NTNU
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Minimering av energiforbruk i fremtidens supermarkeder

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Master i produktutvikling og produksjon
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Norges teknisk-naturvitenskapelige universitet
Institutt for energi- og prosessteknikk

NTNU
Norwegian University
of Science and Technology

Department of Energy
and Process Engineering

EPT-M-2013-68

MASTER THESIS

for

student Erik Junge Klemsdal

Spring 2013

Minimization of energy consumption of new supermarkets

Minimering av energiforbruk i fremtidens supermarkeder

Background and objective

Supermarkets are commercial buildings with major energy consumption and contribute also to relatively large direct emissions of greenhouse (GHG) through emissions of refrigerants from the refrigeration plants and the air conditioning system installed.

Heat recovery systems in supermarkets have in the past been very simple with heat distribution in the ventilation systems, or systems were the heating system and refrigeration systems are separated and the energy from the refrigeration system have been cooled at dry coolers at the roof. There is a growing interest to look into the energy systems and heat recovery in supermarkets due to the high energy consumption per square meter floor surface (annually more than 500 kWh/m^2). The thesis will focus on the heat recovery system with floor heating and possibilities to utilize the heat capacity of the floor to take the daily variations.

The thesis work will follow in parallel with the design and build process of the new supermarket closed to Trondheim (Kroppanmarka). A calculation model has been developed during the past years. This tool should be further adapted.

The following tasks are to be considered:

1. Literature study of the topic in the scope of work
2. Applying Modelica and perform refrigeration and HVAC system simulations (day-cycles)
3. An estimation of the energy consumption when applying different control strategies at various operation conditions
4. Analyzing different control strategies and the consequences for energy consumption for the different system solutions
5. Write a scientific paper with the main results from the thesis
6. Make proposal for future task not solved in this thesis

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Within 14 days of receiving the written text on the master thesis, the candidate shall submit a research plan for his project to the department.

When the thesis is evaluated, emphasis is put on processing of the results, and that they are presented in tabular and/or graphic form in a clear manner, and that they are analyzed carefully.

The thesis should be formulated as a research report with summary both in English and Norwegian, conclusion, literature references, table of contents etc. During the preparation of the text, the candidate should make an effort to produce a well-structured and easily readable report. In order to ease the evaluation of the thesis, it is important that the cross-references are correct. In the making of the report, strong emphasis should be placed on both a thorough discussion of the results and an orderly presentation.

The candidate is requested to initiate and keep close contact with his/her academic supervisor(s) throughout the working period. The candidate must follow the rules and regulations of NTNU as well as passive directions given by the Department of Energy and Process Engineering.

Risk assessment of the candidate's work shall be carried out according to the department's procedures. The risk assessment must be documented and included as part of the final report. Events related to the candidate's work adversely affecting the health, safety or security, must be documented and included as part of the final report. If the documentation on risk assessment represents a large number of pages, the full version is to be submitted electronically to the supervisor and an excerpt is included in the report.

Pursuant to "Regulations concerning the supplementary provisions to the technology study program/Master of Science" at NTNU §20, the Department reserves the permission to utilize all the results and data for teaching and research purposes as well as in future publications.

The final report is to be submitted digitally in DAIM. An executive summary of the thesis including title, student's name, supervisor's name, year, department name, and NTNU's logo and name, shall be submitted to the department as a separate pdf file. The final report in Word and PDF format, sientific paper and all other material and documents should be given to the academic supervisor in digital format on a DVD/CD-rom at the time of submission in DAIM.

☐ Work to be done in lab (Water power lab, Fluids engineering lab, Thermal engineering lab)
☒ Field work

Department of Energy and Process Engineering, January 16th 2013

Olav Bolland

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Preface

This represents my last work at NTNU and I feel that I have learned more in these last 4 months than the first 4 years of my time here. I've certainly studied more.

In this thesis I have worked closely with Maren Titze, though not in the geographical sense. Communication has mostly been via e-mails and Skype between Braunschweig and Trondheim which have not always been easy, but we have managed to overcome both the geographical and linguistic difficulties. I would like to thank Maren specially for helping me during my work. It would not be possible without her.

I would also like to thank Trygve Eikevik who has been my academic supervisor. He has been very supportive and made sure I was always on the right track. During the work I also encountered interesting and complicated aspects of refrigeration technology where I was not able to find the right answers on my own. In these situations I turned to my co-supervisor at SINTEF, Armin Hafner, whose door was always open. I owe him a big thanks for helping when I needed it. Although not made an official co-supervisor in this thesis I have also gotten much help from Ola Jonassen at Asplan Viak. He always had time to talk and happily shared his experience to which I am truly grateful.

Lastly I want to thank my lovely girlfriend Malin who has listened to engineer ramble for too long and pretended to be interested.

Trondheim, June 2013

Erik Junge Klemsdal

Erik Junge Klemsdal

Summary

Background:

Supermarkets are energy intensive buildings responsible for large amounts of greenhouse gas emissions. In order to reduce both the energy consumption and the environmental impact of these buildings new efficient system solutions using green refrigerants have been introduced. Hydrofluorcarbons (HFC's) are today the most common in refrigeration, but due to their high global warming impact there are now many incentives to use natural refrigerants instead. In colder climates, such as in Norway, there is also substantial need for space heating in supermarkets. Heat integration between the refrigeration plant and the ventilation system is a widely demonstrated solution with a high potential for energy savings. Due to the complicated thermodynamic interactions between the different subsystems in a supermarket, efficient heat integration may often prove difficult. In the last decade has seen great improvements in predictive models and simulations tools for supermarkets and their system solutions. The results from these simulations are only as good as the input data from which they are generated, which is why the results need to be interpreted with caution.

Design and Methods:

SINTEF has in collaboration with the supermarket franchise REMA 1000 started the building of a new high efficiency green supermarket in Trondheim. This is a pilot project that may become the new standard for future stores. Researchers from SINTEF have been involved in every part of the design of the supermarket with the goal of making it as efficient and environmentally friendly as possible while still accommodate the needs of a sales market. The key to achieve this goal is smart heat integration. The heart of the system is a transcritical CO_2 booster system which provides heat to both a floor heating circuit and the ventilation system while still generating the necessary refrigeration for the sales products. An energy well provides free cooling and can work as an extra evaporator for the refrigeration plant while surplus heat can be stored in accumulation tanks.
A model of the refrigeration plant and the ventilation system of this new supermarket has been created. Input data for both models have been generated in this thesis. Simulations of the refrigeration plant were conducted to investigate the efficiency and annual energy consumption at varying conditions and control strategies for the plant. This was tested against two cases of annual heat demand. The heat available to the floor

heating circuit is dependent on the water return temperature. This is an unknown temperature which is why three cases were simulated to test the different outcomes. There were three possible modes of operation of the refrigeration plant, each with an associated heat production, efficiency and power consumption.

Results and discussion:

The results suggest that in the worst case of floor return temperature the new system is capable of saving 50-60 % of the total energy needed for refrigeration and space heating compared to a system without heat recovery. For the most favourable floor return temperature tested the savings were only around 3% higher. This is due to the fact that the majority of heat recovered goes to the other gas cooler which is not affected by the floor return temperature. The results also suggest that the extra evaporator removes the need for additional electrical heating. The seasonal performance factor was estimated above 4.6 for the least favourable scenario. The main reason for these promising results is the presence of the heat accumulation tanks that minimizes waste of available heat. The special properties of a transcritical CO_2 cycle is also well utilized by rejecting the heat across three gas coolers in series providing both a large heating capacity and sufficient cooling before expansion of the CO_2. This leads to a high total coefficient of performance. The positive results are affected by the model considering a near ideal situation, but this is to a large extent outweighed by the fact that the simulated operational modes of the plant are far from optimized, but rather extremes of possible operation. This was done to test the boundaries of the plant. There is therefore reason to believe that the actual energy savings could be in the range of the estimated values. This is still only results from simulations based on constructed input data which will always be associated with a degree of uncertainty.

Conclusion:

The present study suggests that the new supermarket system designed by SINTEF is capable of considerable energy savings. This is achieved due to an excellent refrigeration- and heat recovery system. The models should be further improved to reduce the number of simplifications and assumptions of ideal thermodynamic processes to enhance the credibility of the results. Simulations with optimized cycle operation will also be very interesting to perform so that the true potential of the new system can be investigated.

Sammendrag

Bakgrunn:
Dagligvarebutikker er energikrevende bygninger som er ansvarlige for store mengder utslipp av klimagasser. For å redusere både energiforbruk og miljøpåvirkning fra disse bygningene har nye effektive systemløsninger blitt introdusert. Hydrofluorkarbongasser er i dag de mest vanlige arbeidsmediene i kuldeanlegg, men på grunn av høyt globalt oppvarmingspotensial er det nå kommet mange insentiver for å bruke naturlige kuldemedier i stedet. I kjølige klimaer, som i Norge, er det også et stort oppvarmingsbehov i butikker. Varmeintegrasjon mellom kjøle- og ventilasjonsanlegg er en bredt praktisert løsning med et stort potensiale for energisparing. På grunn av de kompliserte termodynamiske interaksjonene mellom de ulike undersystemene i dagligvarebutikker, kan effektiv varmeintegrasjon ofte være vanskelig å få til. Det siste tiåret har sett store forbedringer i prediktive modeller og simuleringsverktøy for supermarkeder og deres systemløsninger. Resultatene fra disse simuleringene er bare så gode som de inndata de er generert fra, noe som er viktig å huske på når man skal trekke konklusjoner.

Design og Metode:
SINTEF har i samarbeid med butikkjeden REMA 1000 startet byggingen av en ny høyeffektiv og miljøvennlig butikk i Trondheim. Dette er et pilotprosjekt som kan bli den nye standarden for fremtidige butikker. SINTEF-forskere har vært involvert i alle deler av utformingen av butikken med mål om å gjøre den så effektivt og miljøvennlig som mulig. Nøkkelen til å oppnå dette målet er smart varmeintegrering. Kjernen i systemet er et transkritisk CO_2-booster-system som gir varme både til en gulvvarmekrets og til ventilasjonsanlegget, samtidig som nødvendig kjøling av varer blir levert. En energibrønn gir frikjøling og kan fungere som en ekstra fordamper for varmepumpefunksjon av kjøleanlegget, mens varmeakkumulasjonstanker sørger for at minimalt med varme går til spille.
Det har blitt laget simuleringsmodeller av kjøleaggregatet og ventilasjonsanlegget til dette nye supermarkede. Inndata for begge modeller har blitt generert som del av denne masteroppgaven. Simuleringer av kjøleanlegget har også blitt utført for å undersøke effektiviteten og det årlige energiforbruket ved varierende forhold og kontrollstrategier for anlegget. Dette ble testet mot to tilfeller av årlig oppvarmingsbehov. Den

tilgjengelige varmen til gulvvarmekretsen avhenger av returtemperaturen til vannet. Dette er en ukjent verdi og det ble derfor gjennomført simuleringer for tre forskjellige temperaturer. Tre driftsmetoder for kjøleanlegget var mulig i simuleringene.

