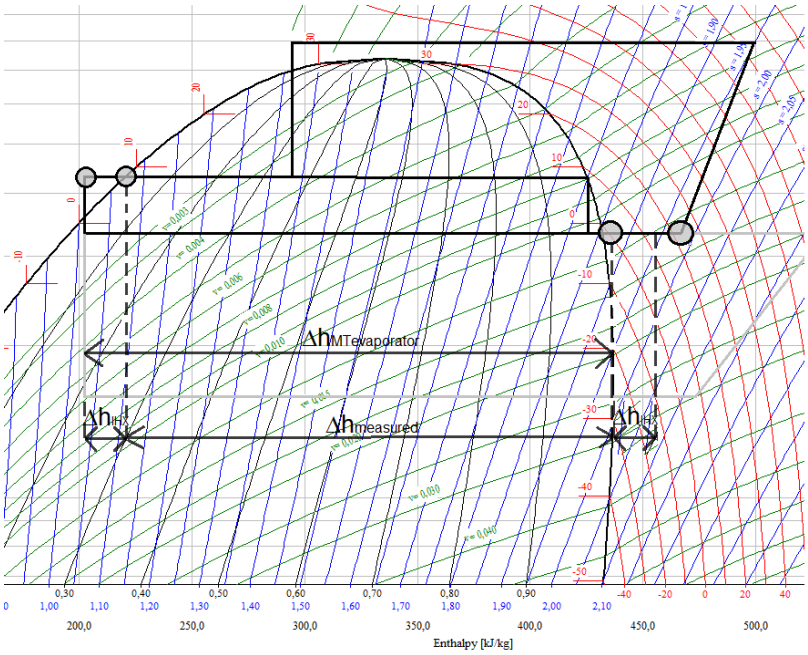


Figure 18: Log P-H-diagram of the refrigeration cycles at Spar Snarøya with state points before and after IHX3

[b] The enthalpy differences marked in Figure 18 are misleading, as they are not equal due to different mass flows on cold and hot side of the IHX 3

While the other two refrigeration systems other than Spar Snarøya have no extra heat exchanger at this part of the refrigeration circuit, their numbers are used to calculate the effect of the IHX on extra superheating in the system of Spar Snarøya. As can be observed in Fig. 19, the variation of the *Total Superheat* is abrupt and there is no significant correlation between the MT evaporator exit temperature and the MT compressor suction line inlet temperature at any of the refrigeration systems.

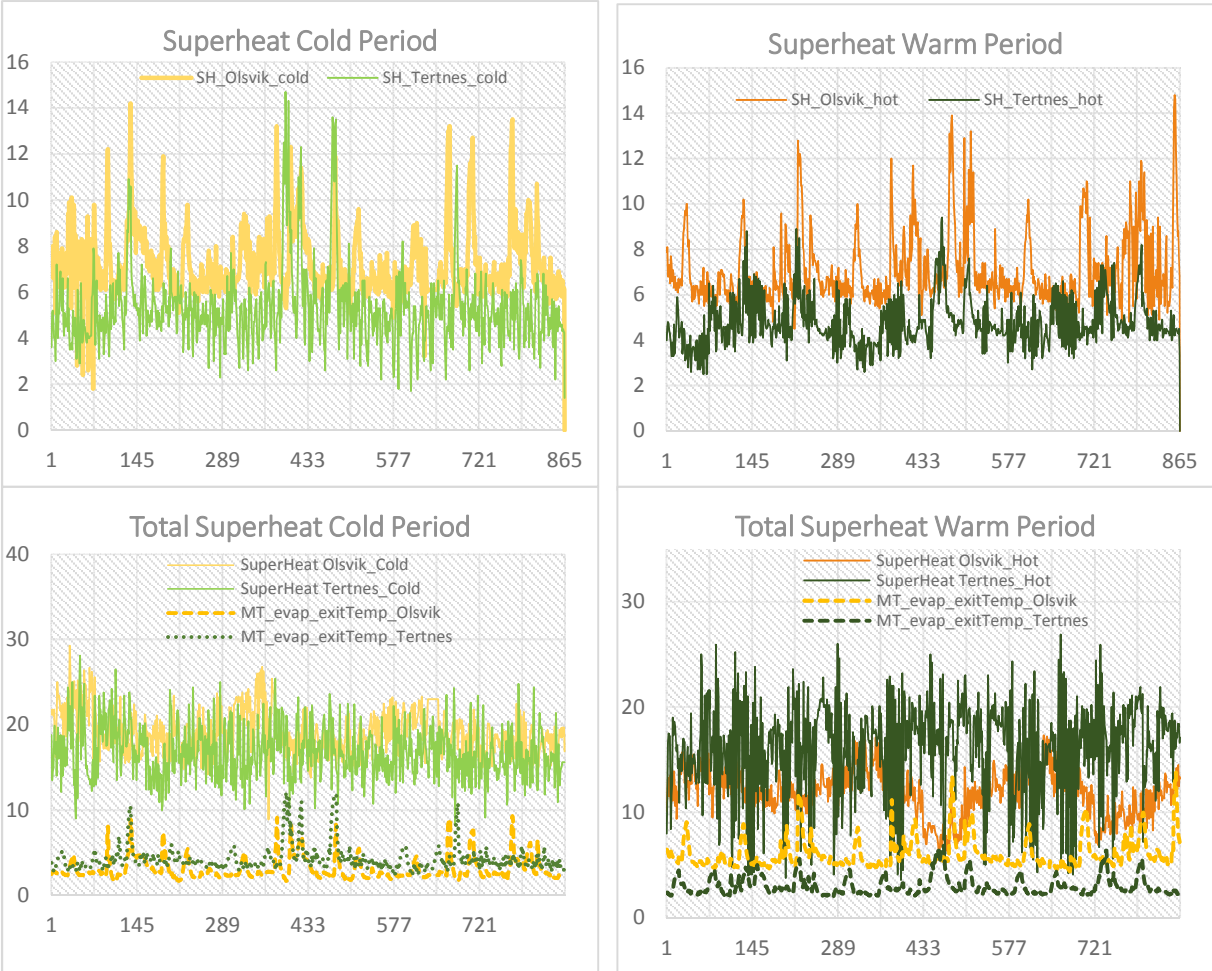


Figure 19: Graphic presentation of the Superheat and the Total Superheat at the warm and cold period in 2019

The evaporator exit temperature is more stable as the ambient temperature at the cooling compartments changes at a slow rate and the cooling workload is rather stable, while the suction line temperature varies along with the compressor's various workload. From the bottom right graph in Fig. 19 it is observed that the MT evaporator exit temperature at Olsvik during the warm period has to a certain degree notable similar fluctuation as the total superheat in the same system. The average temperature difference between the MT compressor suction line MT temperature and the exit temperature of the MT evaporators at the system of Kiwi Olsvik during the warm period is used to estimate the external superheat in the MT

region of Spar Snarøya refrigeration system. The estimate of the enthalpy difference across the IHX 3 is given by the following equations:

$$\Delta h_{IHX3,cold\ side} \cong h(t_{suction\ line\ comp} - \bar{\tau}) \ - h(t_{exit\ evap}) \tag{5}$$

$$\Delta h_{IHX3,hot\ side} \cong \left(h(t_{suction\ line\ comp} - \bar{\tau}) \ - h(t_{exit\ evap})\right) \cdot \frac{\dot{m}_{MT,evap}}{\dot{m}_{liq}} \tag{6}$$

$\bar{\tau}$ is the average temperature difference between the MT compressor suction line inlet temperature and the exit temperature of the MT evaporators at the system of Kiwi Olsvik during the warm period, and is calculated to be approximately 5 K. The enthalpy difference across IHX 3 is calculated for every state of interest by using RnLib. The mass flow of the refrigerant in the MT evaporator and the liquid CO_2 mass flow leaving the separator are denoted as $\dot{m}_{MT,evap}$ and $\dot{m}_{liq}$ respectively. The calculation behind these estimates are presented in the next section.

The outlet temperature of MT evaporators varies, and this is taken into consideration when calculating a common outlet temperature for all the MT evaporators. As described in Chapter 3.1, Spar Snarøya and the other supermarkets have five kinds of low temperature storage compartments; cooling cabinets, cooling rooms, freezing cabinets, freezing rooms and fruit and vegetables cabinets. The temperature of the outlet CO_2 is within a range of the set-point temperature of the storage compartment and will be in different temperature ranges based on in which compartment the evaporator operates. Because the mass flow rate is measured in the pipe systems leading the CO_2 from the evaporators as a whole section and not measured individually or for small groups of evaporators, an average internal superheat is calculated for the whole MT and LT evaporator section. The average evaporator exit temperature is calculated by using the following equations:

$$\bar{t}_{out,LT} = \frac{a \cdot \bar{t}_{-21°C\ to\ -18°C}}{a} \tag{7}$$

$$\bar{t}_{out,MT} = \frac{b \cdot \bar{t}_{1°C\ to\ 3°C} + c \cdot \bar{t}_{3°C\ to\ 6°C}}{b + c} \tag{8}$$

The letters a, b and c represent the number of evaporators with an average CO_2 outlet temperature within the temperature range indicated by the subscripts of the average temperatures $\bar{t}$. One evaporator unit is selected to represent each of the temperature ranges. The comprehensive work of calculating the average internal superheat based on all evaporators at each time step of measurement is out of scope for this thesis. For practical reasons only one evaporator for each temperature range is chosen. Two MT evaporators could not be used as reference

MT evaporators due to misleading measurement, see Chapter 3.4. The reference MT evaporators were chosen based on amount of data points available.

4.2.2 Mass flow estimate

Mass flow meters are not installed in any of the supermarket refrigeration systems. There are certain ways of calculating the mass flow, depending on which data are available. The electrical power for each compressor is continuously measured and logged in the web monitor systems. The specific enthalpy, *h*, is for every superheated state of the refrigerant (i.e. suction and discharge gas state) dependent on temperature and pressure. The pressure is given by the saturation pressure value of the temperature sensors Ss or Sd and is measured by the sensor named P_O or P_C (see Table 6). By using a set of external Excel VBA (Excel Visual Basic Application) functions the specific enthalpy is found for each suction and discharge gas state. The mass flow across the LT and MT evaporators are measured and accessible at IWMAC for the system of Spar Snarøya, however for the other two systems the mass flow is calculated by using the total efficiency method.

$$\dot{E} = \dot{m}_{ref} \cdot \Delta h_{comp} = \frac{\dot{m}_{ref} \cdot (h_{discharge,is} - h_{inlet})}{\eta_{is}} \tag{9}$$

By rearranging Eq. 9 it is evident that finding the mass flow rates over the compressors, $\dot{m}_{ref}$, is straight forward when the compressor power and the enthalpy change are known variables. As Fig. 20 below shows, the relations of the mass flows are:

$$\dot{m}_{tot} = \dot{m}_{LT,evap} + \dot{m}_{MT,evap} + \dot{m}_{flash\ gas} \tag{10}$$

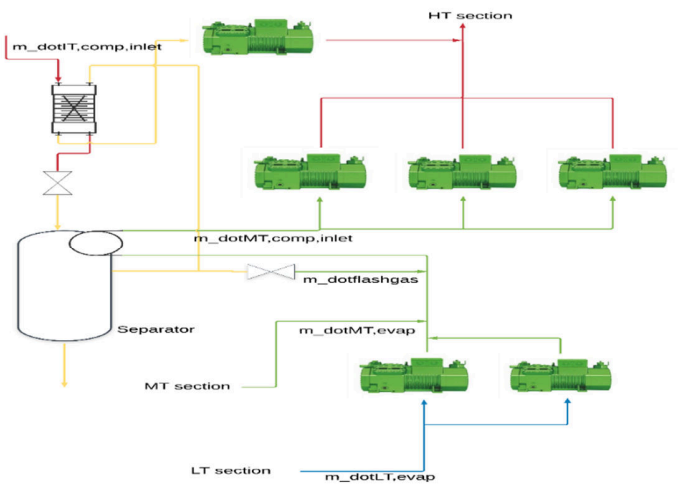


Figure 20: Mass flows relation CO_2 booster refrigeration system with parallel compressor

The mass flow across the LT evaporator is equal to the mass flow across the LT compressors. The two unknown fractions of the total mass flow at this point are the

mass flow of the flash gas escaping the separator and the mass flow across the MT evaporator.

At the system of Spar Snarøya, both the LT and MT mass flows are provided by IWMAC. The other systems are limited by having no such available information. The mass flows across the LT and MT compressors are calculated by using Eq. 9. There are two unknown variables in Eq. 10 in the cases of the refrigeration systems of Kiwi Olsvik and Kiwi Tertnes. By expressing the mass flow of the flash gas as a function of the portion of vapour leaving the separator, a solvable set of equations is obtained. The relation of vapour leaving the separator is estimated in Equation 11 below by assuming conservation of mass in the tank at the time of calculation:

$$\dot{m}_{inlet,rec} = \dot{m}_{outlet,rec} \rightarrow \dot{m}_{SHP} = \dot{m}_{flash\ gas} + \dot{m}_{liquid} \tag{11}$$

The subscript *SHP* indicates that this mass flow is calculated at the location of the inlet of the HPV. This sensor measures the temperature of the refrigerant before the throttling process prior to the separator tank. The enthalpy in this state is calculated and is used to calculate the ratio of mass of vapour to the total mass of the mixture in the separator, defined in Eq. 12:

$$x = \frac{m_{vapour}}{m_{liquid} + m_{vapour}} \tag{12}$$

This relation determines how much mass of each state leaving the separator in this simplistic calculation of $\dot{m}_{flash\ gas}$ and $\dot{m}_{liquid}$. The equilibrium of the liquid and gas changes with change of pressure, and the pressure determines how much liquid and flash gas are leaving the separator. The calculations are made by taking 20-minutes intervals of a process that is transient, which leaves rooms for errors in the calculation.

Equations 11 and 12 are rearranged to Eq. 13 and 14 respectively:

$$x = \frac{m_{vapour}}{m_{liquid} + m_{vapour}} \rightarrow \dot{m}_{liquid} = \dot{m}_{flash\ gas} \cdot \left(\frac{1}{x} - 1\right) \tag{13}$$

$$\dot{m}_{SHP} = \dot{m}_{flash\ gas} + \dot{m}_{liquid} \rightarrow \dot{m}_{liquid} = \dot{m}_{SHP} - \dot{m}_{flash\ gas} \tag{14}$$

Eq. 15 is found by inserting Eq. 13 into Eq. 14

$$\dot{m}_{flash\ gas} = x \cdot \dot{m}_{SHP} \tag{15}$$

$\dot{m}_{MT,evap}$ is found by calculating $\dot{m}_{liquid}$:

$$\dot{m}_{liquid} = \dot{m}_{LT,evap} + \dot{m}_{MT,evap} \tag{16}$$

While the parallel compressor in the systems of Kiwi Tertnes and Spar Snarøya is running, typically at temperatures above 15°C [18], flash gas is compressed in the PC compressor rather than in the MT compressors. As mentioned earlier, there is no sensor attached to the suction line of the parallel compressor. The heat load in the gas cooler is calculated based on measured enthalpy change and the high-pressure mass flow. This mass flow equals to the mass flow over the MT compressor at conventional refrigeration modus, but in parallel compression mode mass flow of the PC contributes to a higher mass flow rate. Instead of having an expensive pump system feeding the integrated air condition cooling battery, the necessary refrigerant is provided by letting the AC valve controlling high pressure side CO_2 downstream of the gas-cooler and the internal heat exchanger, IHX 1 in Fig. 14 and Fig. 15. The PC mass flow is the sum of the flash gas mass flow and the AC cooling mass flow:

$$\dot{m}_{PC} = \dot{m}_{flash\ gas} + \dot{m}_{AC} \tag{17}$$

A Bitzer web software was used to estimate the $\dot{m}_{PC}$. The input data are measured data for the system of Kiwi Tertnes. The power of the compressor is logged, and by adjusting the frequency input in the web software, an estimate of the mass flow of the CO_2 is obtained at the point where the measured and the simulated compressor power of the simulation are close to equal. Data sheets from the calculation of one simulation is in Appendix C. Linear regression was used for finding the correlation between the compressor power and the CO_2 mass flow.