Resultater og diskusjon:
Resultatene tyder på at det nye systemet er i stand til å spare 50-60% av det totale energibehovet for kjøling og oppvarming sammenlignet med et system uten varmeintegrering ved det minst gunstige tilfellet av returtemperatur. Ved det mest gunstige tilfellet var energisparingen bare 3% større. Årsaken er at majoriteten av den gjenvunne varmen går til den andre varmeveksleren som er uavhengig returtemperaturen på vannet i gulvvarmekretsen. Resultatene viser også at ved bruk av energibrønnen som ekstra fordamper unngår man behovet for elektrisk oppvarming. Den årlige ytelsesfaktoren er estimert til over 4.6 selv under de minst gunstige forholdene simulert. Hovedårsaken til disse lovende tallene er takket være varmeakkumulasjonstankene som sørger for at så mye som mulig av den tilgjengelige varmen blir brukt. I tillegg blir de gode egenskapene til den transkritiske CO_2 prosessen godt utnyttet ved å avgi varme til tre gasskjølere i seriekobling. Dette fører både til stor varmekapasitet og sørger for at arbeidsmediet blir nedkjølt tilstrekkelig før struping noe som gir høy total effektivitetsgrad. De positive resultatene er også påvirket av at simuleringsmodellen antar en nær ideell prosess, men dette er til en stor grad oppveid av det faktum at driften av kjøle- og varmepumpeprosessene er langt fra optimert i simuleringene. Det er derfor grunn til å tro at den faktiske energisparingen kan være i nærheten av de beregnede tallene. Det er allikevel viktig å huske at disse resultatene er produsert i et simuleringsverktøy basert på konstruerte verdier som det alltid vil være knyttet en grad av usikkerhet til.

Konklusjon:
Denne studien viser at det nye varme- og kjølesystemet designet av SINTEF kan oppnå en betydelig energisparing. Dette er takket være et spesielt godt kjøle- og varmegjenvinningssystem. Simuleringsmodellene bør videreutvikles for å redusere antall forenklinger og antagelser om ideelle termodynamiske prosesser for å styrke resultatenes troverdighet. Det vil også være interessant å optimere driften av anlegget i simuleringene slik at det fulle potensialet til det nye systemet kan undersøkes.

Table of Contents

List of figures

List of tables

Abreviations

In order of appearance

HFC	HydroFluorCarbon
CFC	ChlorFlourCarbon
GWP	Global Warming Potential
LCC	Life Cycle Cost
LCA	Life Cycle Assessment
ODP	Ozone Depletion Potential
LT	Low Temperature
MT	Medium Temperature
HT	High Temperature
COP	Coeffecient of Performance
DHW	Domestic Hot Water
SH	Space Heating
AHU	Air Handling Unit
GC	Gas-Cooler
HVAC	Heating Ventilation & Air Conditioning
RH	Relative Humdity
P	Pressure
H	Enthalpy
T	Temperature
HSE	Health, Safety and Environment
SPF	Seasonal Performance Factor
STD	Standard
HP	High Pressure
EE	Extra Evaporator

1 INTRODUCTION

Supermarkets are one of the most energy intensive buildings in the commercial sector. In industrialized countries the power consumption represents 3-5% of the total annual consumption [1]. With huge cooling demands for food storage the industry applies great amounts of refrigerant. It is estimated that supermarkets are responsible for 28% of the world's refrigerant consumption [2]. Annual average leakage is in the range of 15-30% [3], with Europe being in the lower part of the range. The most common refrigerants in Europe are the HFC gases R134a and R404a, both of which have a high global warming potential [4]. In countries like Norway and Denmark HFC gases have a CO_2 equivalence tax of 0.225 kr/kg [5]. Climate change, environmental damage and depletion of natural resources followed by more laws and regulations has forced engineers to come up with smart solutions for supermarkets that are both energy efficient and environmentally friendly. This has increased interest for natural working fluids. CO_2 is a safe and green refrigerant with special properties that can be utilized in refrigeration and heat recovery. A new supermarket is being built in Trondheim in collaboration with SINTEF and their CREATIV project. A transcritical CO_2 booster system has been selected to provide both refrigeration and heating for the whole store. The complete system is carefully designed to optimize efficiency and reduce environmental damage.

A literature review of working fluids and system solutions in supermarkets are presented in this thesis. The new system designed by SINTEF is presented. A program in excel has been created to produce reliable input data for a model of the ventilation system at the new supermarket. The results are evaluated and discussed. Simulations of the refrigeration plant have been performed and the results are presented and used to make estimations on annual energy consumption and efficiency for different control strategies of the plant.

Using and creating programs for data processing as been a major part of this thesis. A lot of hours have been spent in organizing simulation results to make them presentable. Some of this work is described in Appendix B.

2 OBJECTIVES

The main tasks of this thesis were to create reliable input data for the simulation models of the refrigeration and ventilation system in the new supermarket called Kroppanmarka, perform simulations and analyse the results. The goal is to obtain information on how the complete system at Kroppanmarka will perform under varying conditions and different control strategies with respect to efficiency end total energy consumption. The models have been created by Dipl.-Ing. Maren Titze at the Technical University of Braunschweig, Institute for Thermodynamics. The ventilation model was unfortunately not ready for simulations at the time when this thesis was written so no simulations of this model were performed. Uncertainty regarding of the readiness of the model was known at the beginning of the work. However, all the input data necessary for the model was still created. Simulations of the refrigeration plant were carried out as planned and the results are presented. Some of the specific tasks received were:

1. *Scenarios*
    a. *Create a set of weather- day- scenarios and a distribution of these scenarios so that they represent the ambient conditions for a whole year in Trondheim. Meteonorm should be used to download weather data on the parameters humidity, temperature and direct and diffuse vertical radiation for Trondheim over a whole year.*
    b. *Create 3 day scenarios and their distribution for the heat load on the cooling and freezing cabinets. The scenarios should be based on measurements from the plant at REMA 1000 Dragvoll.*
2. *Simulations of refrigeration plant*
    a. *Perform simulations for the cases of summer, winter and spring/fall while changing the return temperature from the floor heating under three different modes of operation for the plant.*
    b. *Create curves of available heat and COP for a whole year for each of the combinations of operational mode and return temperature.*
3. *Extra: Simulation of the ventilation System*
    a. *Use the scenarios created in task 1a. to create an annual heat demand curve for the store.*

3 LITERATURE REVIEW

The depletion of ozone in the stratosphere combined with the atmospheric greenhouse effect due to refrigerant emissions has led to dramatic changes in the development of refrigeration and air conditioning technology since the start of the 1990s. Until just a few years ago the most common refrigerants were R12, and R22 and R502, plus for special applications R114, R12B1, R13B1, R13 and R503. These are of the ozone depletion type which is why they have been phased out over the course of the last two decades. In 1993 CFC gases were banned in Norway through the Montreal protocol and since 2002 there has been a continuous decrease of HCFC import which today is also completely phased out [4]. These ozone depletion gases were then replaced by now well established HFC (chlorine free) refrigerants such as R134a, R404A, R507A, R407C and R410A which are all chemically engineered. These refrigerants are not without drawbacks either.

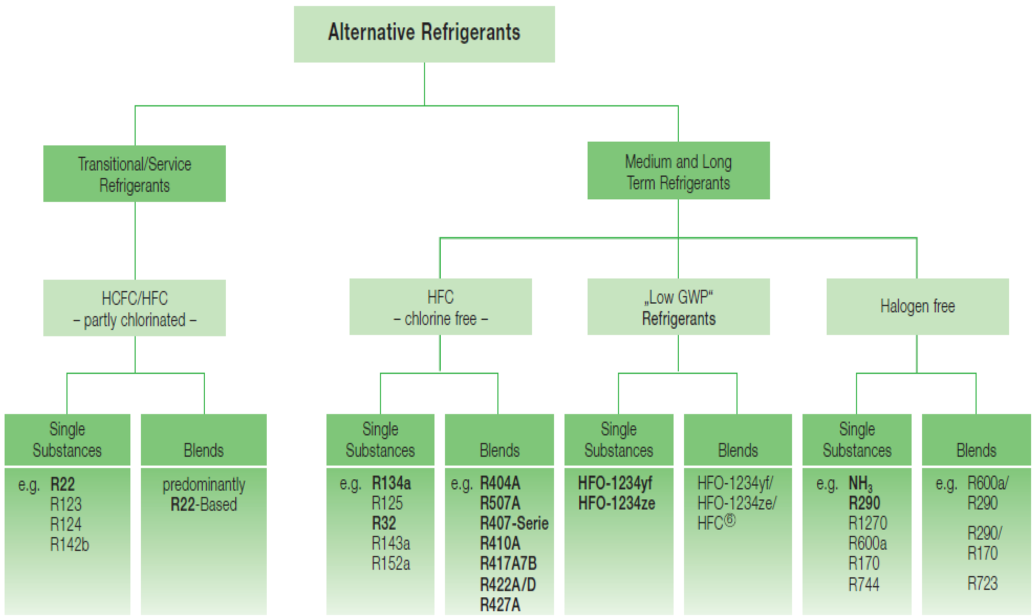

Figure 3-1 Refrigerants used in the industry [6]

How much a refrigerant contributes to global warming can be described through the GWP (Global Warming Potential). The release of a refrigerant is described as the equivalent release of CO_2. R134a has a GWP of 1430 which means that the release of 1 kg R134a has the same impact on global warming as the release of 1430 kg CO_2. This is based on a 100 year time horizon [6]. The GWP of R404A is about 3800. It then becomes evident why it is desireable to reduce leakage of such substances. Well sealed plants with lower refrigerant charge help achieve this goal, but also the effectiveness of the plant is important. The compressors and pumps of a refrigeration plant consume power. Although the majority of electrical power generated in Norway comes from clean hydropower (95%) [7] there is a high percentage of fossil fuels used in power stations across Europe. The average European CO_2 release is approximately 0.6 kg per kWh of electrical energy produced [6]. Consequently the energy consumption of a plant is (indirectly) a substantial contributor to its global warming impact. As a result of this, a significant greenhouse effect occurs over the lifetime of a plant in addition to the leakage. The use of a low GWP-refrigerant in an ineffective plant is in other words also environmentally unfriendly.

A method of taking both the direct and indirect contributions into account has been introduced. It is called TEWI which stands for Total Equivalent Warming Impact. With this method all of the factors contributing to global warming are considered. This can be quite useful in the process of trying to select a ecological solution for a refrigeration plant or a heat pump. When evaluating technologies and making investment decisions the economic aspects are naturally also important. It is often the decisive factor. Technical solutions resulting in a reduction of environmental impact is often associated with high cost, whereas low cost frequently involves increased ecological consequences. An ideal situation would be where both cost and environmental impact was reduced. A concept called *Eco-Efficiency* is considering both the life cycle cost analysis (LCC) which includes investment costs, cost of operation and capital cost, and life cycle assessment (LCA) which take into account the ecological performance which includes the direct and indirect emissions. It is in other words based on the relationship between a products economic value and the resulting environmental impact.

Figure 3-2 shows the economical aspect on the y-axis and the ecological on the x-axis with the degree of eco-efficiency as diagonal lines in the graph. Systems lying on the same diagonal line can then have different life cycle costs and environmental impact while exhibiting the same Eco-Efficiency.

$$TEWI = (GWP \times L \times n) + (GWP \times m[1 - \alpha_{recovery}]) + (n \times E_{annual} \times \beta) \quad (1)$$

L	=	Leakage per year [kg]
n	=	System operating time [years]
m	=	Refrigerant charge [kg]
$\alpha_{recovery}$	=	Recycling factor [-]
E_{annual}	=	Energy consumption per year [kWh]
β	=	CO_2-Emission [kg CO_2/ kWh]

[8]

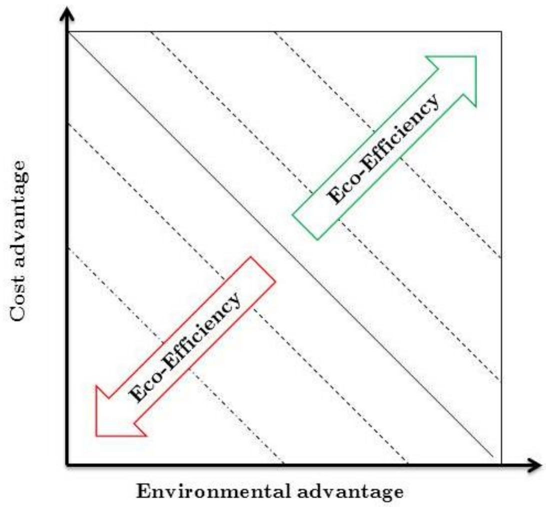

Figure 3-2 Eco-Efficiency diagram [6]

3.1 Natural refrigerants

Because of more strict regulations and environmental awareness natural refrigerants have in recent years increased in popularity as an alternative for refrigeration- and heat pump systems [4]. The most common are R717 (ammonia), R290 (propane), R600a (isobutene), R1270 (propylene) and R744 (CO_2). The advantage of these fluids is that they exist naturally in the biosphere and have GWP factors that are practically negligible or zero. At the same time they show great thermodynamic properties.

3.2 R717

Ammonia has been used as a refrigerant for more than a century in industrial and larger refrigeration plants. Both ODP (ozone depletion potential) and GWP is equal to zero and because of outstanding thermodynamic capabilities (see Table 3-1) the efficiency is as least as good as with R22 [6].

Ammonia is reference (=1.0). 1* indicates when "High" or "Low" values of the thermophysical properties are favorable for the heat transfer conditions in the heat exchangers. [9]

Properties	**1***	**R717**	**R134a**	**R507A**	**R407C**	**R410A**
Thermal conductivity, liquid	High	1.0	0.2	0.15	0.2	0.2
Thermal conductivity, vapour	High	1.0	0.5	0.6	0.5	0.6
Dynamic viscosity, liquid	Low	1.0	1.55	0.95	1.2	0.9
Dynamic viscosity, vapour	Low	1.0	1.2	1.25	1.3	1.4
Spesific enthalpy of evaporation	High	1.0	0.15	0.1	0.15	0.15
Density, liquid	Low	1.0	2	1.75	1.9	1.75
Density, vapour	Low	1.0	4.2	9.6	5.8	9.1
Ratio of liquid/vapour density	High	1.0	0.5	0.2	0.3	0.2
Spesific heat, liquid and vapour	High	1.0	0.3	0.35	0.35	0.4

Table 3-1 Thermophysical properties at saturated state in relative values for ammonia and selected HFCs.

The disadvantage of ammonia is that it is highly toxic and also inflammable. It is classified in group B2 in NS-EN 378 (2008). This means that special safety measures have to be made for the construction and operation of the plant which comes at an expense. Ammonia is also corrosive to copper containing materials which much be considered for tubes and other components. Another disadvantage is the high isentropic exponent which results in very high discharge temperatures. Two stage compression with generously sized oil coolers is required already at medium pressure levels. Ammonia is therefore not a candidate for conversion of existing HFC plants and everything must be constructed with completely new components. Ammonia is still a widely used refrigerant. In the last decade ammonia has been accepted in new chiller applications mainly because of technological improvements in safety. [2]

3.3 Hydrocarbons

Systems using propane (R290) and other hydrocarbons have been in operation worldwide for many years, especially in the industrial area, but also in smaller compact systems with low refrigerant charge. Pressure levels and refrigeration capacity is in the same range as for R22 and the temperature characteristics are as favourable as with R134a. There is no negative effect to the ozone layer in case of leakage and a negligible GWP of 3 makes it an environmentally friendly candidate. One obvious downside to the use of hydrocarbon is the high flammability. It is classified as safety group A3 and the corresponding flame-proof measures must be deployed in the design according to the regulations. Apart from this there aren't any special features required in the medium and low temperature ranges compared to plants using HFC. [6]

3.4 CO_2

CO_2 has been in refrigeration since the 19th century, but it had nearly disappeared by the 1950s. High operational pressure and low critical temperature being unfavourable for the usual applications in refrigeration was the main reason. The last ten years has brought back CO_2 due to a number facts. CO_2 is one of the most environmentally friendly refrigerants there is. The GWP is defined as 1 and the ODP is zero. It is not toxic in the classical sense, non-flammable, chemically inactive and it's very much available. These advantages combined with today's environmental awareness has made a number of producers of refrigeration equipment start making components specifically tailored for the properties of carbon dioxide. This has reduced some of the drawbacks while the good thermophysical properties have been further exploited.

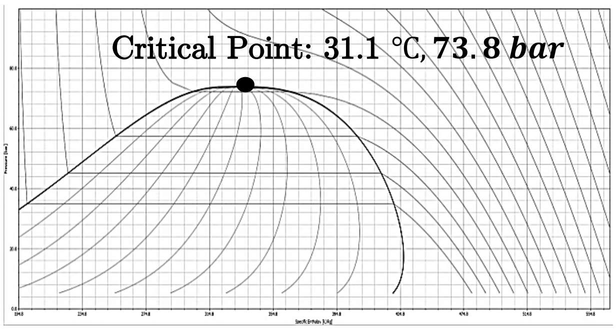

Figure 3-3 Pressure-Enthaply Diagram showing the critical point of CO_2

High operating pressure leads to high volumetric heating/cooling capacity and a smaller required compressor volume. It also means higher density which result in lower fluid velocity and therefore small pressure losses. The saturation pressure-temperature curve is also very steep which leads to even lower temperature drop per pressure drop. Required Tubes and components sizes are smaller and so material costs and radiation losses are also reduced. The minimization of tubes and components also leads to easier installation to cabinets and lower refrigerant charge. The compressor pressure ratio is small since the operational pressure is high which results in high compressor effectiveness. [10]

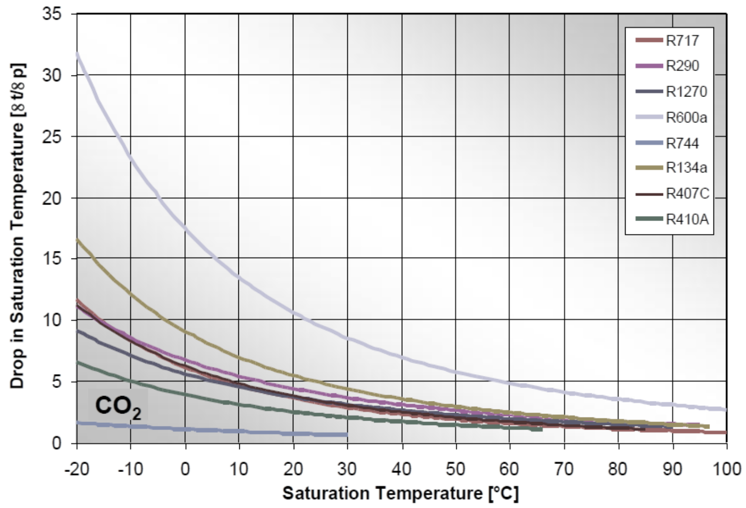

Figure 3-4 Temperature loss vs. pressure loss in saturated state for selected refrigerants [10]

3.5 CO_2 systems in supermarket refrigeration

The first CO_2 refrigeration systems that where commercially applied was for freezing temperatures in indirect systems. The main reasons for using CO_2 as a secondary working fluid are the simplicity of the systems and the possibility of using components for other refrigerants to build circuits. At freezing temperature CO_2 has a reasonable working pressure (11 bars at -37°C) and since it is poses no hazard or danger to the public it is safe to use it in the sales area. In indirect arrangements (Figure 3-5) the CO_2 is connected to the primary cycle via its evaporator. The CO_2 is condensed on one side while the primary refrigerant evaporates on the other. [11]

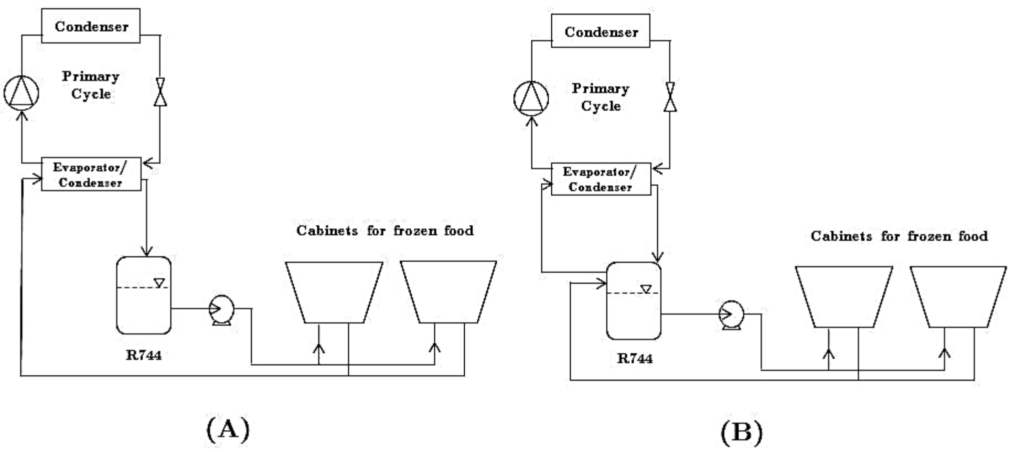

Figure 3-5 Two solutions of indirect arrangement

The system is usually equipped with a vessel that accumulates CO_2 returning from the evaporator/condenser. The arrangement in Figure 3-5 B is more favourable than A because of expected higher heat transfer coefficient and better stability under load variations. A pump circulates the CO_2 at a maintained circulation rate in the circuit which keeps the evaporators (cabinets) wet. A high circulation ratio gives a low vapour fraction at the outlet of the cabinets which ensures good heat transfer. This is also an advantage in terms of frost formation which is likely to be modest and more uniform. [12]

An indirect solution for the medium temperature has also been applied. CO_2 will then be operating at about 28-31 bar at corresponding saturation temperatures -8 to -4 °C. Components able to handle 40 bar are acceptable. Indirect systems will in general have higher energy consumption because of the temperature difference that exists in the additional heat exchanger compared to direct systems. This forces the evaporation temperature of the primary loop down which makes the compressor work harder for the same refrigeration capacity. In addition, one must add cost of running an extra pump for the secondary fluid. The temperature drop can be partly compensated by the low pressure drop in tubes and components associated with CO_2. [12]

Due to the growing interest in CO_2 as an alternative to the synthetic fluids components specially designed to handle CO_2 have become cheaper and more available. As a result there are now more system arrangements for this natural refrigerant in addition to the indirect solution. [12]

3.6 Cascade

Cascade systems using CO_2 at the low temperature level is today used in many supermarkets and it is becoming a competitive alternative to HFC. There are multiple arrangements possible. The chiller (refrigerant in the high temperature stage) is usually operatied with R290, R717 or R404A.