4.2.3 Calculation of coefficients of performances

The methodological approach in receiving and organizing the data for the calculation of the COPs of the three transcritical CO_2 refrigeration systems; Kiwi Olsvik, Kiwi Tertnes and Spar Snarøya, is described in the two previous sections. This section deals with the calculations itself. All three systems are two-stage compression systems with two connected pressure levels in the refrigeration section, which complicates the calculations of the COPs compared with conventional one-stage compression refrigeration systems.

There are two ambient air temperature conditions evaluated for each refrigeration system, warm ambient temperature conditions (~20°C plus) and cold ambient temperature conditions (~0°C). The main difference between these two weather conditions are the magnitude of the heat rejection load. When the ambient temperatures are above 25°C, the system operates transcritical without any significant heat load stored to the heat recovery cycle or utilized for AC heating or floor heating. The systems' total refrigeration COP, designated COP_{ref}, is therefore the main indicator for evaluating the systems against each other during warm periods. During the colder ambient temperature conditions, the main indicator for evaluating the systems is the total system COP_{tot}. This section presents the definitions of the COPs of interest and the wider explanation for the calculations behind them.

The definition of COP_{tot} is given by equation 18:

$$COP_{tot} = \frac{\dot{Q}_{LT} + \dot{Q}_{MT} + \dot{Q}_{AC} + \dot{Q}_{HR}}{\dot{E}_{LT} + \dot{E}_{MT} + \dot{E}_{PC} + \dot{E}_{fan}} \tag{18}$$

Data of the electrical power of the IT compressor in the system of Spar Snarøya is unfortunately not available. This data is available for the other system with a parallel compressor unit, the system of Kiwi Tertnes. There is not enough data to quantify $\dot{Q}_{AC}$ for the system of Spar Snarøya. The systems using the web interface monitor system ENVO have available data of the electrical power of the dry cooler fans, and the pumps of the heat recovery cycle in the case of Kiwi Olsvik. $\dot{E}_{fan}$ for the system of Spar Snarøya is calculated by using the data on the total running capacity of the fans and the information regarding the electricity power of fans, see Appendix B for the data sheet. Another approach would be to estimate the power of the gas cooler to be 3% of the heat rejected in the gas cooler of the system. This estimate is reliable according to a major CO_2 gas cooler manufacturer [22]. The system of Spar Snarøya is evaluated as a system without parallel compressor unit due to this non-available data. This means that the system of Kiwi Tertnes is the only system with a parallel compressor included in the calculation of COP_{tot}.

Another important performance factor when evaluating a refrigeration system is clearly the total refrigeration COP_{ref} and the equation and definition is present in Equation 1. During the warm period neither of the systems have any heat demand for the heat storage cycles/heat rejection exploitation. When using COP_{ref} to compare the systems against each other, only the data from the warm period is used. At periods where the refrigeration systems provide heat loads for the heat exploiting installations, using minimum floating condensing pressure is a method of calculating the COP_{ref} [22].

The total cooling load $\dot{Q}_{cooling}$ [kW] includes, unlike the cooling load for the calculation of COP_{tot}, the non-useful cooling of the CO_2 in the refrigeration system. This factor is important for calculating the COP_{LT} and COP_{MT} for systems calculated without parallel compression, and the relation between the equations are as the following:

$$\dot{Q}_{cooling} = \left(\dot{Q}_{LT} + \dot{Q}_{LT,xSH}\right) + \left(\dot{Q}_{MT} + \dot{Q}_{MT,xSH}\right) + \dot{Q}_{AC} \tag{19}$$

$$COP_{cooling} = \frac{\dot{Q}_{cooling}}{\left(\dot{E}_{MT} + \dot{E}_{fan}\right)} \tag{20}$$

$$COP_{MT} = \frac{\dot{Q}_{MT}}{\left[\left(\dot{Q}_{MT} + \dot{Q}_{MT,xSH}\right)/COP_{cooling}\right]} \tag{21}$$

$$COP_{LT} = \frac{\dot{Q}_{LT}}{\left[\dot{E}_{LT} + \left(\dot{Q}_{LT} + \dot{Q}_{LT,xSH}\right)/COP_{cooling}\right]} \tag{22}$$

Equations 19-22 applies for the systems without parallel compression. This includes the system of Spar Snarøya due to the missing data for the parallel compressor and the AC cooling, and for all systems during the cold period where the cooling demand for AC is zero.

For the calculation of COP_{ref}, COP_{MT} and COP_{LT} the Equations 19-22 must be adapted to include only the fraction of the compressor power $\dot{E}_{PC}$ used for AC cooling. An estimate of $\dot{E}_{PC}$ is obtained by using the compressor producer's web software, as explained in section 4.2.2. By assuming the enthalpy difference over the AC valve to be equal the enthalpy difference over HPV, the $\dot{Q}_{AC}$ is estimated according to Eq. 4.

Another method of estimating $\dot{Q}_{AC}$ would be to evaluate the hot side of the cooling battery. This cooling load is a function of the supply air flow, the temperature difference over the cooling coil, the specific heat capacities of the moist air and the relative humidity. Because the relative humidity ratio is a rather complex calculation with a missing parameter in relative humidity, this method was rejected.

After all parameters are collected and organized it is evident that the comparison between the systems is based on comparing the COP_{tot} during the cold period and on a broader scale of energy performances during the warm period. The next chapter consists of the results of the evaluation of the refrigeration system of the three supermarkets of interest in this thesis.

5 Results

Each refrigeration system evaluated in this thesis is different when it comes to technical solutions and this affects how the COPs are calculated. Each refrigeration system is firstly evaluated for its performances in specific sections. The comparison between the systems regarding their energy efficiency performances is evaluated and discussed in chapter 5.

5.1 Kiwi Olsvik

The heat recovery heat exchanger works as the condenser/gas cooler as described in chapter 3.2. The ratio of heat load utilized in the heating battery of the AHU to the heat rejected in the HR HX is estimated to be:

$$\frac{\dot{Q}_{heat\_vent}}{\dot{Q}_{HR\_HX,cold\ side}} \approx \frac{T_{inlet,heat\_vent} - T_{outlet,heat\ vent}}{T_{outlet,HRHX,cold\ side} - T_{inlet,HRHX,cold\ side}} \tag{23}$$

By assuming no heat loss in the heat exchanger, $\dot{Q}_{HR\_HX}$ equals $\dot{Q}_{HR\_HX,cold\ side}$. At high ambient temperatures, the brine of the heat recovery cycle will by-pass the heat ventilation battery. At cold ambient temperatures, the opposite happens. The return temperature of the heating battery is in fact some degrees higher than the return temperature of the dry-cooler due to internal heating in the pipes, and the ratio of $\dot{Q}_{heat\_vent}$ to $\dot{Q}_{HR\_HX}$ is equal to 1.

Equation (18) is specifically for Kiwi Olsvik during the cold period:

$$COP_{tot} = \frac{\dot{Q}_{LT} + \dot{Q}_{MT} + \dot{Q}_{heat\_vent}}{\dot{E}_{LT} + \dot{E}_{MT} + \dot{E}_{fan,drycooler} + \dot{E}_{drycooler,pump}}$$

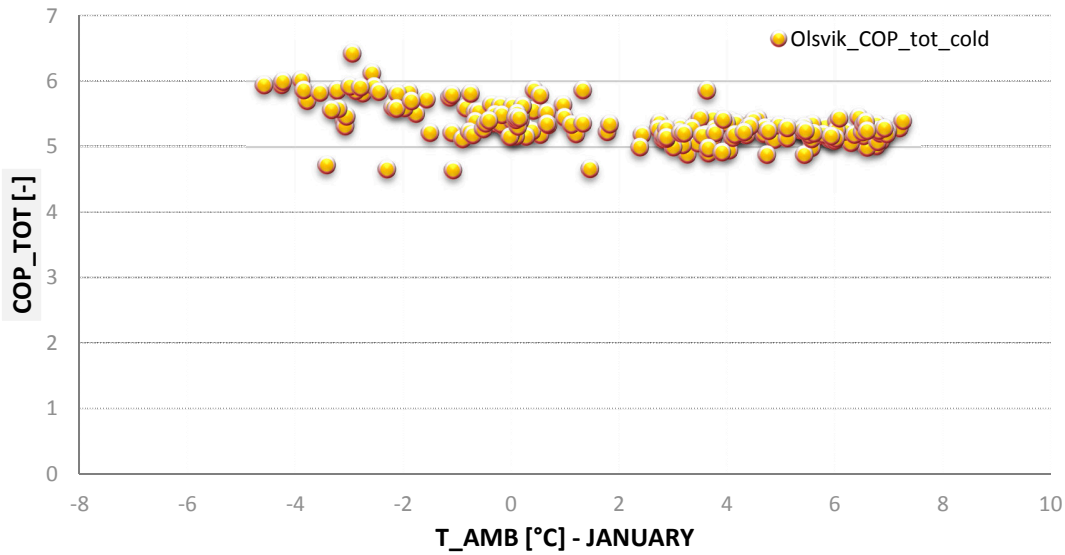
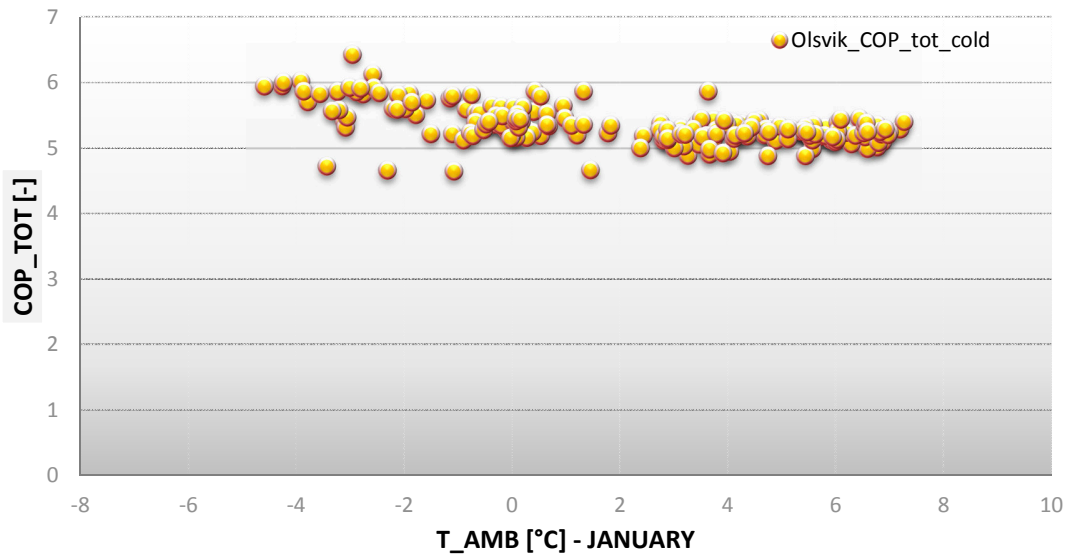


Figure 21: 20-min averaged total system COP for Kiwi Olsvik in the period of 28.01-30.01 2019



The dry cooler never runs during the evaluated cold period, which means that $\dot{E}_{fan,drycooler}$ is zero. Figure 21 presents the COP_{tot} of the transcritical CO_2 refrigeration booster system:

As seen in Figure 22, the average heating and cooling loads have little variation and the smaller columns are during midnight and 6 am upon where the supermarket is closed. The flat trend that can be observed of the COP_{tot} values in Figure 22 is closely related to these small variations. The only power used for operating the

refrigeration cycle is the LT and MT compressor units and the heat recovery cycle pumps, as the dry-cooler fans are off. P_{gc} is rather low during cold periods, 57-69 bar, and the high-pressure is within subcritical pressure region. Another parameter indicating a high COP_{tot} is the HR HX exit temperature. By assuming insignificant external heating of the cooled CO_2, the inlet temperature of the HPV and the exit temperature of the HX HR are equal. This temperature (*Sgc* in Table 7) is stable around 25°C during the supermarkets opening hours. These factors explain the high COP_{tot} during low temperatures.

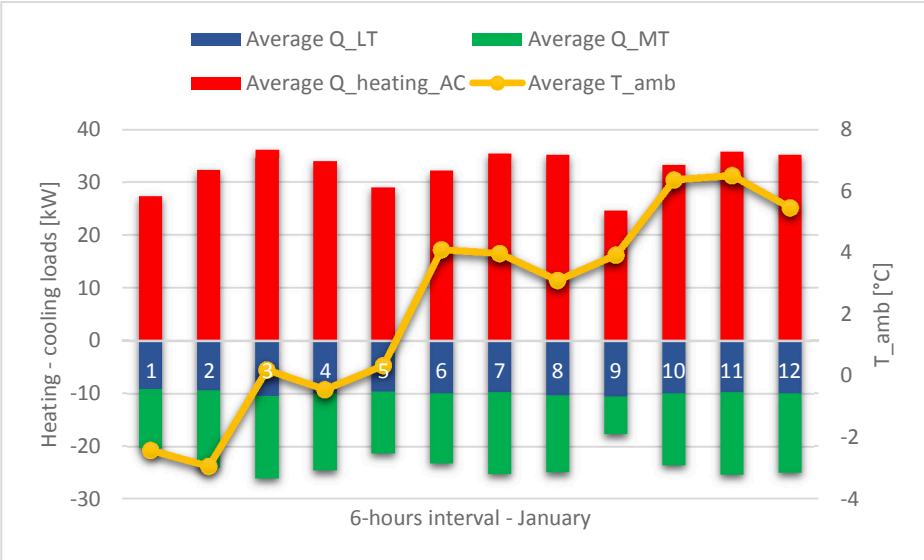


Figure 22: 6-hourly averaged cooling and heating loads for Kiwi Olsvik 28.01-30.01 2019

As the heat demand for the heat recovery cycle during the warm period in July is zero, the compressor loads are used for only refrigeration and thus satisfy the refrigeration-only criteria of Equation 19-21. The following figures illustrates the energy distribution of the refrigeration system of Kiwi Olsvik during July 2019 using COP_{tot} (COP_{ref}), COP_{LT} and COP_{MT}. The COP_{tot} is halved compared to the cold study, as seen in Figure 23 and compared to the values of COP_{tot} in Figure 21. The ambient temperatures are very high, and the system is running in transcritical operation at large portions of the time with the heat rejection pressure being in the range of 65 to 85 bar. The HR HX outlet temperature is as high as 37.4°C at its highest, which limits the system refrigeration capacity at lower MT compressor discharge pressure levels. As can be observed from Figure 23 the COP_{tot} is near 1 at instances with very high ambient temperatures. This is also visible in Figure 24 where the COP_{LT} is below 1 at instances where the ambient temperature is higher than 34°C.