The medium temperature level can also be with CO_2 in an indirect manor. CO_2 systems will usually have lower COP especially at the high/medium temperature level because of the low critical point, but if it is used in both the low and medium stage the system will require one heat exchanger less. A single vessel can work as a receiver for both levels instead. Having only one heat exchanger means lower investment costs and the additional temperature differenceis avoided. For solution A in Figure 3-7 the hot discharge gas from the low stage compressors enters the cascade condenser after mixing with the saturated vapour from the vessel. The mixing de-superheats the discharge gas before being further de-superheated and finally condensed in the cascade condenser. The hot discharge gas could also pass through the tank and de-superheat by boiling of some of the liquid CO_2 in the tank (B). A third option (C) for the cascade joint is to have all heat exchange take place in the vessel by letting the high stage refrigerant evaporate in tubes submerged in the liquid CO_2. Direct expansion can be used in the low temperature level which requires a certain amount of superheat (E). While this gives slightly reduced heat transfer in the evaporator compared to a flooded solution like evaporator arrangement 1 A, it doesn't require the extra pump which will increase the power consumption. Another solution (D) is to use a vessel to sub-cool the refrigerant before expansion. This is achieved without superheating and the refrigerant entering the compressor is saturated vapour. For large scale installations the flooded solution is more likely to be selected. [12]

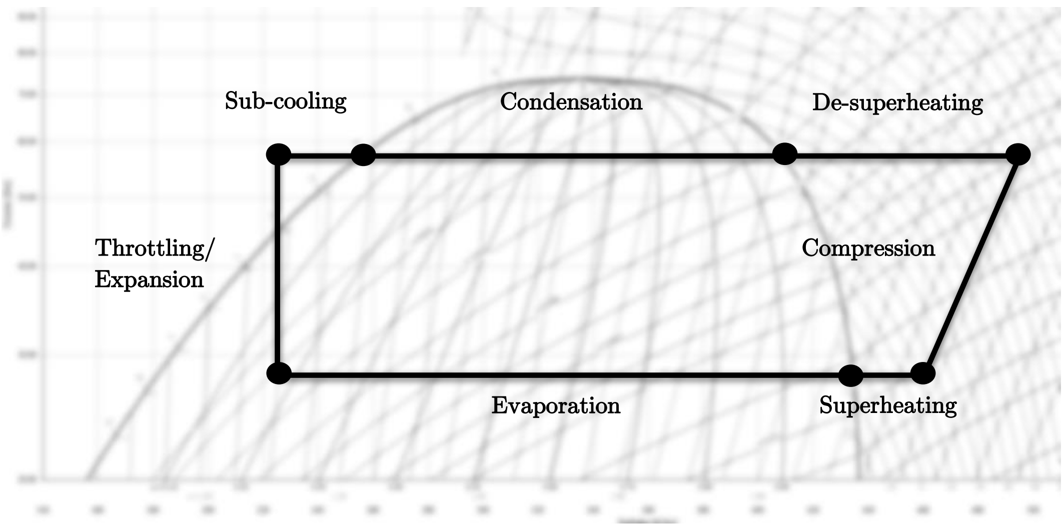

Figure 3-6 Main steps in a vapour compression cycle in a P-H diagram

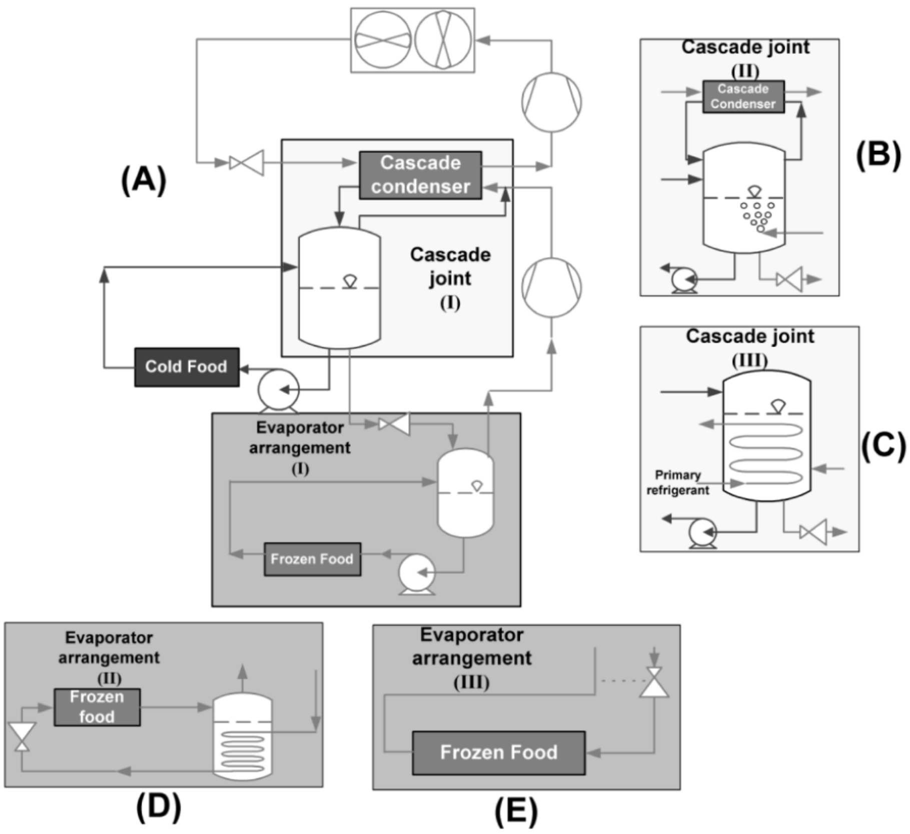

Figure 3-7 Variations of the cascade solution [12]

3.7 Systems with CO_2 only

The three main solutions for systems using only CO_2 in supermarkets are Parallel, Centralized and the Booster systems. The parallel solution includes two separate circuits that are installed with direct expansion at both the medium and low temperature level. The low temperature level uses two-stage compression for reduced discharge temperatures and compression losses. Each compressor stage then operates at a lower pressure ratio compared to one single stage which enhances efficiency. A centralized system means that the central refrigeration unit (high stage) is located in the machine room. An accumulator merges this circuit with both the medium and low temperature circuits so that all three are connected via the tank. The medium temperature cabinets are flooded and circulated with a pump while the low stage uses direct expansion in one stage.

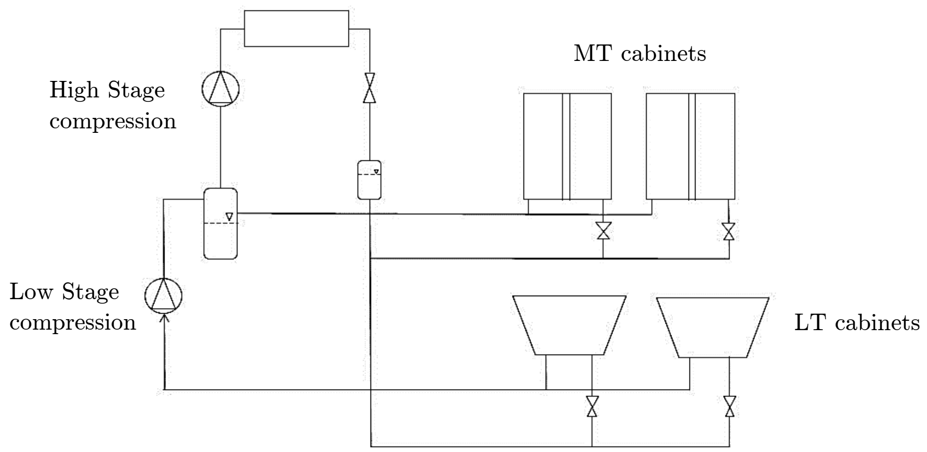

Figure 3-8 Simple Booster layout

The booster layout (Figure 3-8) has two compression stages; one for the low stage where the discharge pressure equals the evaporation pressure of the medium temperature cabinets and one high stage compression which must handle the loads of both the medium and low temperature levels. An advantage with this solutions compared to the cascade concept is the absence of the cascade condenser and the resulting temperature difference. A drawback is that the condensing pressure at the high stage will be much higher than for any other refrigerant which leads to increased energy consumption. This is especially true when the ambient temperature or heat sink is high. For this reason a booster solution has up until now only been common in colder climates where the condensation pressure can be lowered to reasonable levels. With the introduction of equipment such as ejectors and expanders this situation may change. An example of ambient temperature dependency is illustrated in Table 3-2. The data has been produced using CoolPack. Isentropic efficiency is assumed to be 0.7 for both compression stages, pressure losses have been neglected and evaporation occurs at -30 and -8℃ for the CO_2. A 5 K temperature difference across the condenser is assumed for all ambient temperatures.

Refrigeration Performance for CO_2 booster system					
Ambient Temperature [℃]	0	5	10	15	20
Refrigeration COP [-]	3.88	3.29	2.79	2.37	1.97

Table 3-2 COP as a function of ambient temperature

3.8 CO_2 in heat pumps

Heat pumps using R744 as working fluid can achieve high cycle performance if the individual components as well as the systems on the hot and cold side are designed to utilize the unique qualities of CO_2. The low critical temperature of CO_2 means that the operation is likely to be transcritical. The fluid is cooled rather than condensed which means that heat rejection occurs at gliding temperature. Pressure and temperature are independent quantities in this situation. It is important for a CO_2 cycle to reject useful heat over a big temperature range which will result in a large enthalpy difference across the gas cooler, letting the CO_2 outlet temperature become relatively low in order to achieve high values of COP. This means that the *inlet* temperature of the secondary fluid, which is to be heated, should be low. This is contrary to a conventional heat pump where heat rejection (condensation) is at relatively constant temperature and the COP is to a large extent determined by set point- or outlet temperature of the secondary fluid [13]

3.9 Optimum gas cooler pressure

For a spesific evaporation temperature and outlet temperature of the gas cooler there will exist an optimum gas cooler pressure that will result in the highest COP. The specific heat capacity c_p is heavily dependent on the pressure and temperature.

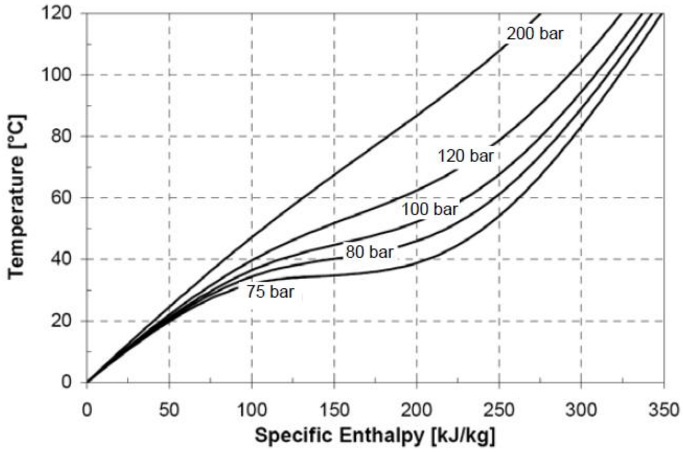

Figure 3-9 Isobars for CO_2 [13]

At supercritical pressure and temperature near the critical point the isobars are almost horizontal meaning a very high c_p and a relatively constant temperature during heat rejection. At supercritical pressure and a temperature below critical or above 50-60 ℃ the isobars are relatively steep, meaning the CO_2 temperature decreases rapidly under heat rejection. The higher the pressure the more constant the c_p during heat rejection causing more rectilinear isobars, as can be witnessed in Figure 3-9. The minimum temperature difference in the gas cooler in counter flow between CO_2 and a cold stream occurs at what is called the *pinch point*. If the pinch point is inside the gas cooler it constraints the heat transfer process and affects the heating capacity and outlet temperature of the CO_2. This is shown in the example in Figure 3-10 and Figure 3-11 taken from [13]. For this example water is heated from 25 ℃ to 50 ℃ and the gas cooler pressure is 75 bar. CO_2 is cooled from 78℃ to 33℃. The broken blue line representing the T-Q curve for the water is practically linear since c_p for water at these temperatures is almost constant. The curve of the CO_2 has a more sway-backed shape because of the varying c_p for CO_2 at these conditions. The mass flow rate of CO_2 is constant while the mass flow rate of water is set to meet the fixed outlet temperature. An increase in water flow rate to reduce the CO_2 outlet temperature will make the outlet temperature of water drop below set point which is usually undesirable. By increasing the gas cooler pressure the CO_2 curve will become straighter and the water flow rate can be increased until the T-Q curve becomes parallel to the CO_2 isobar. The increase in water flow will in turn cool down the CO_2 and consequently create a larger heating capacity for the heat pump.

Increasing the gas cooler pressure also means more compressor work hence there is an optimum gas cooler pressure. Theoretically one should increase the gas cooler pressure as long as the marginal gain in heating capacity is bigger than the marginal increase in compressor work. Figure 3-11 show the results by going up to 85 and 100 bar gas cooler pressure.

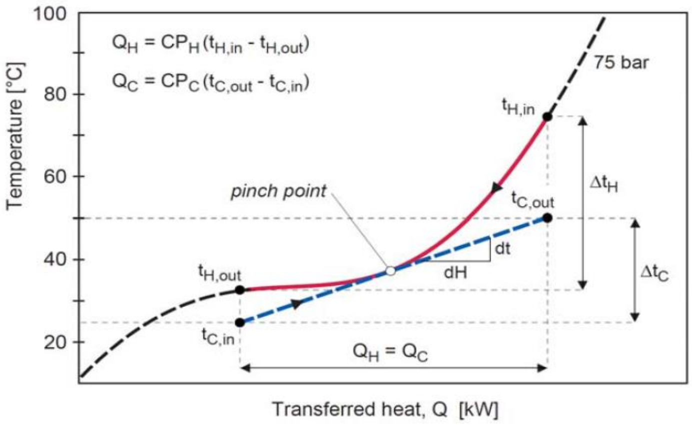

Figure 3-10 Pinch point occurs inside gas cooler [13]

High energy efficiency in a transcritical heat pump cycle is achieved by good temperature fit between CO_2 and water over a large range, excellent heat transfer properties of CO_2 and relatively high isentropic compressor efficiency. This makes production of domestic hot water (DHW) one of the most promising applications for CO_2 heat pumps. With CO_2 one is also capable of supplying high temperature hot water up to 95 ℃ eliminating the need for supplementary heating.

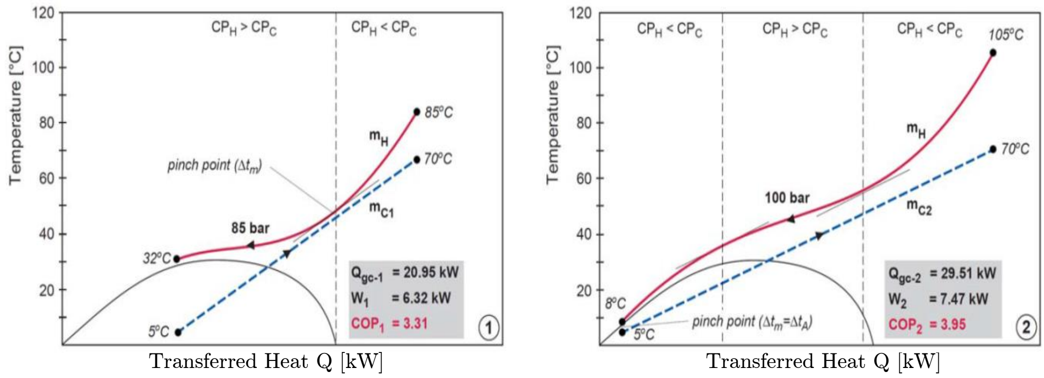

Figure 3-11 Effect of increased gas cooler pressure [13]

3.10 Gas cooler design

CO_2 systems used for space heating (SH) *only* will have a lower performance than systems using conventional working fluids because the outlet temperature of the CO_2 is limited by the return temperature of the SH system (which is usually high) and poor temperature fit. The efficiency can however be quite good if the system combines space heating with hot water heating. The key thing for CO_2 is to reject the heat over a large temperature range. Installing gas coolers at different temperature levels in series can accomplish this. A CO_2 system (Figure 3-12) with a tripartite gas cooler used for preheating of DHW (Gas cooler A), space heating (gas cooler B) and finally reheating of DHW (gas cooler C) is regarded to be the most energy efficient gas cooler design for the combination of SH and DHW. [13]

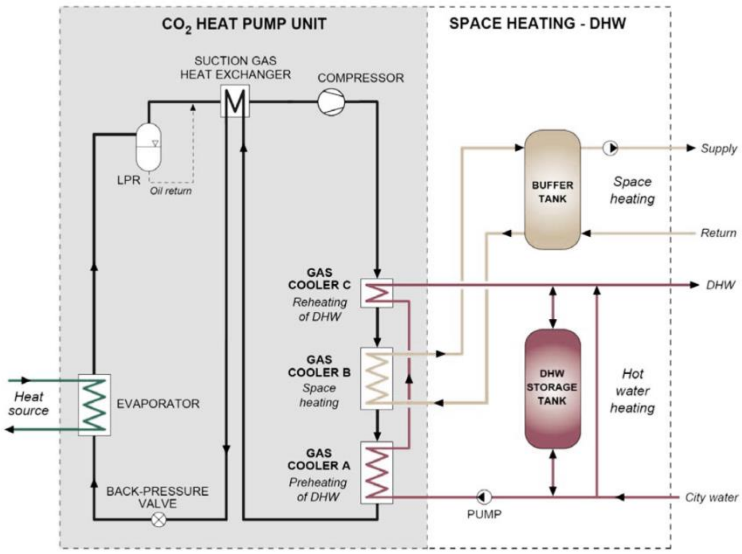

Figure 3-12 Tripartite gas cooler layout for SH and DHW [13]

The system can either operate in SH mode, DHW mode or in a combined mode. A and B is connected to the DHW storage tank. It is important to prevent mixing of the hot water in the top of the tank with the cold in the bottom since this will both increase the CO_2 outlet temperature and waste hot water. B is connected to a buffer tank and a low temperature hydronic heat distribution system for floor heating, convectors or fan coils. The buffer tank is primarily there as thermal energy storage to ensure steady operation, eliminating starts and stops for the compressor at low SH demands. A state of the art conventional heat pump will perform at higher efficiency if the DHW heating demand is under 40-45% of the total heating demand because of the poor temperature fit in SH mode for CO_2 systems. Since the CO_2 system is more suited for DHW there exist a break-even point at a DHW capacity ratio of 45-55%. Above this point CO_2 systems are the most efficient. For buildings such as low energy and passive houses where the DHW heating demand is large (50-80%) the CO_2 system will therefore be most suitable. In these cases the combined mode gives a very good temperature fit as illustrated in Figure 3-13.

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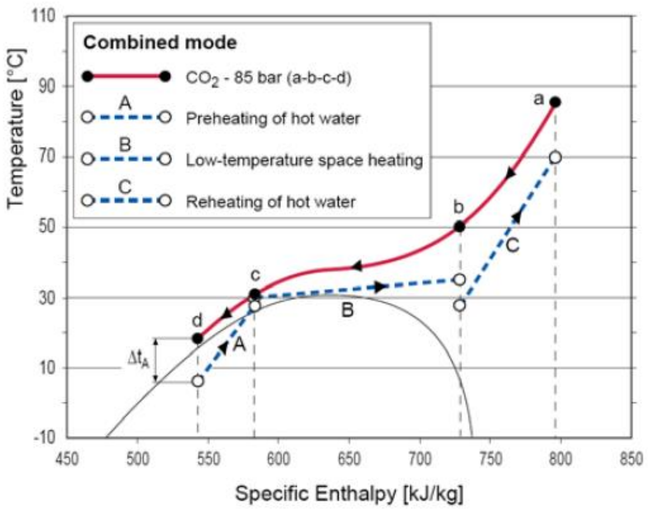

Figure 3-13 T-H diagram for tripartite gas cooler [13]

3.11 Integrated systems in supermarkets with natural refrigerants

Integrating a refrigeration plant with the HVAC system in a supermarket is a widely demonstrated solution that can reduce the energy consumption significantly. In the case where liquid chillers and heat pumps provide both heating energy and space cooling there is a good possibility of integrating vapour compression cycles at different temperature levels reclaiming waste heat from the refrigeration plants. There are typically three different temperature levels operating in a supermarket; low temperature for freezing of food, medium temperature for cold storage and high temperature for space heating and there are multiple opportunities to integrate all these levels. [2]

3.12 System configurations in heat recovery

The conventional solution after the phase out of ozone depleting refrigerants is a R404A system with two separate circuits. Direct expansion at the low temperature unit and brine/R404A at medium temperature level is a common set up. A heat exchanger connecting the medium and low stages is used to further sub cool the liquid coming out from the low temperature condenser. The regulations on HFC charges have become stricter as an attempt to eliminate R404A in refrigeration systems. Indirect systems using CO_2 in the low temperature stage is a natural choice. To recover the heat from the high stage condenser there are a number of solutions, but the most common include heating of space air for the ventilation system in some way. Some basic principles are mentioned under.

- Fixed head pressure: the heat is reclaimed from the condenser at a pressure required to provide the proper temperature for the heating system.
- De-superheater: before the discharge gas from the high stage compressor is condensed it rejects heat to a de-superheater that can supply the HVAC or a hydronic system. The pressure is controlled by a regulating valve according to the heating demand from the de-superheater.
- Heat pump cascade: instead of using the heat from the condenser directly it can be used as a heat source for a heat pump that in turn delivers the required effect to the HVAC. Condensing pressure for the refrigeration system can then be lowered.
- Heat pump cascade for sub-cooling: the sub-cooler is installed after the condenser supplying heat to a heat pump while improving the efficiency of the refrigeration system. The refrigeration system can operate at low condensing pressure when ambient temperature is low and heating is needed. [11]

As previously shown CO_2 can be a refrigerant suited for the purpose of both refrigeration and heat pumping, it is in other words a very interesting agent for integrated systems in supermarkets. With state of the art equipment, optimum gas-cooler pressure control and a hydronic heat system a R744 plant in a cold climate could prove both very effective and desirable from an ecological point of view. Still, it is hard to determine whether a solution is better or worse compared to the conventional method without actually installing and testing the systems. Trial and error is an expensive approach and the last decades have seen great improvements in predictive models and simulation tools for supermarket systems which include heat recovery.