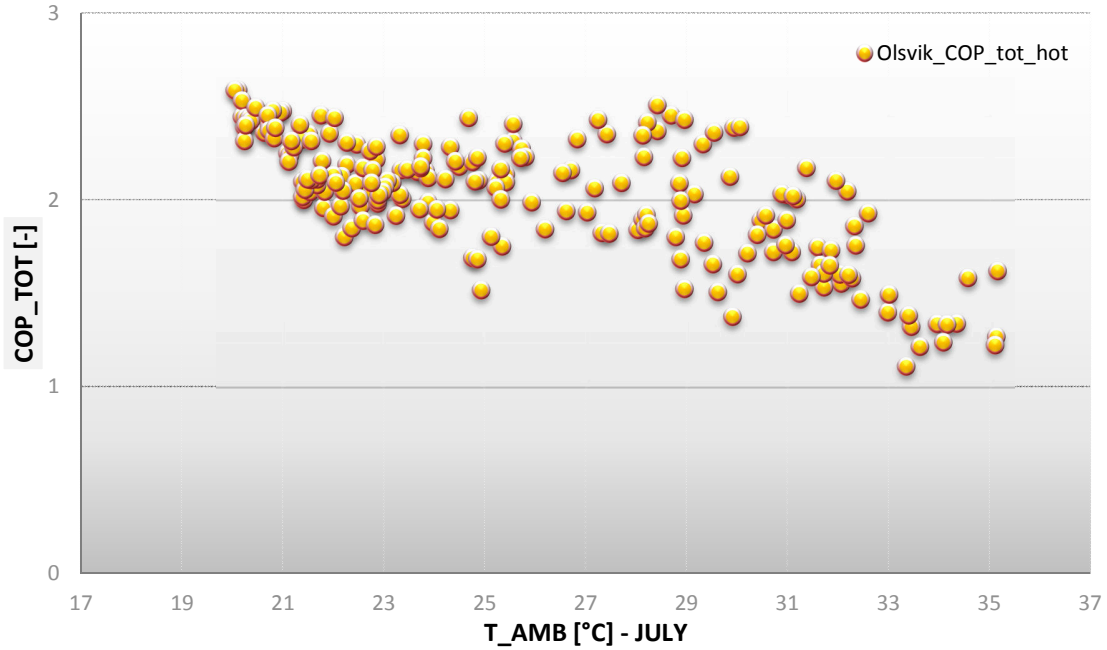


Figure 23: 20-min averaged total COP for Kiwi Olsvik in the period of 25.07-27.07 2019

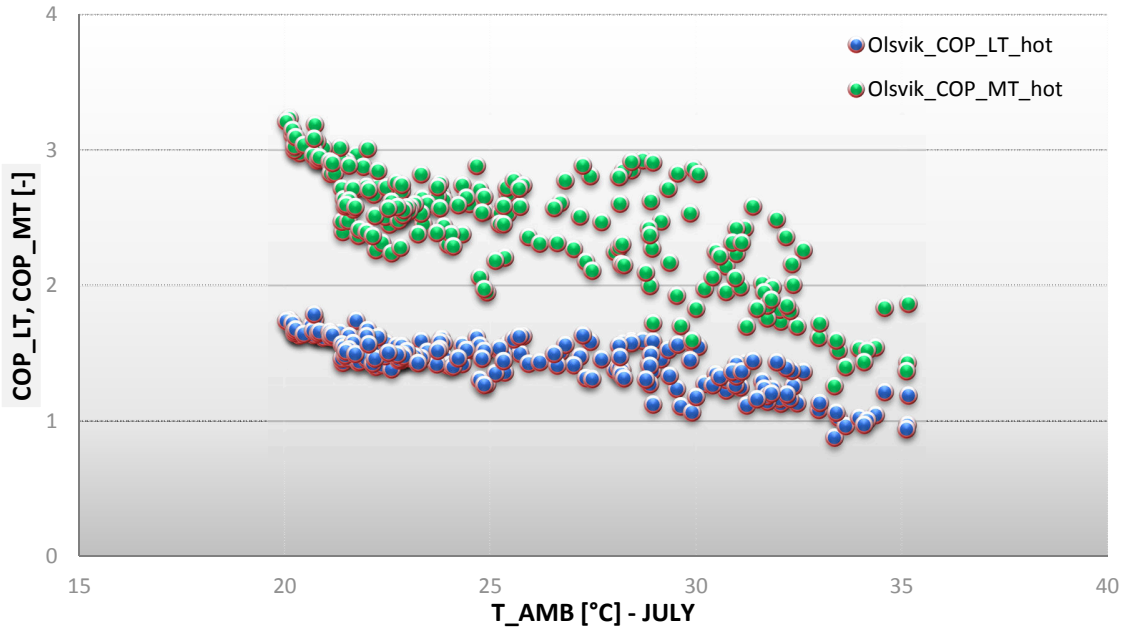


Figure 24: 20-min averaged COP_LT and COP_MT for Kiwi Olsvik in the period of 25.07-27.07 2019

The heat absorbed to the heat recovery cycle is not utilized for the heating battery of the AHU due to no heat demand as the natural consequence of hot ambient temperatures. The red columns of Figure 22 are replaced with brown columns, representing the heat released to the ambient area of Kiwi Olsvik.

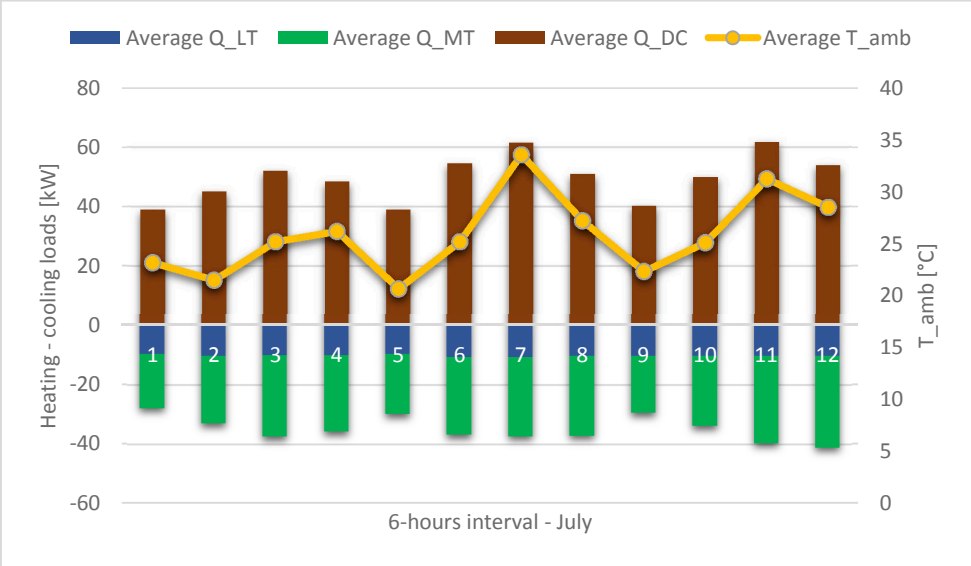


Figure 25: 6-hourly averaged cooling and heating loads for Kiwi Olsvik 25.07-27.07 2019

During such high ambient temperature, the refrigeration system is really at test for fulfilling the cooling and freezing temperatures limit requirements, as shown in Figure 26. Although the temperature exceeds the limits for the MT cabinet and MT cooling room used for calculating the MT evaporator exit temperature in the energy calculations, it is only for short periods of time. The refrigeration system keeps the

storing temperature low enough, but a consequence is higher compressor loads and a corresponding lower COP_{ref}.

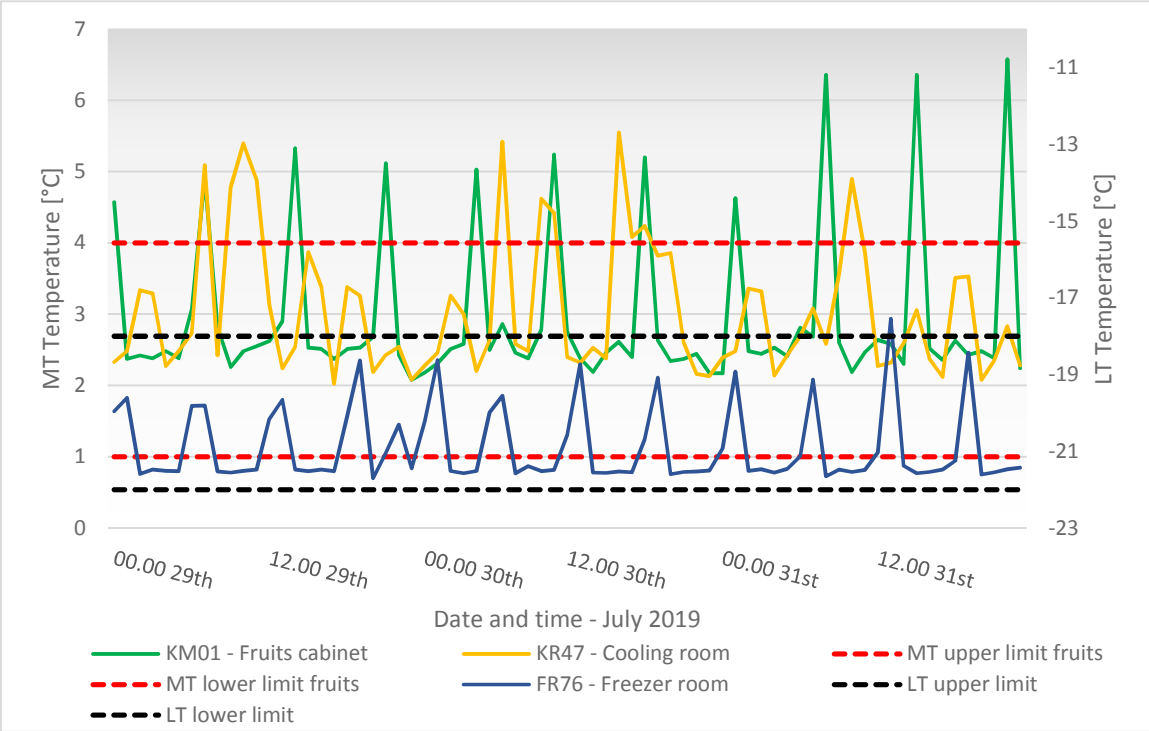


Figure 26: Cooling and freezing temperatures in Kiwi Olsvik July 2019

5.2 Kiwi Tertnes

The refrigeration performance of Kiwi Tertnes is expected to be quite similar to the system of Kiwi Olsvik during colder periods. The heat rejection process is different as described in Chapter 3. Other factors which could explain different results are the total cooling demand and freezing demand, but both the weather and technical conditions are quite alike.

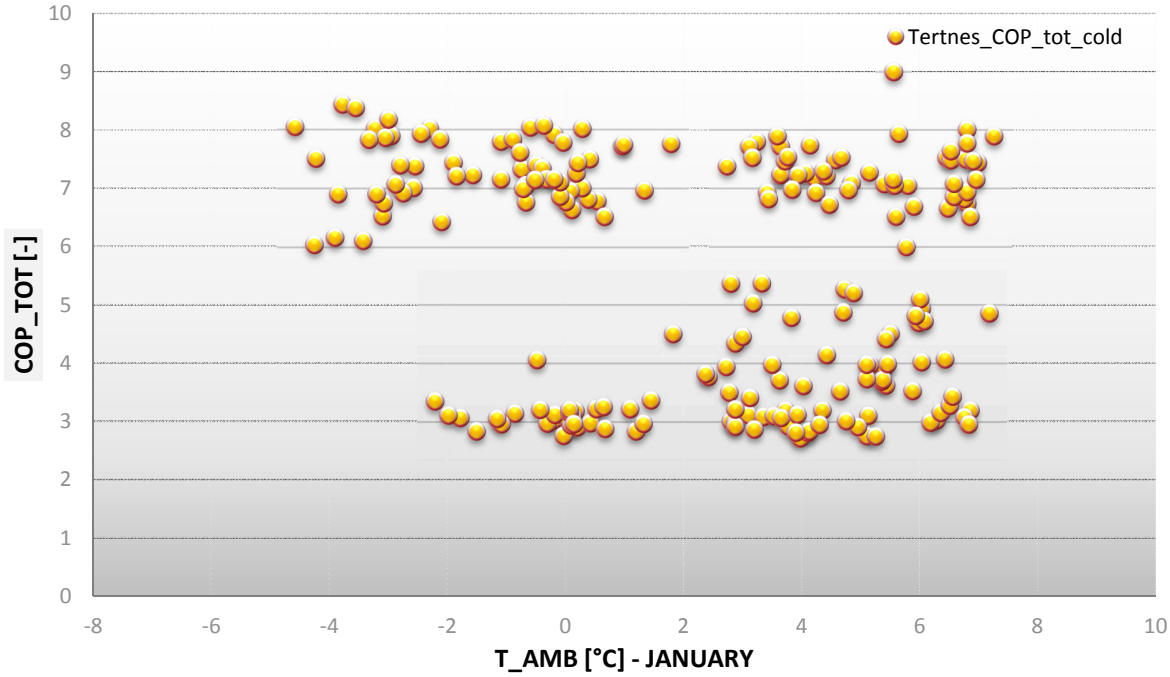


Figure 27: 20-min averaged total COP for Kiwi Tertnes in the period of 28.01-30.07 2019

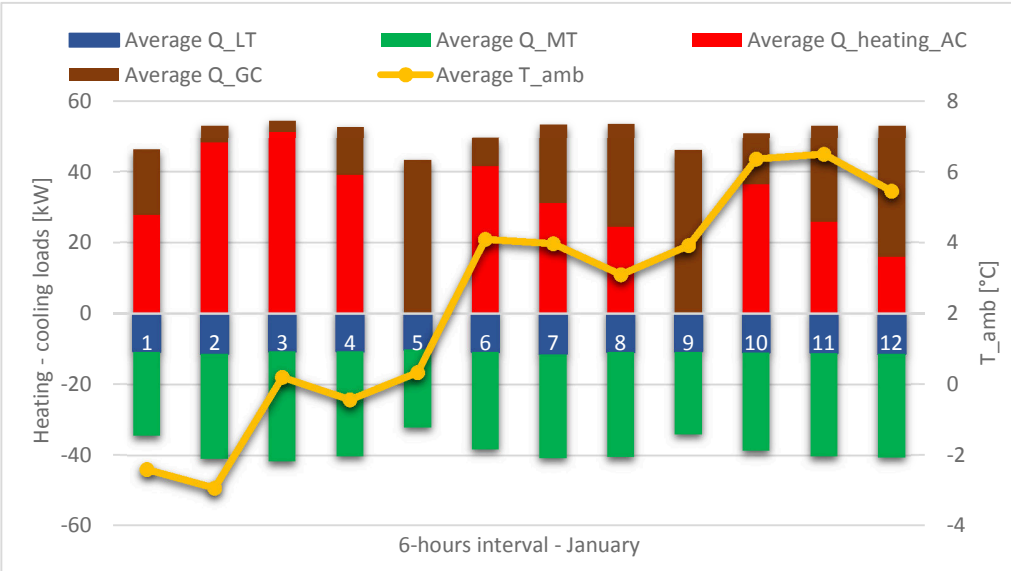


Figure 28: 6-hourly averaged cooling and heating loads for Kiwi Tertnes 28.01-30.01 2019

The COP_{tot} has a variation of values from just beneath 3 to above 8. The reason why the COP_{tot} values are seemingly divided in two groups are the presence of heating loads rejected to the ambient in the gas-cooler, as can be seen in Figure 28. The COP_{tot} is lower when the heat is rejected, because this simply means a lower heat demand for AC heating, thus reducing the overall amount of heat utilized for heating purposes. These low numbers for AC heating coincide with

nighttime, meaning the system's actual COP_{tot} during low ambient temperatures are between 6 and 9. The high-stage pressure is stable around 80 bar and the inlet temperature at the HPV is in the range of 12-19°C when most of the heat is utilized for AC heating.