3.13 Simulation tools

A supermarket is thermodynamically a very complicated building. There are multiple interacting subsystems and temperatures levels that range far away from the desired indoor temperature or the ambient temperature. This means that it can be very hard to predict how each sub system will react in the presence of the others and the way they should be controlled. The purpose of any simulation tool is to get a better understanding of the actual system simulated and to assist in making a decision that can improve the system. This assumes that the simulation model is accurate enough so that in fact the right decision is made. The model should be detailed and reasonably accurate in terms of what is actually happening in the system. It is better to be approximately right than exactly wrong. Being able to vary parameters and see how they influence a complete system is easier, faster and cheaper in a computer program compared to on site testing and measurements. When one tries to simulate heat behaviour in a supermarket the end goal is to reduce the energy consumption. This is preferable both from an environmental point of view as well as a financial. The cost of energy compared to the total turnover for a supermarket is in the same order as the profits [1]. If the cost of energy and profits represents 1 and 3% of the turnover respectively, then a reduction of 50 % in energy cost would increase profits by 15%. Methods like the Eco-Efficiency and TEWI can be included as a way of measuring performance between solutions.

3.14 Testing of system solutions using simulation models

In the article in Applied Thermal Engineering 48 by Luca Cecchinato, Marco Corradi and Silvia Minetto [2] a set of different system solutions with natural refrigerants were simulated using Energy Plus which is a building simulation tool with advanced capabilities for supermarkets as well as other types of buildings. It is developed by the U.S department of Energy. Each plant option was simulated in the climate for Treviso (northern Italy), Stockholm and Singapore. R717 (ammonia) and R290 (propane) were chosen as refrigerants for the chillers and heat pumps while

MT units used R717, R290 or R744 (CO_2). R744 is used for the LT units. When R744 was chosen in both MT and LT units the booster configuration was assumed.

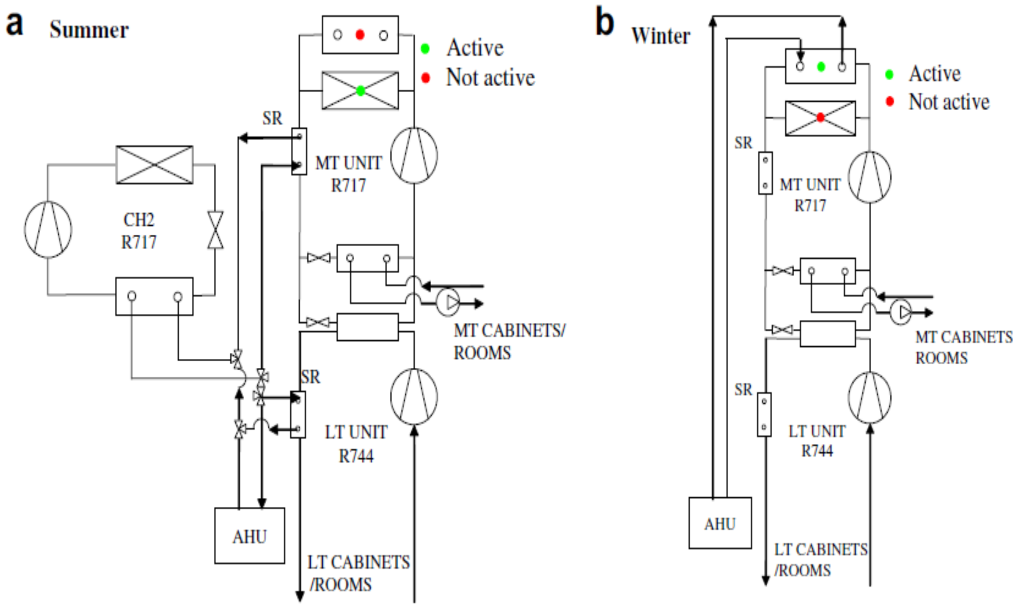

Figure 3-14 Plant option A1 [2]

Plant option A1 during summer has a configuration where Ammonia is used as a chiller to feed the cooling coil in the air handling unit (AHU) and to condense the MT unit which is also with ammonia. R744 is used as the heat transfer fluid for the cabinets. The LT unit uses R744. During winter the heat from the condenser is directly reclaimed to the AHU. The heat could alternatively be used as a heat source for the chiller that would work as a heat pump and deliver heat to the AHU. This option was called A2. Another configuration was the booster layout with R744 in both the MT and LT unit. During summer an ammonia chiller condenses the refrigeration unit and during winter the heat is reclaimed via a water loop to the AHU or alternatively to the R717 unit which then works as a heat pump (A5).

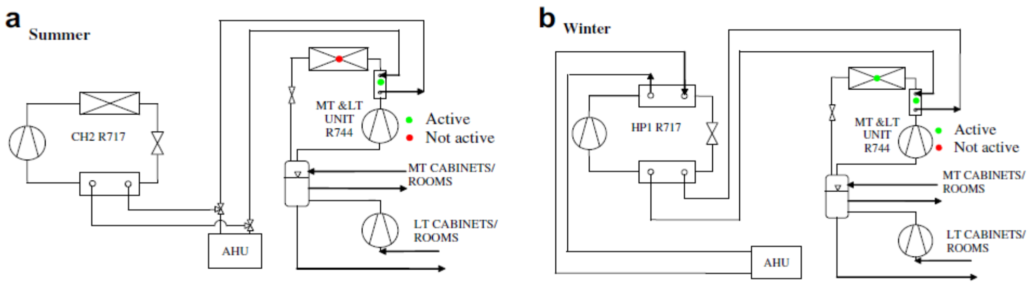

Figure 3-15 Plant option A5 [2]

The same configurations were used with R290 replacing R717 which was called plant configurations P1-P5. An advantage with the R290 solutions is that the chiller can be reversed to work as a heat pump by the switch of a four way valve. R717 units with a flooded evaporator can't be reversed and need therefore more complicated piping and changes when used as a heat pump also. P1 is in any other way a typical cascade layout analogous to A1.

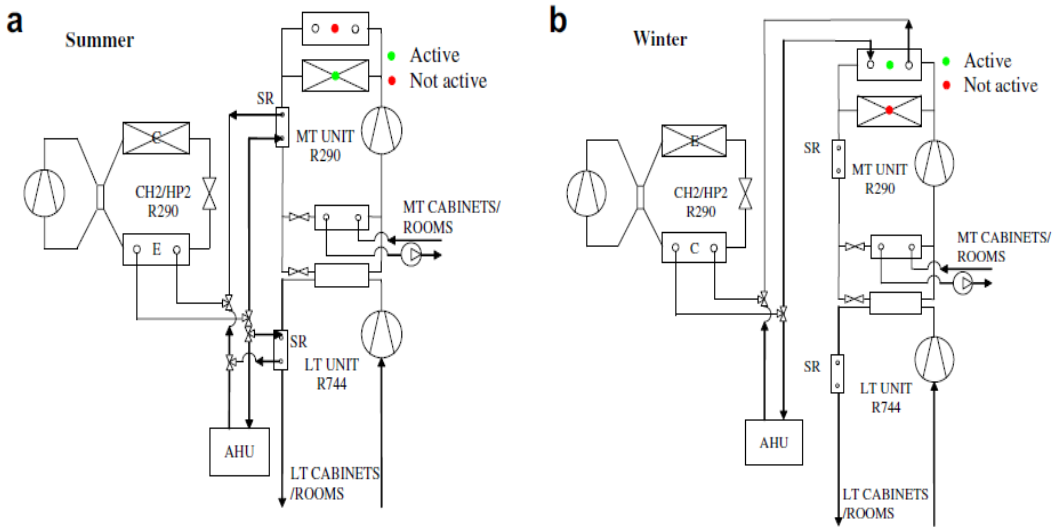

Figure 3-16 Plant option P1 [2]

From the simulations in the study the plant options A1 and P1 performed the best in all climates in terms of annual energy consumption. They were also considered to have the simplest layout with lower complexity in the piping integration system. It should be pointed out that the economical aspect beyond the electrical consumption was not considered in this study. Investment costs, space and safety requirements and the ecological aspects are as earlier mentioned important factors that should be considered. Still, this is a good way to investigate the differences and weigh pros and cons of different system solutions.

Models can also be used to optimize the control strategy of a specific plant once a plant configuration has been selected. The operation will have to change and adapt during the year due to changes in ambient conditions if the highest performance is to be achieved. If the system is tested under varying conditions and different scenarios in a simulation program it is more likely that one is prepared to engage the right control in the real case. If the results are to be reliable then the input data representing the variable conditions should be as realistic as possible. It is also important to be aware of all simplifications and assumptions that have been made so that the results can be evaluated and correctly interpreted.

4 CASE STUDY

REMA 1000 is a Norwegian grocery store chain that belongs to "Reitangruppen" which is a private family owned company. The REMA 1000 stores are based on the franchise concept andis the biggest chain in Norway based on annual turnover with a maket share of 21.3% [14]. A new REMA 1000 supermarket is currently under construction in Trondheim. It is a pilot project in collaboration with SINTEF and their CREATIV initiation which is a platform for research and development with focus on energy efficiency in industry. SINTEF scientists have been involved with every part of the design of the supermarket to make it as environmentally friendly and efficient as possible while still accommodate the needs of a sales market. Smart heat recovery is an important part of the solution. This section will give an explanation of how this is accomplished. The supermarket is refered to as "Kroppanmarka".

4.1 Kroppanmarka system

Figure 4-1 shows a principle sketch of the highly integrated system at Kroppanmarka. The goal of this solution is to reduce the energy consumption of the store to a minimum by reclaiming the heat from the refrigeration system. The need for electrical heating is hopefully eliminated with the right control strategy of the plant. This is investigated later in the paper. A challenge with most supermarkets is that there is a conflict of interest between the merchant, the franchise and the owner of the building. The ventilation system is the responsibility of the owner and as long as rules and regulations are followed he will most likely buy the cheapest one he can get. The franchise on the other hand is responsible for the refrigeration plant, and investment costs are also a decisive factor here. As a result, integration between the two systems may prove difficult due to the fact that they were never meant for each other in the first place. Finally the merchant is left with the electrical bill and a relatively inefficient system which also have a bad effect on the environment. With this new project this challenge will not be an issue as the parts of the system have been carefully selected together. The system consist of 5 main parts.