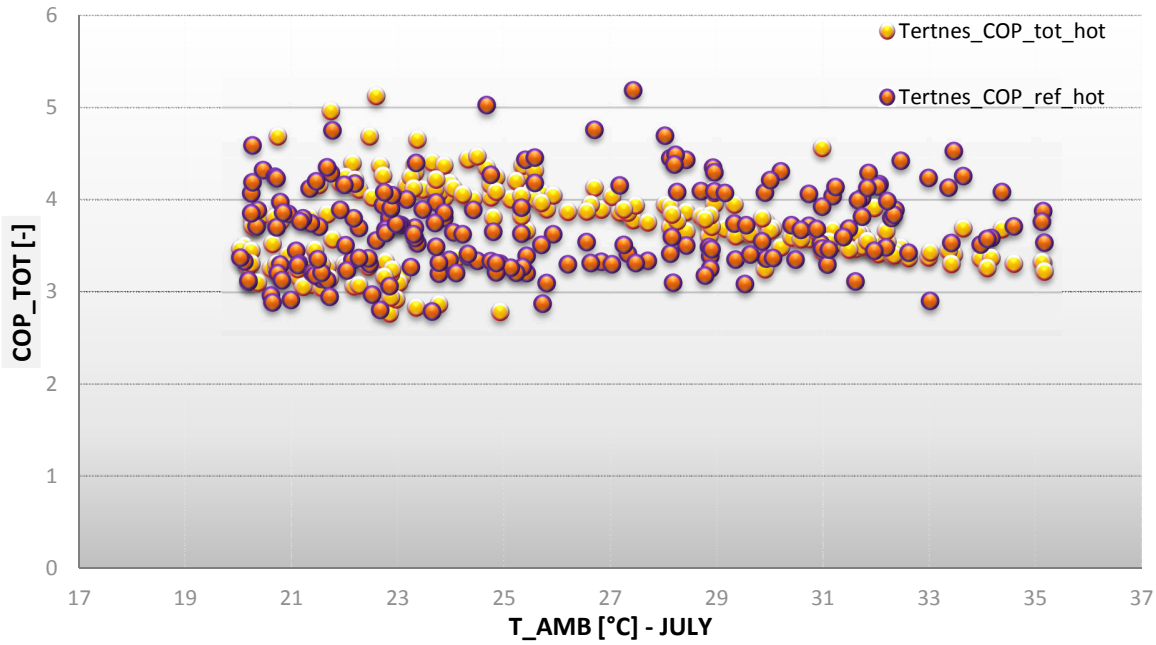


Figure 29: 20-min averaged total COP for Kiwi Tertnes in the period of 25.07-27.07 2019

The reason why COP_{tot} and COP_{ref} are quite like, although one should expect higher COP_{tot} with a positive $COP_{AC,cooling}$, is the lack of any reliable and useful data calculated for $\dot{Q}_{AC}$. The method of simulating the mass flow of the PC proved to be unsuccessful. The simulated mass flow and the calculated flash gas are very similar in terms of quantity, resulting in a small or even negative AC cooling mass flow. With an expected $\dot{Q}_{AC}$ in the range of 15 kW at maximum heat demand when comparing with a similar study [22] and usig an Advansor data sheet obtained from the technical projector of the supermarket (Appendix E), the inclusion of $\dot{Q}_{AC}$ should be significant for a higher COP_{tot}.

The COP_{tot} has a clear trend of dropping with increasing temperatures, as seen in Figure 29. This is expected as the HPV inlet temperature increases due to the limiting condensing conditions due to warm weather, and not least due to the missing cooling load. There is no heat reclaim in the AC during this warm period of July 2019, all heat is released to the ambient. The correlation between ambient temperature and highs-stage pressure is significant, as seen in Figure 32. It can be observed that when the ambient temperature exceeds approximately 25°C, the system is running transcritical.

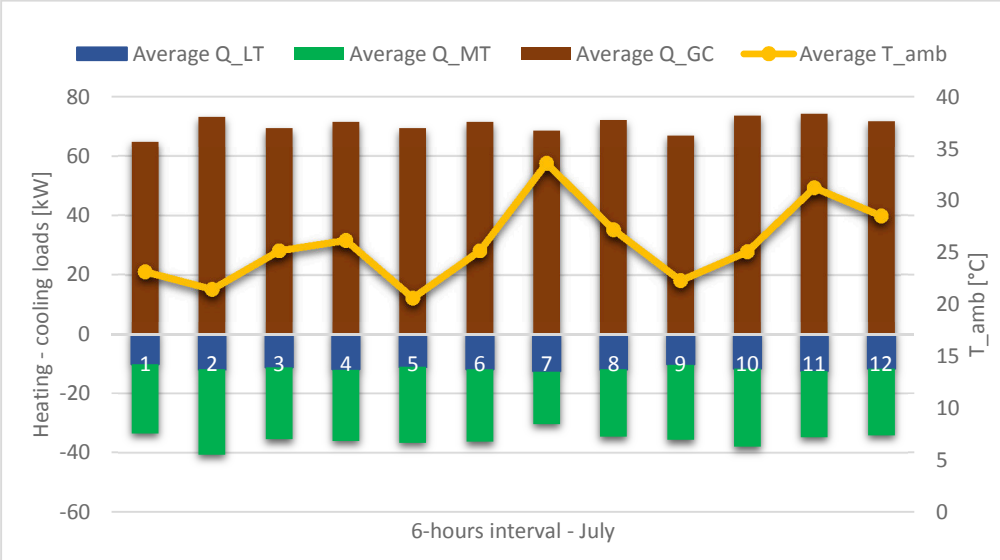


Figure 30: 6-hourly averaged cooling and heating loads for Kiwi Tertnes 25.07-27.07 2019

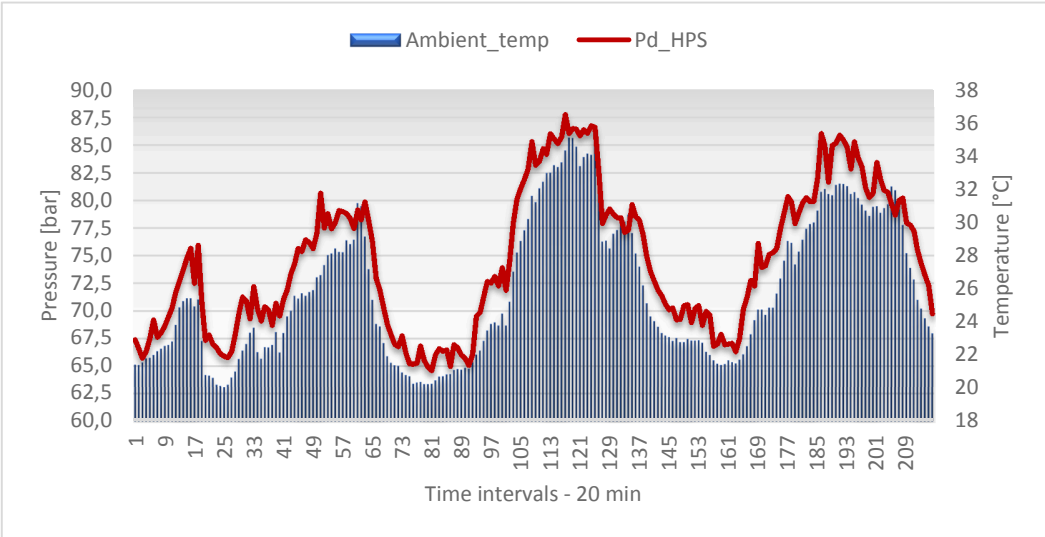


Figure 31: 20-min average ambient temperature vs high-stage pressure - July

During the three-day evaluated, the 20-min average of the high-stage pressure exceeds the critical pressure of 73.8 bar for CO_2 in 50.5% of the time.

The COP_{LT} and COP_{MT} have same trend as COP_{tot}, by decreasing to increasing ambient temperatures, see Figure 32. The MT cooling load varies, as seen in Figure 30. This could indicate that the cooling cabinets are experiencing heat gains from high temperatures inside the supermarket. The mechanism behind the low COPs are the same as for the system of Kiwi Olsvik, and later as will be seen, for the system of Spar Snarøya. None of the cooling cabinets or cooling rooms checked

upon had temperatures exceeding the limits for safe storage conditions.

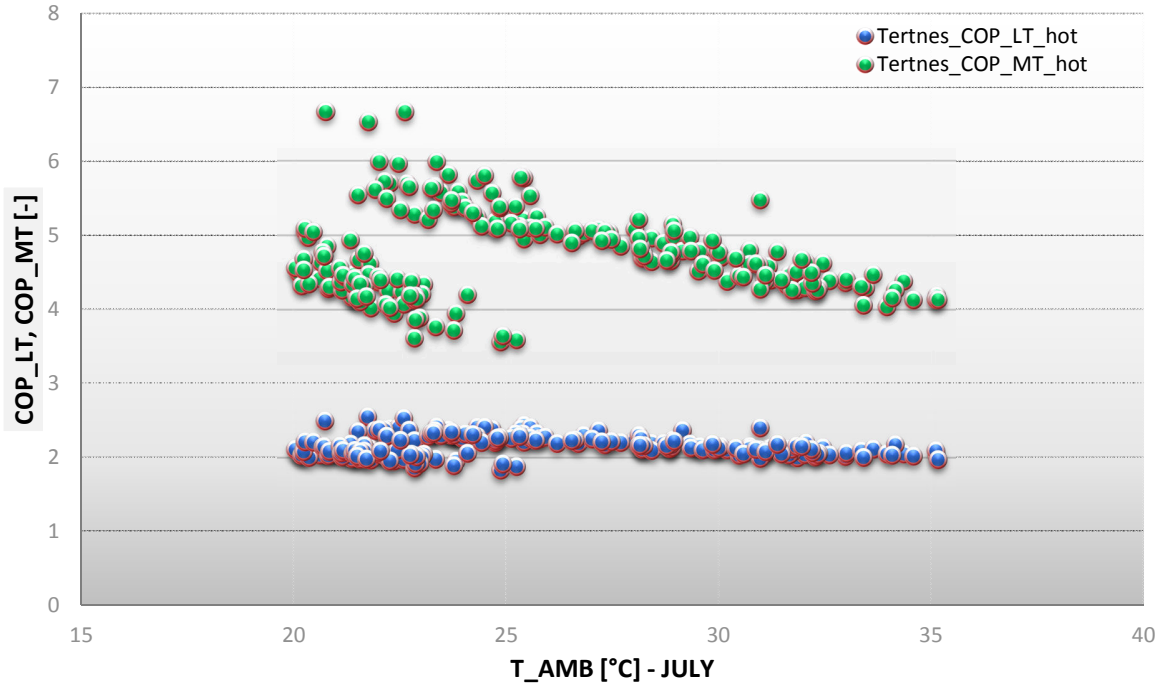


Figure 32: 20-min averaged COP_LT and COP_MT for Kiwi Tertnes in the period of 25.07-27.07 2019

5.3 Spar Snarøya

Snarøya has a similar refrigeration system design as Kiwi Tertnes, but with some supplementing installations. The calculation of IHX 2 is carefully described in chapter 4.2.1, and one could suspect a small positive effect on the COP_{LT}, COP_{MT} and COP_{tot}, both for January and July weather conditions.

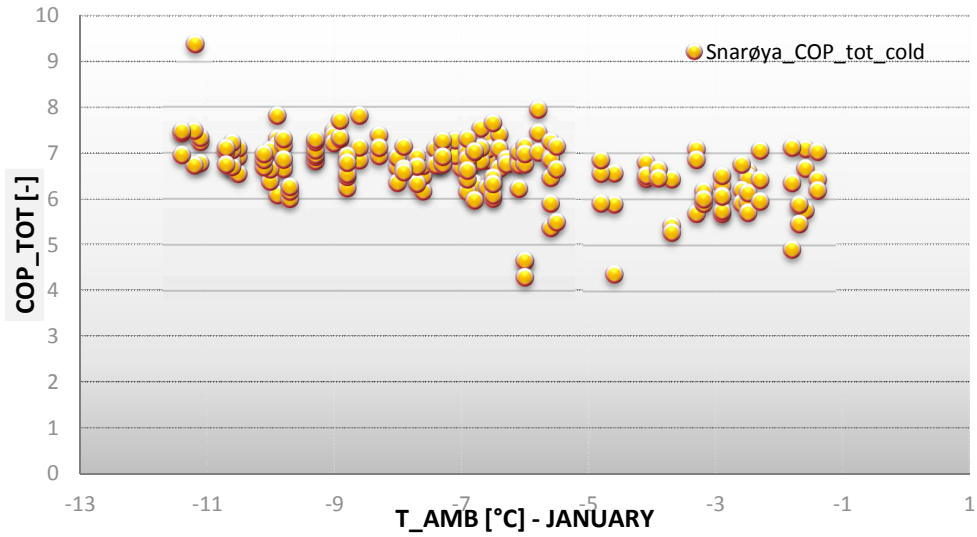


Figure 33: 20-min averaged total COP for Spar Snarøya in the period of 29.01-31.01 2019

The average high-side pressure approximately 80 bar over the course of the three cold days evaluated. The inlet HPV temperature is averaging as low as approximately 6.5°C. It would be interesting seeing the COP_{LT} and COP_{MT} for the period, because as Figure 33 implies, the COPs are high.

Figure 34 presents the energy loads during the cold period, and the results are similar to the results of Kiwi Tertnes.

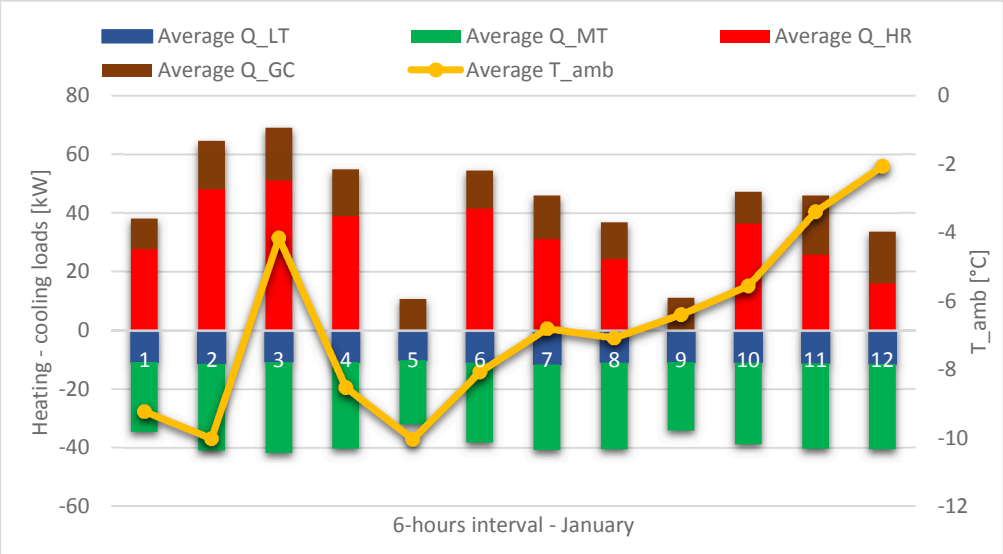


Figure 34: 6-hourly averaged cooling and heating loads for Spar Snarøya 29.01-31.01 2019

The heat recovery cycle, or heat storage cycle, of Spar Snarøya has a high capacity for heat recovery due to the heat storage tanks. This results in less heat are wasted, and consequently a higher COP_{tot} for the system. The heat may also be utilized for heating of domestic tap-water.

The COP-values of Spar Snarøya is high also during the warm period. As can be observed in Figures 35 and 36, there are some missing data for the calculations of the COPs. As for the system of Kiwi Tertnes, the COPs are calculated without the effects of the parallel compressor. As for the system of Kiwi Tertnes during warm weather, the COP_{tot} should be expected to be higher with a positive COP_{AC}.

Next chapter is a thorough comparison between the systems and includes a discussion on the missing data of parallel compressor mass flow and AC cooling and their possible impact on the COP_{tot} and COP_{ref}.

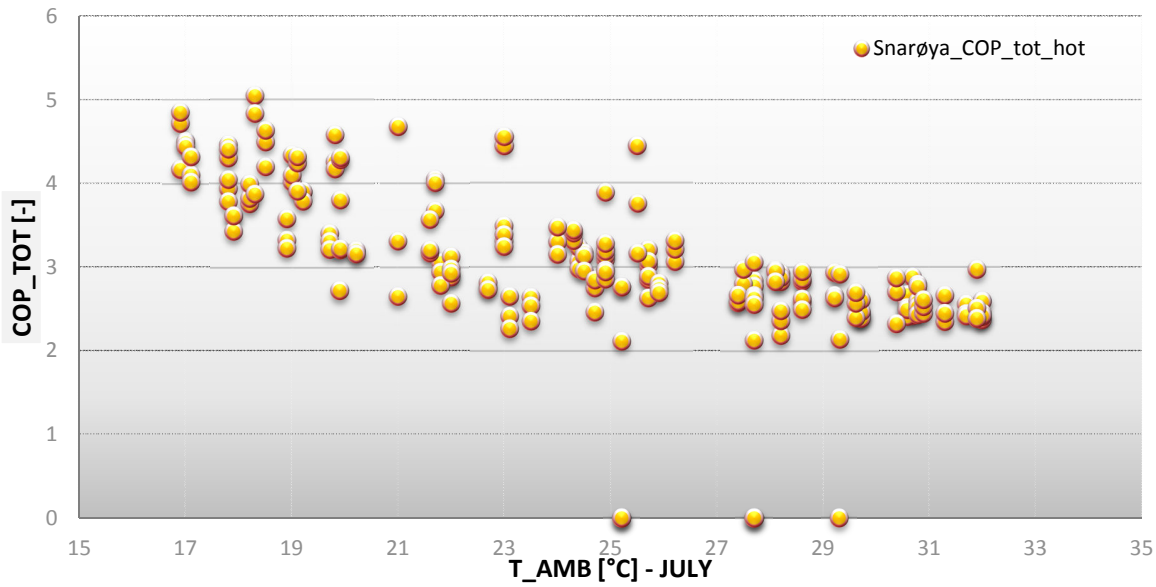


Figure 35: 20-min averaged COP_LT and COP_MT for Spar Snarøya in the period of 25.07-27.07 2019

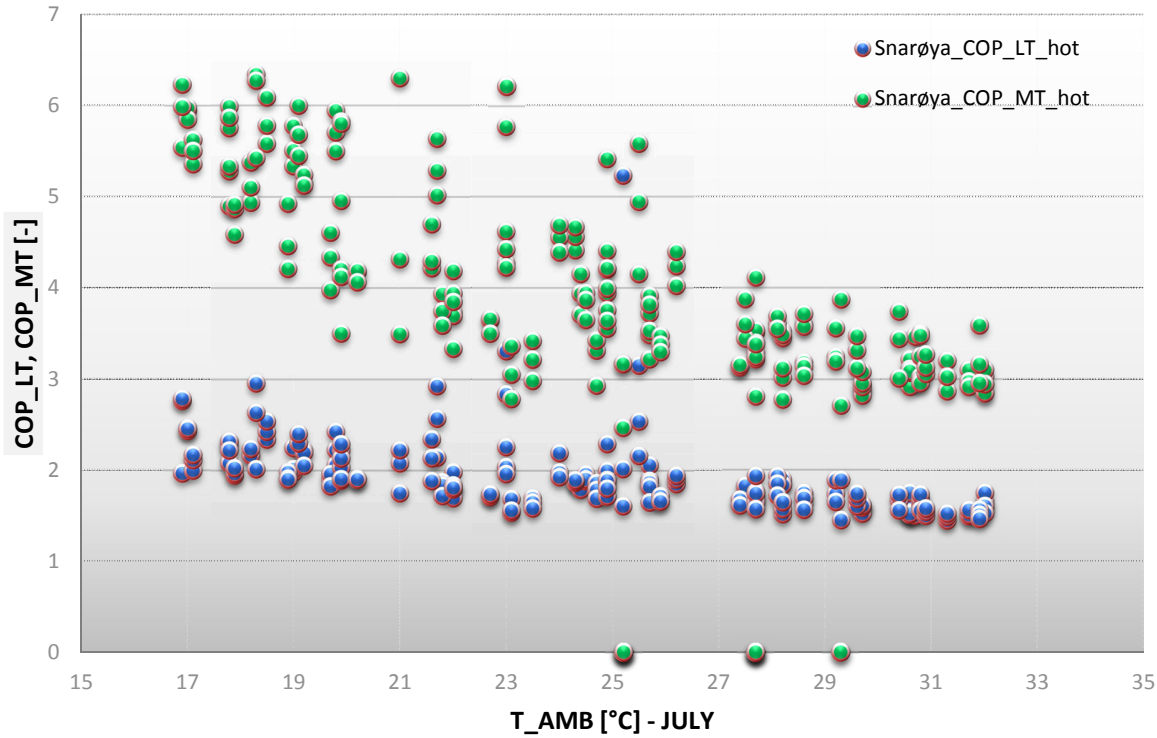


Figure 36: 20-min averaged COP_LT and COP_MT for Spar Snarøya in the period of 25.07-27.07 2019

6 Discussion

When comparing the three transcritical CO_2 refrigeration systems with integrated heat recovery systems, the emphasis is on heat recovery during the cold three-day period and on refrigeration COP during the hot three-day period.

6.1 Refrigeration performance July

As can be observed in the stabled bar diagrams in Chapter 5, the LT refrigeration loads are stable for all three systems during the whole day. The LT refrigeration loads seems nearly independent of weather conditions, which to a large extent can be ascribed to enclosed freezer cabinets with glass lids and relatively small heat gains.

Table 8: Key parameters for energy efficiency evaluation of three refrigeration systems; Kiwi Olsvik, Kiwi Tertnes and Spar Snarøya

27.07 2019 12.00-18.00	**KIWI OLSVIK**		**KIWI TERTNES**		**SPAR SNARØYA**	
FUNCTION	Avg	Range	Avg	Range	Avg	Range
LT REFRIGERATION LOAD [KW]	10.3	7.7-12.2	12.4	10.8-15.3	10.6	10.3-10.8
MT REFRIGERATION LOAD [KW]	29.4	24.3-32.5	22.3	12.8-27.4	36.7	30.4-55.4
HEAT REJECTION LOAD [KW]	61.8	42.0-73.1	74.0	66.9-79.0	66.7	57.7-92.9
GASCOOLER/DRY COOLER EXIT TEMP. [°C]	35.8	33.8-37.4	32.3	30.5-33.5	31.6	31.0-32.1
HPV INLET TEMPERATURE [°C]	35.8	33.8-37.4	31.9	30.3-33.1	31.1	30.6-31.6
HR PRESSURE [BAR]	84.9	84.4-86.0	82.2	79.0-84.8	81.1	79.8-82.0
COP_{REF} [-]	1.6	1.3-1.9	3.6	3.3-4.1	2.5	2.4-3.0

The average temperature between 12.00-18.00 27.07 is very similar between the two supermarkets in Bergen (31°C) and the one in Bærum (31.5°C). Table 8 consist of key parameters for evaluating and comparing the COP_{ref} between the systems. The LT refrigeration load is similar for all systems, especially when observing the refrigeration loads over longer time. The reason why the LT refrigeration load at Kiwi Tertnes is higher could be due to number of reasons; grocery items being replaced, lids being left open, etc. The freezing capacity is similar for Kiwi Tertnes (11.4kW) and Kiwi Olsvik (13.5 kW). The cooling capacities are also similar; 54.7 kW and 47.7 kW respectively, for these two systems. It is fair to assume the cooling and freezing capacity of Spar Snarøya to be in the same region based on the number of refrigeration compartments installed (see Table 2).

The MT refrigeration load is higher at Spar Snarøya compared with the other two systems. The cooling load in IHX 2 of the system (see Fig. 15) contributes positively to the refrigeration capacity. The positive cooling load is on average 0.7 kW for LT and 4 kW for MT during the same time span as the parameters in table 8.

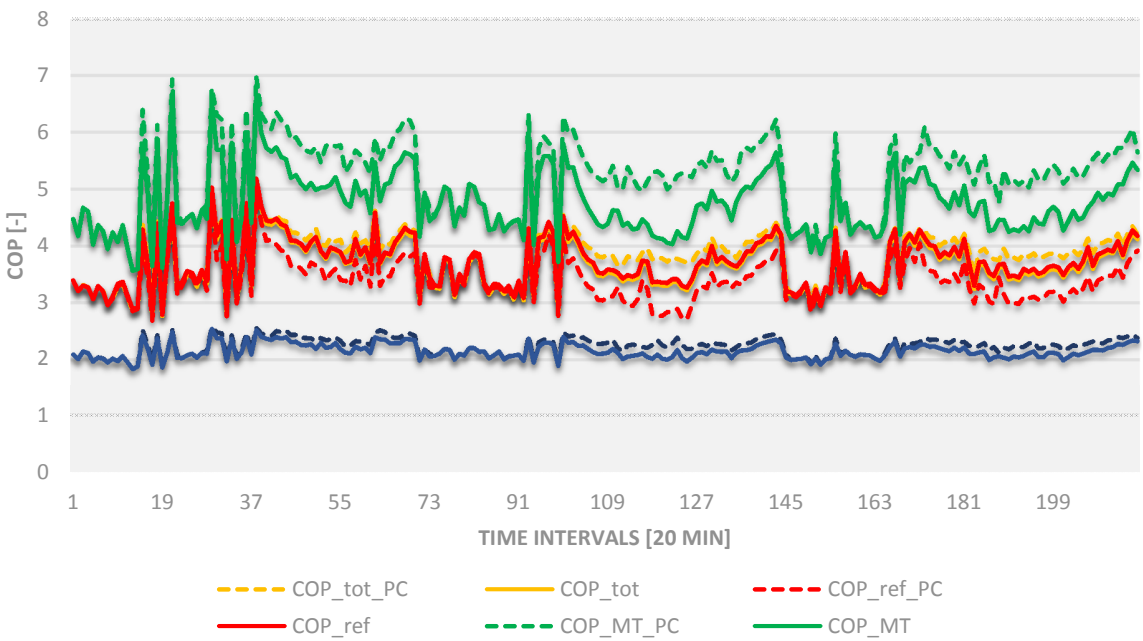


Figure 37: Graph illustrating the effect of including an estimate of AC cooling load in the COP calculations of Kiwi Tertnes during the hot period, 25.07-27.06

The significance of the exclusion of the contribution of the parallel compressor power $\dot{E}_{PC}$ and the cooling load $\dot{Q}_{AC}$ on the COP_{tot} is dependent on the cooling capacity of the cooling battery and the running capacity of the parallel compressor. Some rough estimates are made where $\dot{E}_{AC}$ is half of $\dot{E}_{PC}$ and $\dot{Q}_{AC}$ the maximum cooling effect of 15.9 kW multiplied with the working ratio of the PC compressor. Figure 37 presents a scenario where the estimate of $\dot{Q}_{AC}$ is included in the calculations of the COPs in the system of Kiwi Tertnes. Even though the values for

$COP_{with\ PC}$ are pure estimates, these values are closer to reality than the COP calculated without PC. The graphs of COP_{ref} and COP_{tot} are equal when there is no AC cooling load, heat rejection load or parallel compressor power involved (hence no sign of the yellow graph, it is covered by the red graph). The effect of having additional AC cooling and a parallel compressor has clearly a positive effect on the COP_{tot}, but it is must be emphasized that Figure 38 does not represent a real presentation of the transcritical CO_2 booster refrigeration system of Kiwi Tertnes with parallel compression.

The system of Kiwi Olsvik, Kiwi Tertnes and Spar Snarøya have at an ambient temperature around 30°C approximately a COP_{tot} of 1.6, 3.9 and 2.4 respectively. The value of Spar Snarøya is the same as COP_{ref} as no contribution of the PC is included. It is evident that nevertheless the lack of precise and reliable data on the PC, that a CO_2 refrigeration system with integrated AC cooling performs significantly better than conventional refrigeration system. This is supported in the field of science of refrigeration [14, 18, 22].

6.2 Refrigeration system total performance January

A comparison of the systems is more applicable during colder weather when the systems operate at similar technical conditions. Only the total COP is evaluated, due to the lack of possibility to run a reliable floating condensing pressure/temperature model for the refrigeration systems. Based on the refrigeration COP during the warm period, Spar Snarøya is expected to perform better than the other two systems due to the inclusion of the internal heat exchanger downstream of the separator tank.

The plotting of the COP values in Figure 21, Figure 27 and Figure 33 shows the following:

- Kiwi Olsvik has COP_{tot} in the region of 5-6.
- Kiwi Tertnes has COP_{tot} in the region of 3-4 and 7-8.
- Spar Snarøya has COP_{tot} in the region of 6-7.

The variations of the COPs are attributed to the heat rejection systems. The system of Kiwi Olsvik will continuously reject heat by applying the ventilation heat system, whereas the system of Spar Snarøya has the possibility to store heat in a heat storage system. This is energy saving as less energy is used to drive the ventilation system. It is especially noticeable that the system of Olsvik never rejects heat to the ambient during the evaluated three cold-days, indicating that the system must utilize all the excess heat for heating. The MT discharge pressure is different. During the hottest evaluated 6-hour window, the refrigeration system of Kiwi Olsvik was running at subcritical, while the refrigeration system of Kiwi Tertnes ran

transcritical. In Figure 38, the high-side pressure is plotted against the heat recovery load of the two systems:

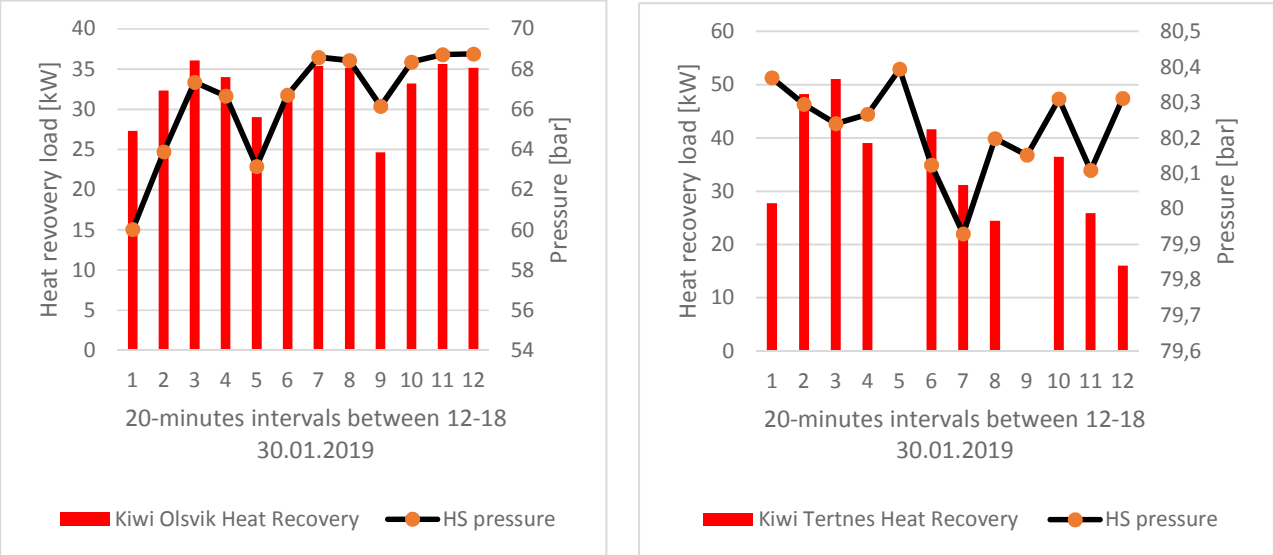


Figure 38: High-side pressure vs Heat recovery load, Kiwi Olsvik left, Kiwi Tertnes right

Even with lower MT discharge pressure, the system of Kiwi Olsvik has lower COP_{tot} compared to Kiwi Tertnes, and consequently the similar system of Spar Snarøya when it comes to heat rejection. The system of Kiwi Olsvik require more electricity.