- CO_2 Refrigeration Plant
- Hydronic Floor Heat System
- Heat Accumulation Tanks
- Energy Wells
- HVAC

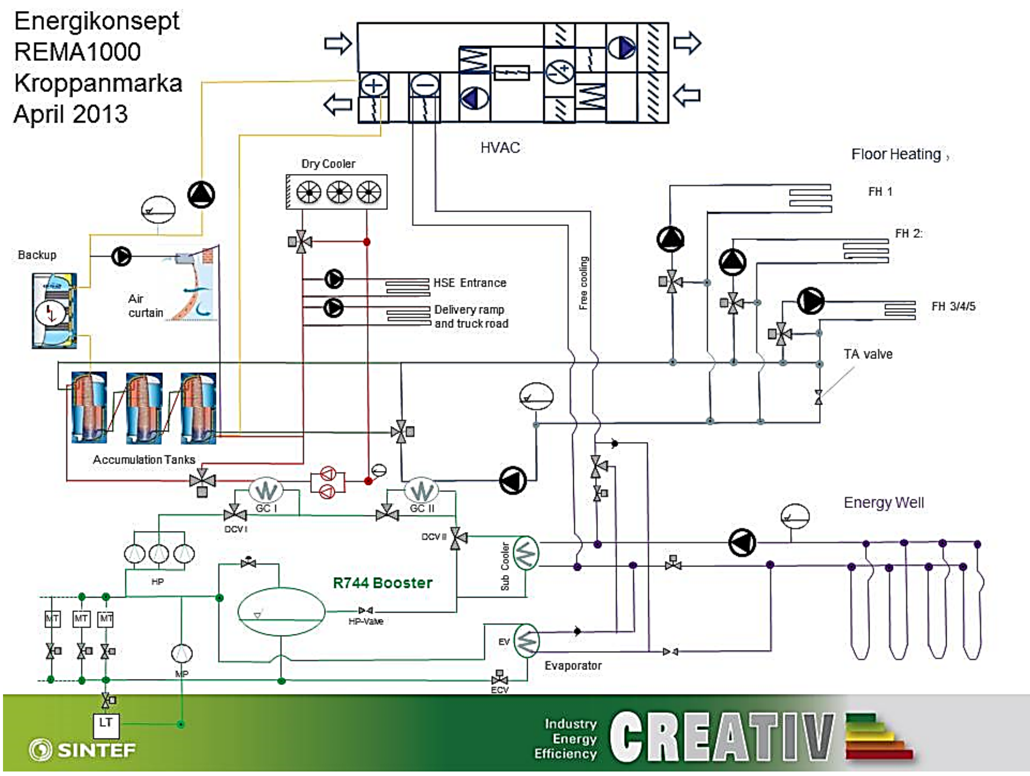

Figure 4-1 System principle sketch of the SINTEF solution for Kroppanmarka

4.2 Refrigeration plant

The base of the system is the R744 refrigeration plant. It's a transcritical booster system with double compression and double expansion which assures evaporation at two levels. Its main purpose is to deliver low and medium temperature to the cabinets and cold rooms at -8 and -35 C°. The heat generated in the refrigeration process is reclaimed for space heating and so the operation of the plant is largely dependent on the heat demand of the building as well. A Pressure-enthalpy diagram is shown in Figure 4-2. Heat is rejected in step 2-3 to one or in a combination of three gas coolers.

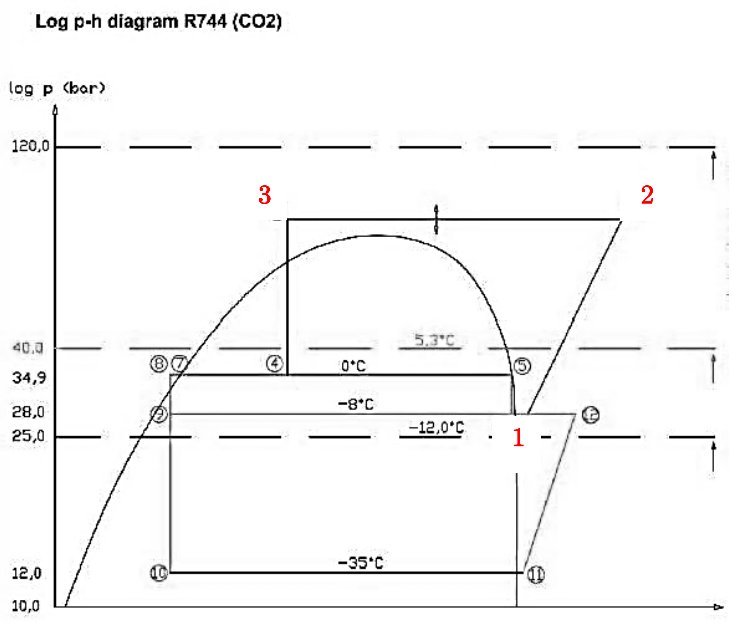

Figure 4-2: The CO_2 Refrigeration cycle displayed in a log p-h diagram

Gas cooler 1 (GCI) exchanges heat with the accumulation tanks, gas cooler 2 (GCII) with the floor heating circuit and a sub-cooler (GCIII) connected to the energy well makes sure the temperature of the refrigerant is low before throttling. A forth heat exchanger works as an extra evaporator which increases the mass flow rate allowing more heat to be reclaimed at the cost of additional compressor work. Another way to increase the heat available for reclaim is to raise the pressure ratio (step 1-2) of the high pressure stage. The operation of the plant can be both sub- and transcritical depending on the heat demand which means condensation or gas cooling take place in the heat exchangers. Regardless of the operation, they will be referred to as gas coolers for the remainder of this paper. The different strategies are investigated in chapter 8.

4.3 Floor Heating

Water is heated in GCII and will be circulating in tubes under the concrete floor of the store. If there is an additional heat demand to the floor more heat can be extracted through a spiral heat exchanger in the accumulation tanks. There are four different *loops* in the store with individual control to maximize efficiency and comfort. There are six *zones* defined in the supermarket.

- Zone 1: is dedicated to the entrance and checkout. This is the zone with the highest priority and temperature. Staff will occupy this area for long periods of time which requires an increased need to provide comfort.
- Zone 2: is the area of fruit and vegetables which require lower temperatures.
- Zone 3: is not in the sales area, but where staff can reside.
- Zone 4: is the biggest sales zone and is therefore the second priority zone.
- Zone 5: is the main refrigeration zone. Cabinets, plugins and freezers are located in this zone which means only a minimum of heating is used.
- Zone 6: is outside the entrance. During winter this will make sure snow and ice doesn't cover the area at the entrance.

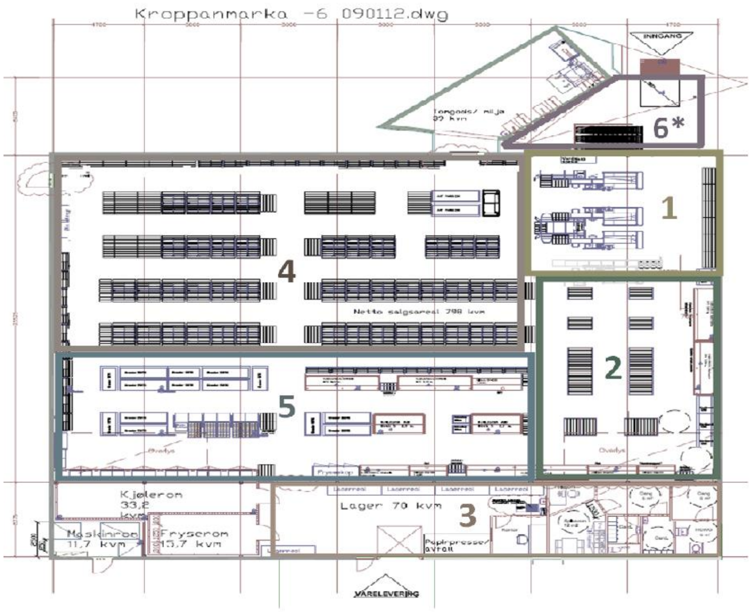

Figure 4-3 Heating zones for the hydronic floor heat system

Using radiant floor heating has some advantages compared to conventional convective systems. It uses relatively low water temperatures which are favourable for heat pumps regarding efficiency. It acts as a mass storage and is able to hold a stable temperature and for longer periods. It is also more preferable in terms of comfort compared to convective systems. The biggest challenge with the concrete floor heating is the consequences of thermal inertia. Because of the huge mass it is has a slow response which make the indoor temperature more difficult to control. [15]

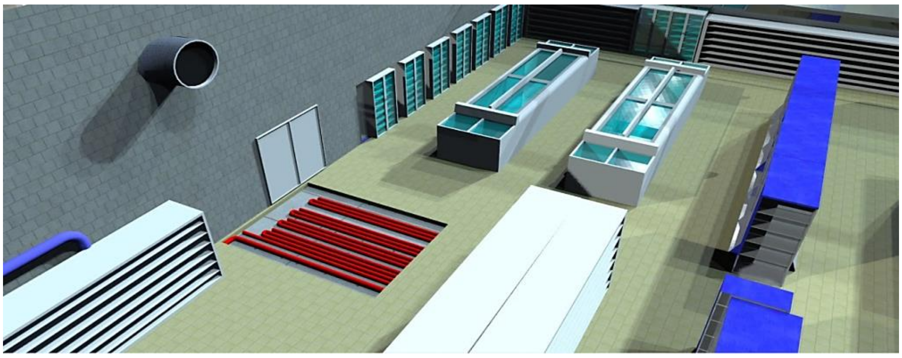

Figure 4-4: Cables with hot water is circulating under the concrete floor

4.4 Heat accumulation tanks

It is important to remember that reclaiming heat only makes sense when there is a need for this heat somewhere. Rejecting heat to the floor when it is not needed will only increase the refrigeration duty on the cabinets and cause an unnecessary cooling demand in the building. In other situations one may have a heat deficit. At Kroppanmarka this problem is solved with heat accumulation tanks. Instead of rejecting the heat to the energy well or dry coolers when the building has a low heat demand it can be stored in the tanks and utilized later when there is a heat deficit. This can also allow for a more steady operation of the compressors, reducing the number of start and stops. [13]

There are three tanks with the combined capacity of 600 litres planned for the system. The tanks are filled with a water-glycol mix that works as thermal storage for the heat transferred through GCI. This heat can then be utilized at a number units connected. The main purpose is to serve a heating coil in the ventilation system, but it can also be a heat source for an air curtain at the entrance, the ground at the HSE entrance or the delivery/service ramp. By switching a three-way valve the floor heat loop can exchange heat with the spiral heat exchanger in the tanks. There is also an electric heater connected to an additional tank as a backup for peak loads or in case the refrigeration heat recovery system is down.

4.5 Energy well

Four wells are drilled 170 meter deep. Because of uncompact material in the ground 50 meters of steel protection casing around the pipes are used. The gap between the collector tubes and bedrock wall is filled with water. Glycol will circulate in the tubes to prevent freezing. When there is no demand to the floor and the accumulation tanks are full, which is a classic hot summer day scenario, the CO_2 must be cooled some other way. The energy well provides free cooling through the sub-cooler (GCIII) at any time ensuring a low CO_2 temperature before throttling. Another circuit is connected to a cooling battery in the HVAC for free indoor cooling. During winter and other times with high heat demand the energy well can work as an extra evaporator for heat pump operation. The extra heat is transferred to the same circuit as the MT cabinets. Heat is extracted from the well which slowly cools the surrounding bedrock. For heat pump systems using bedrock as heat source the temperature of the rock can decrease to a lower equilibrium temperature and which reduces the output effect. There should be a balance between how much heat is extracted *from* the well and how much is rejected *to* the well. Thermal charging, in which the ground is heated by another source, for instance solar heat exchangers or waste ventilation air, can accomplish this [16]. Since the energy well at Kroppanmarka is also used for cooling, meaning heat is rejected to the well, it doesn't need an additional thermal charging source [16].

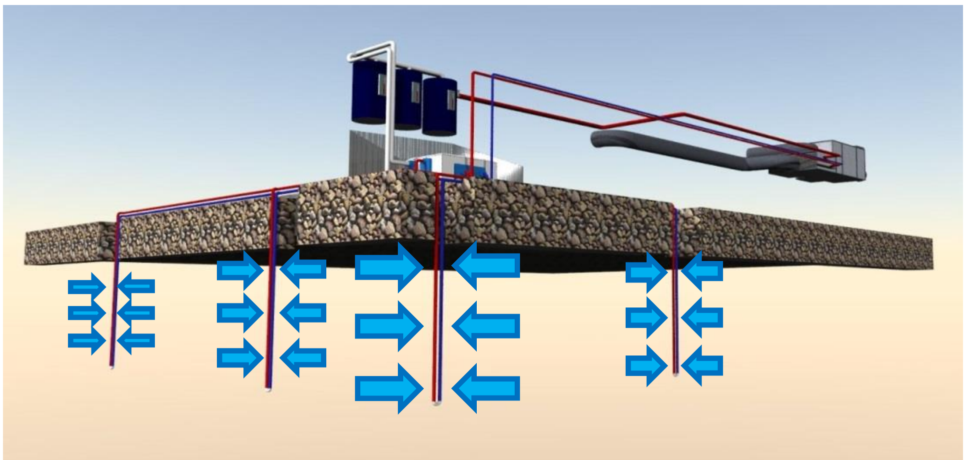

Figure 4-5 The model is displaying the energy wells providing free cooling

4.6 HVAC

There are many different modes of operation for the HVAC. This is related to how the air passing through is handeled. The modes depend on a number of things, but there are 5 key elements that control the operation.