The energy consumption for cooling, freezing and AC of each of the systems are presented in Figure 39 below. It might come surprising that the total electricity load for refrigeration is close to even for Kiwi Olsvik and Kiwi Tertnes. In this investigation it is found that the COP_{LT} and COP_{MT} for Kiwi Tertnes are higher

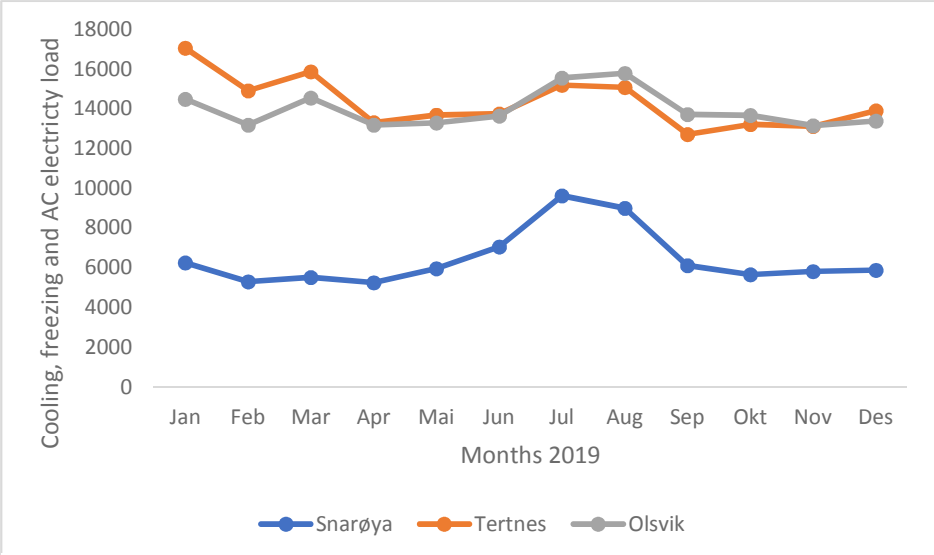


Figure 39: Electricity load for cooling, freezing and AC cooling 2019

compared to the one of Kiwi Olsvik. The COP_{tot} is even during cold climate, which indicates that the systems are equal during most periods of a year. As the Norwegian climate is cold in general, the positive impact of a parallel compression system is limited, and the positive result is small for two equally sized supermarkets in the same region with the same base refrigeration system. It is worth noting that the system of Kiwi Tertnes uses less electricity for cooling during the warmest months. Spar Snarøya is in a warmer climate during the summer and have in addition a wider range of energy saving solutions, thus consume less electricity.

7 Conclusion

This thesis evaluated the refrigeration system performances of three supermarkets in Norway. All three refrigeration systems are transcritical CO_2 systems with integrated heat recovery system. The technical features for increasing the energy efficiency are different for each system. This difference in the system solutions is the basis for the comparison between the systems. The most prominent features for each system are:

- Kiwi Olsvik: Two-stage compression, transcritical booster system, high-stage heat exchanger working as gas cooler. The heat recovery consists of heat ventilation, and a separate circuit is used for heat recovery and heat rejection.
- Kiwi Tertnes: Two-stage compression with parallel compression, transcritical booster system, high-stage heat exchanger is a de-superheater where heat is recovered for ventilation heating. AC cooling is integrated.
- Spar Snarøya: Two-stage compression with parallel compression, transcritical booster system, high-stage heat exchanger is a de-superheater connecting two heat storage tanks, heating ventilation and pre-heating tap water. Pre-heating of tap water is not included in the web-monitor system. AC cooling is integrated.

The parameters for calculating the energy performance of the systems were collected from two web-monitor systems, ENVO and IWMAC. Two periods were used for evaluation of the systems, a cold three-day period in January and a hot three-day period in July. The LT refrigeration loads are stable for all three systems, both during cold and hot ambient temperature conditions. Both LT and MT refrigeration loads were greater for the system of Spar Snarøya. This is ascribed to the inclusion of an internal heat exchanger subcooling the IT liquid CO_2 leaving the separator, and possible a slightly higher MT cooling demand.

Because problems with calculations of the AC cooling load, the COP values calculated for the hot ambient temperature conditions do not tell the whole story. The total COP is significantly higher with the inclusion of parallel compressor, especially at ambient temperatures in the range of 15°C to 25°C, where the COP

(which is lower than the real COP with real data would be) is 1-1.5 times higher for the systems with parallel compressor than without. At lower temperatures, the flash gas by-pass is used for controlling the pressure in the separator tank.

The investigation of the performances of the systems during the colder period is limited by the absence of refrigeration COP calculation with no floating pressure simulation analyzes performed. The COP of each system is high, and the energy efficiency of a transcritical CO_2 booster refrigeration system proves to be competitive when the excess heat of the refrigeration process is utilized for energy saving purposes. Out of the three systems evaluated, Spar Snarøya has the highest energy efficiency potential by being the most flexible in terms of utilizing the energy within the system, and for heating and cooling purposes in the supermarket.

8 Further work

The investigation and calculation of the energy performances of the systems were limited by some missing parameters, especially in the parallel compressor line. A more comprehensive research with more reliable data, meaning more sensors, would add strength to such a research performed with the work of this thesis. All the calculations were done in Microsoft Office Excel, which also limits the work. A combination of Excel and a program such as EES would increase the efficiency of the work, and also the possibility to compare idealized simulations with real data results.

As the natural refrigerant are becoming more and more dominant in the food retail industry due to the protocol restricting the use of non-natural refrigerants, more research on increasing the performance of transcritical CO_2 refrigeration systems should be motivated for. The performances of CO_2 refrigeration systems are competitive in colder climate, hence the challenge lays in having more year-round performing refrigeration systems.

A more comprehensive study of the system of Spar Snarøya is advised, and using the results of the energy performance of the system to increase the overall energy performance of similar systems in warmer climate, such as Southern-Europe, Australia, etc.

List of References

1. Bitzer. *Bitzer*. 2020; Available from: www.bitzer.de.
2. IISD. *Sustainable development*. 2020; Available from: www.iisd.org/topic/sustainable-development.
3. State, U.S.D.o. *The Montreal Protocol on Substances That Deplete the Ozone Layer*. 2019; Available from: ww.state.gov/key-topics-office-of-environmental-quality-and-transboundary-issues/the-montreal-protocol-on-substances-that-deplete-the-ozone-layer/.
4. Arias, J., *Energy Usage in Supermarkets - Modelling and Field Measurements*, in *Industrial Engineering and Management*. 2005, KTH.
5. Jaime Arias, P.L., *Heat recovery and floating condesning in supermarkets.* Energy and Buildings, 2004(38): p. 73-81.
6. Forbes Pearson, S. *Refrigeration Past, Present and Future*. 2004 06.01.2020]; Available from: http://www.r744.com/files/pdf_597.pdf.
7. Calm, J.M., *The Next Generation of Refrigerants - Historical review, considerations, and outlook.* International Journal of Refrigeration, 2008(31): p. 123-133.
8. Nations), U.U., *Amendment to the Montreal Protocol on substances taht deplete the ozone layer (Kigali Amendment)*. 2016.
9. Kauffeld, M., *Trends and Prespectives in Supermarket Refrigeration*. 2008, Institute of Refrigeration, Air Conditioning and Environmental Engineering: Karlsruhe University of Applied Sciences.
10. EmersonClimateTechnologies, *Refrigerant Choices for Commercial Refrigeration*. 2010.
11. Funder-Kristensen, T., *A CO_2 dream solution for a supermarket - a concept case*. 2013, Danfoss.
12. NavigantConsulting, *Case Study: Transcritical Carbon Dioxide Supermarket Refrigeration Systems*. 2015.
13. Sawalha, S., *Carbon Dioxide in Supermarket Refrigeration*, in *Industrial Engineering and Management*. 2008, KTH.
14. A. Hafner, I.C.C., F. Schmidt, R. Olsson, K. Fredslund, P.A. Eriksen, K.B. Madsen, *Efficient and Integrated Energy Systems for Superamarkets*, in *IIR-Gustav Lorentzen Conference on Natural Refrigerants GL2014*. 2014, International Institute of Refrigeration.
15. Sawalha, S., *Commercial Refrigeration.* Education and Culture Lifeling Learning Programme, 2009.
16. Pieter C. Koelet, T.B.G., *Industrial Refrigeration: Principles, design and application*. 1992, Macmillan. p. 73-131.
17. Karampour, M. and S. Sawalha, *State-of-the-art integrated CO2 refrigeration system for supermarkets: A comparative analysis.* International Journal of Refrigeration, 2017(86): p. 239-257.
18. Sawalha, M.K.S., *State-of-the-art integrated CO2 refrigeration system for supermarkets: A comparative analysis.* International Journal of Refrigeration, 2017(86): p. 239-257.
19. IWMAC. Centralized Operation and Surveillance by Use of WEB Technology [cited 2019-2020; Available from: www.iwmac.com/no.
20. Envo. 2019-2020; Available from: www.ems.envo.no.
21. Moran, M.J., Shapiro, Howard N., *Fundamentals of Engineering Thermodynamics*. 5th ed. 2004: John Wiley & Sons, Inc.
22. Karampour, M., Sawalha, Samer, *Energy efficiency evaluation of integrated CO2 trans-critical system in supermarkets: A field measurement and modelling analysis.* International Journal of Refrigeration, 2017. **82**: p. 470-486.

Appendix A

Code for aligning parameters with the same time stamp on the row in an excel-sheet:

```
Private Sub CommandButton1_Click()
Dim rH As Long, Row As Long, Col As Long, NextCellValue As Double
Dim Rowsize As Integer, ColSize As Integer
Dim foundRng As Range
Set DataArk = Worksheets("Disorderly")
Set ResArk = Worksheets("Organized")
'Copy the first two rows
rH = 1
Application.ScreenUpdating = False
   For Row = 1 To 9
      ResArk.Rows(Row).EntireRow.Value = DataArk.Rows(rH).EntireRow.Value
      rH = rH + 1
   Next Row
'Copy column 1 and 2 in DataArk
   DataArk.Columns(1).Copy Destination:=ResArk.Columns(1)
   DataArk.Columns(2).Copy Destination:=ResArk.Columns(2)
'Search through every second column for the associated timestamp in column 1
ColSize = DataArk.Cells(3, Columns.Count).End(xlToLeft).Column
  For Row = 10 To 370
   For Col = 3 To ColSize Step 2
         Rowsize = DataArk.Cells(Rows.Count, Col).End(xlUp).Row
         Set foundRng = Range(DataArk.Cells(3, Col), DataArk.Cells(Rowsize, Col)).Find(DataArk.Cells(Row, 1))
         If foundRng Is Nothing Then
         ResArk.Activate
         ResArk.Cells(Row, Col + 1) = ""
         DataArk.Activate
         Else
         NextCellValue = DataArk.Cells(foundRng.Row, foundRng.Column + 1).Value
         ResArk.Activate
```

```
        ResArk.Cells(Row, Col + 1).Value = NextCellValue
        DataArk.Activate
        End If
    Next Col
Next Row
Application.ScreenUpdating = True
End Sub
```

Appendix B

Customer		Advansor			
Date		11/12/2017			
Project		gas cooler - KCE93E4 G722 SPM C24 EC CB CB3 18343			
Fluid		R744 superheated	Lp(A) @ 10m		35
HxLxP	[mm]	see drawing	Outer Area	[m²]	420
Inner Volume	[l]	33	Inner Area	[m²]	18
Air Side			R744 side		
Entering temperature	[°C]	30	Entering temperature	[°C]	93,9
RH Entering	[%]	50	Outlet temperature	[°C]	32,4
Flow	[m³/h]	37100			
Barometric Pressure	[kPa]	101,325	mass flow	[kg/h]	2171
Altitude	[m]	0	Pressure	[bar]	84,8
fan type		3 x MN280827	Total Capacity	[kW]	130

absorbed power	W	750
rpm		500
weight	kg	685
	V	6,7

Appendix C
BITZER Output data

Project survey

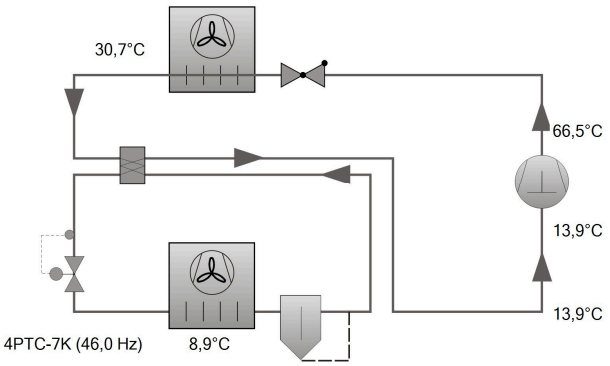
Selected compressors

Semi-hermetic Reciprocating Compressors 1x 4PTC-7K

Selection: Semi-hermetic Reciprocating Compressors

Input Values

Compressor model	(4PTC-7K)
Mode	Refrigeration and Air conditioning
Refrigerant	R744
Reference temperature	Dew point temp.
Evaporating SST	8,90 °C
High pressure	79,0 bar(a)
Gas cooling outlet	30,7 °C
Suct. gas superheat	5,00 K Operating mode Transcritical
Power supply	400V-3-50Hz
Useful superheat	100%

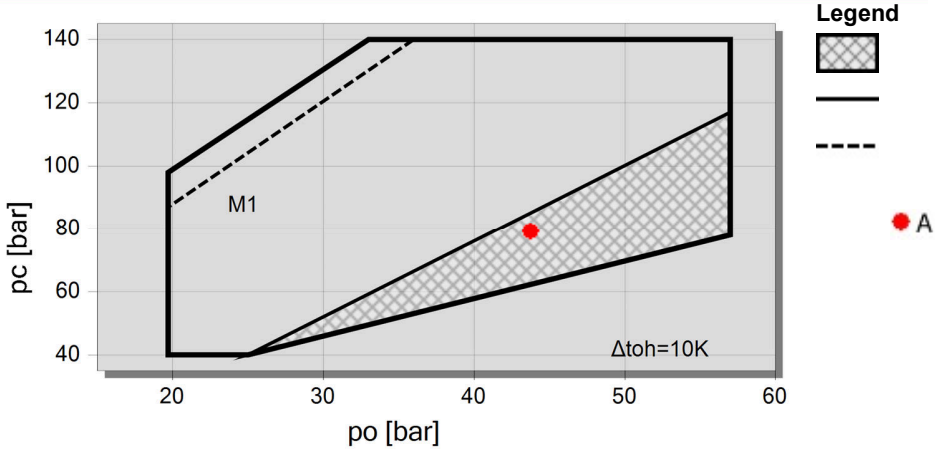


Result

Compressor	**4PTC-7K-40S**
Frequency compressor	46,0 Hz
Cooling capacity	15,71 kW
Evaporator capacity	15,71 kW
Power input	3,92 kW
Current (368V)	8,51 A
Gas cooler capacity	19,63 kW
COP/EER	4,01
Mass flow	388 kg/h
min. cooling capacity	7,73 kW (25 Hz)
max. cooling capacity	25,2 kW (70 Hz)
Discharge gas temp. w/o cooling	66,5 °C

optimal high pressure 75,4 bar(a)

Tentative Data.
Attention, consider operating parameters. See KP-130 or consult BITZER.
Power consumption at compressor inlet.

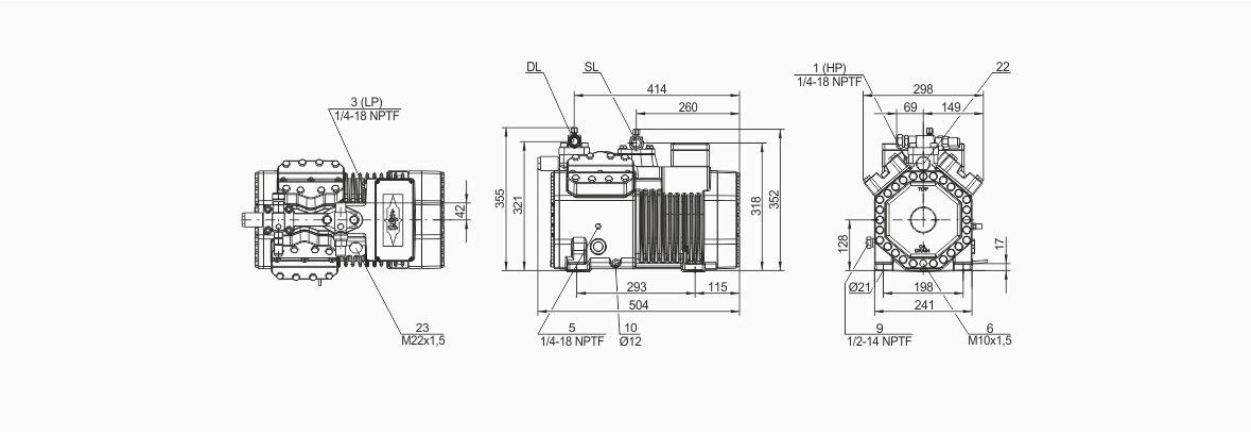


Application Limits mind operating parameters

M1: motor 1

Operation above designated line not allowed for following compressor models: 4PTE, 4PTC

Technical Data: (4PTC-7K)
Dimensions and Connections



Technical Data

Technical Data	
Displacement (1450 RPM 50Hz)	4,3 m3/h
Displacement (1750 RPM 60Hz)	5,2 m3/h
No. of cylinder x bore x stroke	4 x 30mm x 17,5mm
Weight	108 kg
Max. pressure (LP/HP)	100/160 bar
Connection suction line	22 mm - 1 1/8''
Connection discharge line	18 mm - 3/4"
Oil type R744 (CO2)	BSE85K (Standard), p0>40bar, pc>120bar: BSG68K (Option)

Motor data	
Motor version	1
Motor voltage (more on request)	380-420V Y-3-50Hz

Max operating current	15.3 A
Starting current (Rotor locked)	82.4 A
Max. Power input	8,2 kW
Extent of delivery (Standard)	
Motor protection	SE-B1
Enclosure class	IP65
Vibration dampers	Standard
Oil charge	2,00 dm^3
Crankcase heater	0..120 W PTC (Standard)
Available Options	
Connection suction line	Option
Discharge shut-off valve	Option
Oil level monitoring	OLC-K1 (Option)
Sound measurement	
Sound power level (-10°C / 90bar)	81 dB(A) @ 50Hz
Sound pressure level @ 1m (-10°C / 90bar)	73 dB(A) @ 50Hz

Appendix D

Intervall_hot_per	P_rec	H_satliq_rec	LT_To	LT_Po	LT Suction temp	H_Inlet_LT_Comp
1	37,975	208,054	-28,7	14,94	3,00	471,416
2	37,875	207,800	-27,9	15,31	3,13	471,056
3	38	208,118	-27,7	15,41	3,20	470,996
4	38,025	208,182	-28,4	15,09	3,15	471,366
5	37,975	208,054	-28,0	15,26	3,78	471,772
6	38,125	208,436	-27,8	15,40	4,03	471,837
7	37,925	207,927	-28,5	15,01	3,08	471,408
8	38	208,118	-27,9	15,33	2,30	470,197
9	38,175	208,563	-28,1	15,24	2,95	470,966
10	37,925	207,927	-28,8	14,86	3,80	472,324
11	38,025	208,182	-27,2	15,69	4,25	471,673
12	38	208,118	-28,3	15,14	2,15	470,302
13	37,875	207,800	-28,3	15,13	2,63	470,793
14	38	208,118	-27,8	15,37	3,33	471,172
15	38,05	208,245	-28,3	15,14	2,83	470,975
16	37,875	207,800	-28,2	15,18	2,53	470,626
17	38	208,118	-28,4	15,06	2,58	470,843
18	38,025	208,182	-28,1	15,23	3,18	471,207
19	37,975	208,054	-27,9	15,31	3,35	471,281
20	37,925	207,927	-27,9	15,32	3,90	471,813
21	38,025	208,182	-27,5	15,55	4,15	471,759
22	38	208,118	-28,3	15,14	3,13	471,274
23	38	208,118	-27,9	15,32	2,00	469,914
24	38,025	208,182	-28,1	15,23	2,63	470,658
25	37,975	208,054	-28,2	15,16	2,85	470,984
26	37,975	208,054	-27,9	15,35	2,95	470,831
27	38,1	208,373	-27,9	15,32	2,75	470,664
28	37,9	207,863	-28,3	15,11	3,30	471,499
29	38,1	208,373	-27,1	15,74	3,80	471,153
30	38,025	208,182	-28,3	15,12	1,98	470,161
31	37,95	207,991	-29,0	14,77	2,38	471,026
32	38,025	208,182	-28,4	15,08	2,73	470,959
33	38,025	208,182	-28,0	15,27	2,98	470,957
34	37,9	207,863	-28,1	15,23	2,50	470,533
35	38	208,118	-28,1	15,22	2,88	470,925
36	38,05	208,245	-28,4	15,07	3,00	471,250
37	37,95	207,991	-28,1	15,21	3,23	471,290
38	38,025	208,182	-27,7	15,41	3,38	471,171
39	38,025	208,182	-28,3	15,11	2,93	471,125
40	38,05	208,245	-27,7	15,41	2,85	470,645
41	37,975	208,054	-27,9	15,32	2,90	470,814
42	38,075	208,309	-28,0	15,28	2,93	470,890
43	37,95	207,991	-28,2	15,19	2,78	470,858
44	38,075	208,309	-28,1	15,24	3,00	471,016
45	38,025	208,182	-28,1	15,24	2,95	470,966
46	37,95	207,991	-28,9	14,83	3,38	471,935
47	38	208,118	-27,3	15,63	4,08	471,582
48	38,075	208,309	-27,9	15,31	2,00	469,931
49	37,85	207,736	-28,6	14,96	2,23	470,628

LT_temp_out_evap	H_evap_out	Sd_LT_comp	To_MT	Po_MT
-21,6	445,886	79,4	-1,2	33,76
-21,6	445,131	79,1	-1,2	33,74
-21,3	445,244	79,4	-1,2	33,81
-21,4	445,739	78,7	-1,0	33,94
-21,5	445,313	79,9	-1,0	33,94
-21,5	445,072	77,9	-0,9	34,03
-21,6	445,684	79,2	-1,2	33,79
-21,3	445,419	78,1	-1,2	33,74
-21,4	445,449	77,6	-0,8	34,13
-21,0	446,686	79,8	-1,3	33,67
-15,6	451,029	77,9	-0,8	34,13
-16,8	450,745	77,8	-0,9	34,08
-21,1	446,056	78,8	-1,3	33,67
-21,5	445,093	77,3	-1,0	33,92
-21,6	445,503	78,5	-1,0	33,99
-21,4	445,570	77,9	-1,3	33,70
-21,6	445,671	78,2	-1,1	33,90
-21,6	445,306	79,3	-1,0	33,94
-21,4	445,384	78,8	-1,1	33,90
-21,4	445,359	78,0	-1,2	33,81
-21,8	444,492	77,5	-0,9	34,01
-21,6	445,475	78,8	-1,0	33,99
-21,5	445,191	78,1	-1,1	33,90
-21,5	445,446	77,7	-1,0	33,97
-21,4	445,619	77,9	-1,1	33,88
-21,4	445,254	78,0	-1,1	33,88
-21,4	445,331	77,6	-0,9	34,03
-21,0	446,160	78,6	-1,3	33,70
-12,2	454,635	77,2	-0,8	34,13
-16,4	451,166	77,5	-0,9	34,03
-21,3	446,546	78,8	-1,2	33,81
-21,5	445,651	78,0	-1,1	33,88
-21,4	445,429	77,9	-0,9	34,03
-21,5	445,418	77,5	-1,3	33,72
-21,5	445,414	77,3	-1,0	33,94
-21,5	445,703	78,5	-0,9	34,06
-21,5	445,410	78,9	-1,2	33,76
-21,5	445,047	78,7	-1,1	33,90
-21,4	445,743	77,6	-0,9	34,01
-21,4	445,216	77,8	-1,0	33,92
-21,4	445,303	78,3	-0,9	34,03
-21,4	445,404	78,6	-0,8	34,10
-21,4	445,574	78,2	-1,0	33,99
-21,5	445,421	78,2	-1,0	33,92
-21,5	445,393	79,7	-0,2	34,72
-21,2	446,485	79,0	-1,7	33,34
-16,6	450,078	78,6	-1,0	33,99
-15,3	452,075	77,8	-0,2	34,70
-20,6	446,968	78,0	-1,6	33,38

H_discharge_LT_comp	deltaH_LT_comp	n_LT_comp	LT_comp kW	massestrøm_LT
531,070	59,655	1,000	2,25	0,038
530,718	59,662	1,000	2,38	0,040
531,010	60,014	1,000	2,23	0,037
530,219	58,854	1,000	2,16	0,037
531,462	59,690	1,000	1,77	0,030
529,224	57,387	1,000	2,81	0,049
530,868	59,461	1,000	2,81	0,047
529,714	59,517	1,000	2,42	0,041
528,916	57,950	1,000	2,49	0,043
531,587	59,264	1,000	1,81	0,030
529,181	57,508	1,000	2,16	0,037
529,162	58,861	1,000	3,02	0,051
530,505	59,712	1,000	2,45	0,041
528,728	57,557	1,000	2,07	0,036
529,920	58,945	1,000	3,02	0,051
529,563	58,938	1,000	2,26	0,038
529,725	58,882	1,000	2,63	0,045
530,801	59,594	1,000	2,36	0,040
530,333	59,052	1,000	2,11	0,036
529,583	57,770	1,000	1,62	0,028
528,897	57,138	1,000	2,88	0,050
530,211	58,937	1,000	3,26	0,055
529,566	59,653	1,000	2,90	0,049
529,064	58,406	1,000	2,63	0,045
529,399	58,415	1,000	2,40	0,041
529,451	58,621	1,000	2,36	0,040
528,906	58,242	1,000	2,62	0,045
530,250	58,752	1,000	1,88	0,032
528,465	57,312	1,000	2,21	0,039
528,827	58,666	1,000	3,11	0,053
530,429	59,403	1,000	2,77	0,047
529,451	58,492	1,000	2,46	0,042
529,224	58,267	1,000	2,35	0,040
529,097	58,564	1,000	2,88	0,049
528,685	57,760	1,000	2,43	0,042
529,895	58,645	1,000	2,09	0,036
530,516	59,225	1,000	2,28	0,039
530,254	59,083	1,000	2,26	0,038
529,003	57,878	1,000	2,68	0,046
529,284	58,639	1,000	2,45	0,042
529,647	58,833	1,000	2,61	0,044
529,992	59,102	1,000	2,62	0,044
529,656	58,797	1,000	2,42	0,041
529,681	58,666	1,000	2,39	0,041
530,609	59,643	1,000	2,46	0,041
530,945	59,010	1,000	1,85	0,031
530,053	58,470	1,000	1,85	0,032
528,661	58,730	1,000	3,33	0,057
529,830	59,202	1,000	2,80	0,047

deltaH_LT_evap	Q_LT	COP_LT	MT_suction temp	h_inlet_MT_comp
237,831	8,980	3,987	10,63	451,511
237,332	9,468	3,978	10,80	451,799
237,126	8,801	3,951	11,20	452,244
237,557	8,709	4,036	11,53	452,461
237,259	7,045	3,975	10,18	450,540
236,636	11,566	4,124	12,60	453,810
237,756	11,216	3,999	13,43	455,377
237,301	9,659	3,987	12,93	454,766
236,886	10,158	4,088	12,70	453,792
238,759	7,272	4,029	11,13	452,378
242,847	9,100	4,223	12,40	453,373
242,627	12,438	4,122	13,90	455,532
238,257	9,776	3,990	11,45	452,834
236,975	8,512	4,117	11,58	452,572
237,258	12,166	4,025	13,73	455,445
237,771	9,107	4,034	13,38	455,461
237,553	10,621	4,034	14,80	457,062
237,124	9,371	3,979	13,10	454,661
237,329	8,460	4,019	12,10	453,348
237,432	6,668	4,110	10,33	451,002
236,310	11,901	4,136	13,38	454,925
237,357	13,109	4,027	14,20	456,096
237,073	11,505	3,974	13,38	455,117
237,264	10,684	4,062	12,28	453,475
237,564	9,750	4,067	11,95	453,178
237,200	9,559	4,046	12,65	454,154
236,959	10,649	4,069	12,73	453,984
238,296	7,605	4,056	10,70	451,738
246,262	9,507	4,297	12,25	453,162
242,985	12,860	4,142	13,60	455,196
238,555	11,114	4,016	11,63	452,842
237,470	9,987	4,060	12,13	453,423
237,247	9,579	4,072	11,08	451,662
237,554	11,682	4,056	11,18	452,369
237,296	9,993	4,108	12,03	453,164
237,458	8,452	4,049	11,70	452,508
237,419	9,150	4,009	10,18	450,870
236,866	9,040	4,009	10,55	451,159
237,561	10,990	4,105	11,28	451,987
236,971	9,901	4,041	10,78	451,439
237,249	10,515	4,033	10,60	450,984
237,096	10,500	4,012	10,55	450,789
237,583	9,779	4,041	10,55	450,995
237,112	9,650	4,042	10,60	451,190
237,212	9,774	3,977	11,00	450,310
238,494	7,487	4,042	8,85	449,755
241,960	7,635	4,138	10,05	450,277
243,766	13,811	4,151	13,30	453,641
239,232	11,325	4,041	9,95	451,242

T_outlet_MT_evap	H_outlet_evap	deltaH_MT_evap	Sd_HPS	Pd_HPS
5,4	443,777	235,722	76,63	72,8
5,1	443,360	235,560	75,40	72,0
5,0	443,083	234,965	75,70	72,3
4,7	442,348	234,167	76,05	72,3
4,3	441,618	233,564	74,20	72,4
4,3	441,562	233,126	77,58	72,9
4,4	442,193	234,266	79,83	73,2
5,3	443,707	235,589	78,15	71,7
5,7	443,550	234,987	76,63	72,0
4,7	442,841	234,914	77,43	73,4
4,0	440,886	232,705	76,83	72,1
4,0	440,965	232,847	82,03	74,8
3,9	441,606	233,806	82,10	76,2
3,7	440,750	232,632	79,60	74,6
4,0	441,122	232,877	82,38	75,7
4,1	441,913	234,113	82,93	75,3
6,0	444,421	236,303	82,20	73,6
6,8	445,624	237,442	79,48	71,6
5,0	442,816	234,761	73,28	69,5
4,1	441,691	233,764	67,98	66,8
4,0	441,033	232,851	69,50	66,3
4,2	441,381	233,263	72,48	67,0
4,4	441,873	233,755	70,88	66,6
4,6	442,064	233,882	68,85	66,4
4,2	441,724	233,670	67,93	65,6
4,0	441,347	233,293	68,78	65,8
4,1	441,204	232,831	69,98	66,3
4,3	442,149	234,285	69,58	67,3
4,0	440,846	232,474	70,25	68,2
4,1	441,104	232,922	75,93	70,4
4,7	442,576	234,585	76,95	72,1
5,5	443,775	235,593	75,20	71,5
5,9	444,047	235,865	74,80	71,7
5,0	443,232	235,369	74,55	70,7
4,4	441,875	233,757	73,65	70,3
4,3	441,513	233,268	73,55	70,4
4,5	442,397	234,407	74,85	71,3
7,1	446,034	237,853	74,45	72,1
8,0	447,290	239,108	76,98	73,0
6,1	444,605	236,359	76,90	73,1
5,5	443,311	235,257	78,70	74,8
5,6	443,345	235,036	79,95	75,9
5,4	443,250	235,259	78,65	75,5
5,3	443,294	234,985	80,13	76,3
5,6	442,085	233,903	80,40	77,4
5,7	445,083	237,093	82,13	76,9
5,7	443,716	235,597	78,48	76,3
6,0	442,709	234,400	82,63	77,3
6,1	445,633	237,897	87,88	80,3

H_dis_MT_Comp	deltaH_MT_comp	Tot_kW_comp_MT	n_MT_comp_tot
494,304	42,794	9,94	1,0
493,318	41,519	8,90	1,0
493,475	41,232	8,71	1,0
493,974	41,513	8,50	1,0
491,273	40,733	8,06	1,0
495,605	41,794	9,41	1,0
498,438	43,061	9,09	1,0
497,558	42,792	7,62	1,0
495,106	41,314	8,24	1,0
494,861	42,483	8,01	1,0
495,268	41,895	7,29	1,0
500,034	44,502	9,76	1,0
498,828	45,994	9,68	1,0
496,852	44,280	7,80	1,0
499,676	44,231	9,87	1,0
500,881	45,420	8,69	1,0
501,350	44,288	7,36	1,0
499,444	44,784	7,39	1,0
492,859	39,510	7,60	1,0
488,030	37,028	6,67	1,0
490,788	35,864	7,59	1,0
494,203	38,107	8,71	1,0
492,380	37,263	8,25	1,0
489,744	36,270	8,27	1,0
489,222	36,044	7,64	1,0
490,235	36,082	7,31	1,0
491,377	37,393	7,88	1,0
489,862	38,124	7,84	1,0
489,865	36,703	7,67	1,0
495,624	40,428	9,94	1,0
495,492	42,651	10,50	1,0
493,551	40,129	9,22	1,0
492,784	41,121	9,86	1,0
493,419	41,051	9,49	1,0
492,566	39,401	8,60	1,0
492,274	39,766	9,20	1,0
493,301	42,431	9,82	1,0
491,909	40,749	9,60	1,0
494,584	42,597	10,98	1,0
494,404	42,965	10,86	1,0
495,308	44,324	11,49	1,0
496,095	45,306	11,72	1,0
494,635	43,640	11,73	1,0
495,966	44,777	11,40	1,0
495,341	45,031	12,01	1,0
498,239	48,484	12,67	1,0
493,584	43,307	10,19	1,0
498,581	44,939	12,10	1,0
503,325	52,084	14,09	1,0

massestrøm_HPS	h_HPV	x_equilbrioum_rec	massflow_fg	massflow_liq
0,232	305,292	0,438	0,102	0,130
0,214	301,657	0,422	0,091	0,124
0,211	300,864	0,418	0,088	0,123
0,205	304,822	0,436	0,089	0,115
0,198	302,338	0,425	0,084	0,114
0,225	305,938	0,440	0,099	0,126
0,211	308,404	0,453	0,096	0,116
0,178	300,476	0,416	0,074	0,104
0,199	300,734	0,417	0,083	0,116
0,188	307,127	0,447	0,084	0,104
0,174	302,476	0,425	0,074	0,100
0,219	282,624	0,336	0,074	0,146
0,210	285,027	0,348	0,073	0,137
0,176	282,348	0,335	0,059	0,117
0,223	284,153	0,343	0,076	0,147
0,191	283,529	0,341	0,065	0,126
0,166	316,936	0,491	0,082	0,085
0,165	301,257	0,420	0,069	0,096
0,192	289,643	0,368	0,071	0,122
0,180	280,595	0,327	0,059	0,121
0,211	278,637	0,318	0,067	0,144
0,228	281,719	0,332	0,076	0,153
0,221	280,120	0,325	0,072	0,149
0,228	278,792	0,319	0,073	0,155
0,212	277,109	0,311	0,066	0,146
0,203	277,564	0,313	0,063	0,139
0,211	279,727	0,322	0,068	0,143
0,206	283,058	0,339	0,070	0,136
0,209	285,984	0,350	0,073	0,136
0,246	294,369	0,389	0,096	0,150
0,246	301,657	0,422	0,104	0,142
0,230	298,618	0,408	0,094	0,136
0,240	300,094	0,415	0,099	0,140
0,231	295,219	0,393	0,091	0,140
0,218	293,849	0,387	0,084	0,134
0,231	293,745	0,386	0,089	0,142
0,231	299,843	0,414	0,096	0,136
0,236	300,476	0,416	0,098	0,137
0,258	308,029	0,450	0,116	0,142
0,253	309,588	0,457	0,116	0,137
0,259	282,843	0,337	0,087	0,172
0,259	284,345	0,343	0,089	0,170
0,269	281,966	0,333	0,090	0,179
0,255	284,867	0,346	0,088	0,167
0,267	285,915	0,351	0,094	0,173
0,261	285,912	0,351	0,092	0,170
0,235	284,163	0,343	0,081	0,155
0,269	285,538	0,349	0,094	0,175
0,270	291,007	0,375	0,101	0,169

massflow_MT	Q_MT	COP_MT	Sgc_tempHPS	Pgc_pressure	h_in_HRHX
0,093	21,855	2,199	27,33	72,20	494,304
0,084	19,759	2,221	26,93	71,58	493,318
0,086	20,140	2,314	27,08	71,43	493,475
0,079	18,454	2,172	27,13	72,13	493,974
0,084	19,640	2,437	26,90	71,70	491,273
0,077	17,975	1,910	27,43	72,30	495,605
0,068	16,028	1,763	27,65	72,65	498,438
0,063	14,876	1,953	26,88	71,35	497,558
0,073	17,254	2,095	26,85	71,40	495,106
0,074	17,332	2,165	27,63	72,48	494,861
0,062	14,533	1,995	26,95	71,73	495,268
0,094	21,983	2,252	28,48	74,75	500,034
0,096	22,503	2,325	29,08	75,58	498,828
0,081	18,889	2,423	28,30	74,18	496,852
0,095	22,208	2,251	28,88	75,35	499,676
0,088	20,566	2,366	28,70	75,05	500,881
0,040	9,437	1,282	27,95	73,48	501,350
0,056	13,341	1,806	26,83	71,50	499,444
0,086	20,163	2,655	25,53	68,83	492,859
0,093	21,740	3,262	24,18	66,30	488,030
0,094	21,869	2,883	23,98	65,68	490,788
0,097	22,719	2,610	24,63	66,65	494,203
0,101	23,585	2,861	24,23	66,15	492,380
0,110	25,801	3,121	23,85	65,73	489,744
0,105	24,501	3,209	23,53	65,18	489,222
0,099	23,043	3,153	23,73	65,33	490,235
0,098	22,792	2,892	24,23	66,03	491,377
0,104	24,401	3,111	24,90	66,93	489,862
0,097	22,591	2,944	25,08	67,80	489,865
0,097	22,668	2,281	26,30	70,03	495,624
0,096	22,441	2,138	27,05	71,58	495,492
0,094	22,140	2,401	26,48	70,98	493,551
0,100	23,577	2,392	26,73	71,28	492,784
0,091	21,419	2,258	26,38	70,23	493,419
0,092	21,454	2,495	26,15	69,90	492,566
0,107	24,852	2,701	26,10	69,88	492,274
0,097	22,772	2,318	26,95	71,23	493,301
0,099	23,620	2,461	26,90	71,35	491,909
0,095	22,814	2,078	27,50	72,60	494,584
0,095	22,540	2,076	27,58	72,80	494,404
0,127	29,983	2,611	28,48	74,58	495,308
0,126	29,505	2,519	28,93	75,43	496,095
0,138	32,466	2,768	28,33	74,60	494,635
0,126	29,597	2,596	29,03	75,48	495,966
0,132	30,880	2,571	29,50	76,78	495,341
0,138	32,754	2,586	29,35	76,13	498,239
0,123	28,984	2,845	29,00	75,90	493,584
0,119	27,839	2,300	29,48	76,95	498,581
0,122	28,978	2,057	31,13	80,08	503,325

h_out_HRHX	Q_HRHX	Ambient_temp	Q_LT,xSH	Q_MT,xSH	E_pump_DC	E_fan_Dc
305,292	40,505	21,43	0,96	0,72	1,70	1,51
301,657	40,464	21,40	1,03	0,71	1,70	1,51
300,864	39,427	21,55	0,96	0,79	1,70	1,52
304,822	37,429	21,80	0,94	0,80	1,70	1,51
302,338	42,539	21,83	0,79	0,75	1,70	1,51
305,938	40,048	22,00	1,31	0,94	1,70	1,52
308,404	33,817	22,23	1,21	0,90	1,69	1,51
300,476	39,296	22,38	1,01	0,70	1,70	1,51
300,734	36,625	22,55	1,09	0,75	1,69	1,52
307,127	32,645	22,60	0,78	0,70	1,69	1,51
302,476	42,293	22,83	0,77	0,78	1,69	1,51
282,624	45,745	23,83	1,00	1,38	1,69	1,51
285,027	37,637	24,88	1,01	1,08	1,69	1,51
282,348	47,841	25,25	0,94	0,96	1,69	1,51
284,153	41,247	25,43	1,31	1,37	1,69	1,51
283,529	36,121	25,43	0,96	1,19	1,69	1,51
316,936	30,421	24,93	1,13	0,50	1,69	1,51
301,257	38,097	25,35	1,02	0,51	1,70	1,51
289,643	36,578	22,85	0,92	0,90	1,70	1,51
280,595	43,871	20,80	0,74	0,87	1,70	1,51
278,637	48,463	20,75	1,37	1,30	1,70	1,51
281,719	47,015	20,63	1,42	1,43	1,70	1,51
280,120	48,383	20,20	1,20	1,34	1,70	1,51
278,792	44,685	20,13	1,14	1,26	1,69	1,51
277,109	42,958	20,05	1,04	1,20	1,70	1,52
277,564	44,817	20,20	1,03	1,26	1,70	1,51
279,727	43,539	20,65	1,14	1,25	1,69	1,51
283,058	43,232	21,00	0,81	1,00	1,70	1,51
285,984	50,116	21,75	0,64	1,20	1,69	1,51
294,369	49,534	22,28	1,01	1,37	1,69	1,49
301,657	44,536	22,68	1,14	0,98	1,69	1,49
298,618	46,729	23,35	1,06	0,91	1,69	1,50
300,094	44,522	23,65	1,03	0,76	1,69	1,51
295,219	43,260	22,18	1,24	0,83	1,70	1,50
293,849	45,986	21,78	1,07	1,04	1,69	1,50
293,745	45,958	22,48	0,91	1,17	1,70	1,51
299,843	45,564	22,45	1,00	0,82	1,68	1,49
300,476	49,345	22,60	1,00	0,51	1,69	1,50
308,029	47,144	23,38	1,17	0,45	1,69	1,50
309,588	47,889	22,15	1,06	0,65	1,69	1,49
282,843	54,938	23,33	1,13	0,98	1,68	1,48
284,345	56,904	24,33	1,13	0,93	1,68	1,49
281,966	54,157	24,68	1,04	1,07	1,68	1,49
284,867	56,313	25,58	1,04	0,99	1,68	1,49
285,915	54,717	25,40	1,05	1,09	1,68	1,50
285,912	49,948	25,73	0,80	0,65	1,69	1,49
284,163	56,398	25,58	0,68	0,81	1,69	1,49
285,538	57,624	25,80	1,01	1,30	1,68	1,55
291,007	55,073	25,93	1,12	0,68	1,68	1,68

COP_cooling	COP_MT	COP_LT	COP_tot
2,47	2,39	1,43	2,00
2,56	2,47	1,46	2,02
2,57	2,48	1,46	2,05
2,47	2,37	1,44	1,96
2,50	2,41	1,44	2,05
2,52	2,39	1,46	1,92
2,39	2,26	1,40	1,80
2,42	2,31	1,42	1,85
2,56	2,45	1,48	1,97
2,33	2,24	1,38	1,89
2,40	2,28	1,45	1,87
2,84	2,67	1,60	2,15
2,67	2,55	1,51	2,11
2,66	2,54	1,52	2,10
2,84	2,67	1,57	2,14
2,68	2,53	1,51	2,10
2,05	1,95	1,27	1,52
2,29	2,21	1,36	1,75
2,82	2,70	1,56	2,22
3,04	2,92	1,64	2,47
3,38	3,19	1,75	2,47
3,25	3,05	1,70	2,36
3,28	3,11	1,70	2,44
3,39	3,23	1,75	2,59
3,36	3,21	1,74	2,59
3,32	3,15	1,72	2,53
3,23	3,06	1,70	2,44
3,06	2,94	1,64	2,48
3,12	2,96	1,74	2,45
2,89	2,72	1,63	2,19
2,61	2,50	1,49	2,04
2,75	2,64	1,54	2,16
2,68	2,59	1,52	2,15
2,77	2,67	1,55	2,13
2,85	2,71	1,58	2,21
2,85	2,72	1,57	2,30
2,60	2,51	1,48	2,09
2,67	2,62	1,50	2,17
2,50	2,45	1,46	2,01
2,43	2,37	1,42	1,97
2,91	2,82	1,59	2,35
2,83	2,74	1,56	2,29
2,98	2,88	1,62	2,44
2,83	2,74	1,57	2,31
2,82	2,72	1,55	2,30
2,63	2,58	1,50	2,27
2,85	2,77	1,60	2,41
2,87	2,74	1,63	2,23
2,41	2,36	1,42	1,99

Appendix E

Advansor calculator, Kiwi Tertnes, performed by Kelvin As:

EES Ver. 10.360: #4846