- Outside temperature, $T_{ambient}$
- Indoor temperature, T_{shop}
- CO2 concentration, ppm CO_2
- Relative humidity, RH
- Heat available in accumulation tanks, $Q_{available}$

Whenever heating of recirculated or fresh air is needed, the heat from the accumulation tanks will be used. If the air needs to be cooled, free cooling from energy well is provided. A heat recovery rotary wheel can exchange heat between fresh and waste air. This will only be used if the tanks can't provide the required heat because of a high pressure drop over the rotary wheel. If the relative humidity of recirculated air is above 60%, it needs to be cooled and then heated to remove the water to reduce defrosting of cabinet evaorators. This is a specific measure implemented to reduce energy consumption. There are many combinations of situations and all the different scenarios will have the HVAC operating in a certain mode.

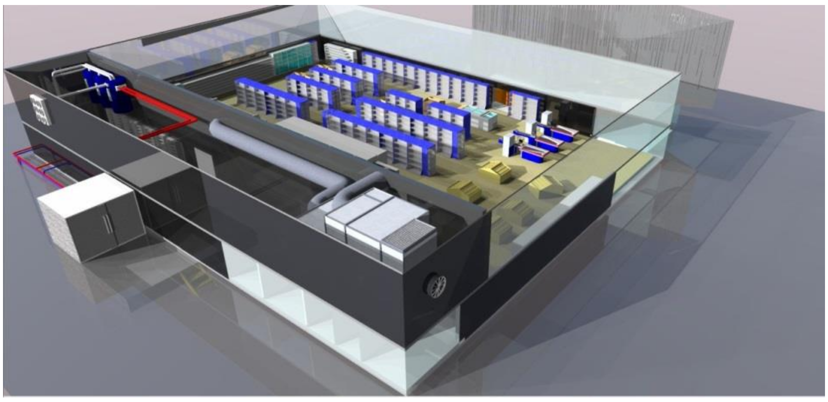

Figure 4-6 The 3D model showing the HVAC unit in the store

Each mode represents a certain flow path of the air through the HVAC unit so that flow path 1 will refer to mode 1 and path 2 with mode 2, and so on. At this stage there are 16 different modes. Some modes will probably dominate the operation and this is part of what the simulation model hopefully will reveal once it is ready. Depending on which path the air from both outside and inside of the shop takes, there will be a corresponding pressure drop which in turn leads to a certain power requirement from the fans.

Figure 4-7 shows the different components in the HVAC that the air will pass through. A description of each of the 16 flow paths can be found in Appendix B.

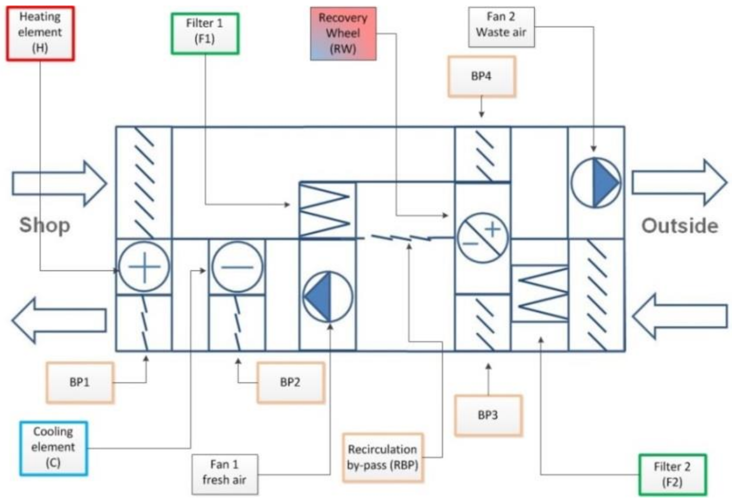

Figure 4-7 Components of the HVAC

Each path is selected so that for a particular situation there is an energy efficient way to provide comfortable air to the store at the right temperature. The control of this operation can be very complex and will rely on signals from a number of measuring devices that report on temperature, humidity, CO_2 concentration levels and more, at different locations.

4.7 Ambient conditions

The supermarket is designed by Snøhetta and has a modern and nice architectonic exterior. As part of the design a large amount of the surface area is made of glass which allows natural light to brighten up the store during day.

Figure 4-8 Model of Kroppanmarka showing the large surface area of fenestration [17]

Windows are not as well insulated as the walls and there is also a significant heat transfer through radiation occurring. A simplified basic equation for the energy flow through fenestration is

$$Q = UA_{pf}(t_{out} - t_{in}) + (SGHC)A_{pf}E_t \tag{2}$$

Where

Q = instantaneous energy flow, [W]
U = overall coefficient of heat transfer, [W/(m^2·K)]
t_{in} = interior air temperature, [°C]
t_{out} = exterior air temperature,[°C]]
A_{pf} = total projected area of fenestration, [m^2]
SHGC = solar heat gain coefficient, [-]
E_t = incident total irradiance, [W/m^2]
Q can be looked upon as consisting of two parts

$$Q = Q_{th} + Q_{sol} \tag{3}$$

Q_{th} is the energy flow through the window caused by the temperature difference between the inside and outside of the window. Q_{sol} is the energy flow through the window caused by solar radiation. It is common to distinguish between three types of radiation [18]:

- Direct
- Diffuse
- Reflected

The direct radiation is the radiation going undisturbed through the atmosphere in a straight line, while the diffuse radiation is the radiation that is scattered by particles in the atmosphere. The last type is the reflected radiation coming off the surroundings and the ground which also depend on the albedo reflection factor. This quantity is much smaller than the other two and is sometimes included in diffuse radiation [18]. The type of radiation that dominates between direct and diffuse will depend on the weather conditions and the position of the sun. On a cloudy day almost all radiation will be diffuse while a clear-sky-day will be dominated by direct radiation (85%). Because there are a lot more particles in the atmosphere apart from the ones in clouds the amount of diffuse radiation can be substantial on a clear day too. When the sun is just 10° above the horizon the diffuse radiation can be as high as 40% on a sunny day [19]. A vertical surface will receive more radiation when the sun-angle is low while the opposite will be true for a horizontal surface. All these variables will play an important part in estimating the cooling and heating demand for a building. Since the supermarket has a large surface area of glass the weather conditions may play an even bigger role at Kroppanmarka than is normally expected.

5 SCENARIOS FOR THE VENTILATION MODEL

5.1 Purpose

The main goal of this thesis is to get annual estimations on energy consumption for different conditions and control strategies. These estimations will be based on results from the simulation models. The model is programmed to do day simulations. This means that one should in principle do 365 different simulations in order to get a full year. Simulations can however be quite time consuming and unpredictable. In order to reduce the amount of simulations one can let a certain number of scenarios represent a higher number of days. One could for instance choose four scenarios, one for each season. The scenario called "Summer" would represent all days defined as summer and the scenario called "winter" would represent all days of the year defined as winter, and so on.

Figure 5-1 show how the "summer" day-scenario could look like. This data is the input for the ventilation model and a simulation can be performed. The results will show the behaviour of the ventilation system for this day and the associated energy consumption. The same is done for the other scenarios (winter, spring and fall) which make 4 simulations in total. To get an estimation on the annual energy consumption one would then multiply each result with the number days each scenario is defined to represent. The goal is then achieved with only four simulations instead of 365. The results are naturally less accurate compared to simulating each day, but a rough estimation is produced quite easily.

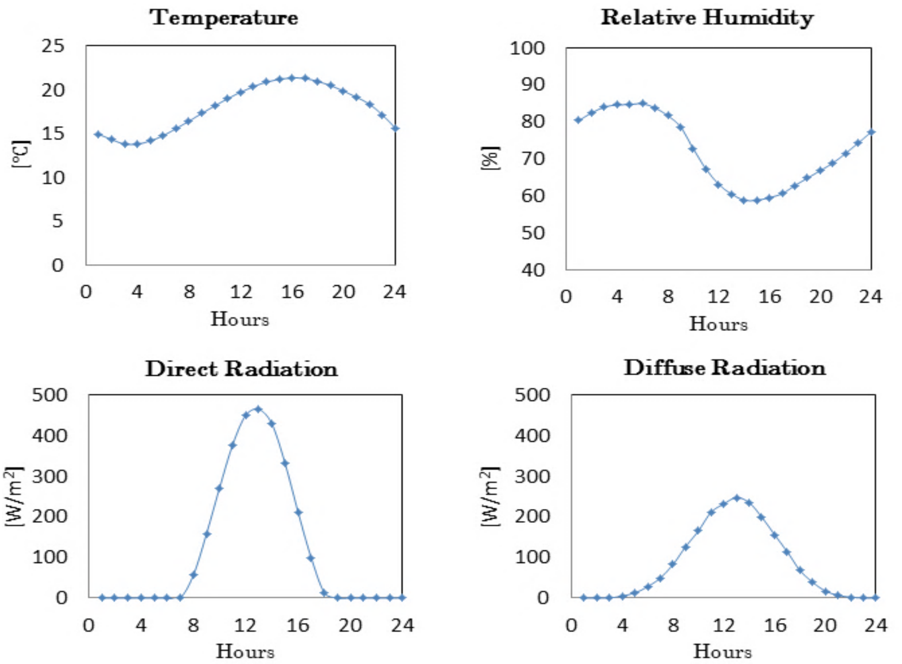

Figure 5-1 Example scenario for a summer day for each of the parameters considered

5.2 Approach

In this work there were four weather condition parameters used for the model

- Temperature
- Relative Humidity
- Direct vertical radiation
- Diffuse vertical radiation

Even though these parameters are related to each other they don't have a constant correlation (a sunny day is not always warm). They therefore need individual scenarios or variations that are independent of each other. The combination of all four parameters with a respective variation makes up a complete scenario and it is these scenarios that are the representatives used as input data for the simulation model.

The scenarios created where based on measurements from a weather station at Værnes downloaded via Meteonorm. This data is in hourly values for each parameter for one year. It is refered to as the *real* data and values, contrary to *scenario* data and values.

The challenge was to make the number of scenarios as small as possible while still maintaining good fit with the real data. To solve the problem a program in excel was created. The basics and the idea behind the program are explained in the following section while the details can be found in The Appendix along with a short description of Meteonorm.

Procedure

The data on temperature, humidity, direct and diffusive vertical radiation over the course of one year in hourly measurements are downloaded to the excel program. The reader should use Figure 5-2 and Figure 5-3 which illustrate the data processing. Each parameter is divided into 5 ranges set by the user. Each range represents a category

- Very Low
- Low
- Medium
- High
- Very High

which defines the magnitude of the parameter. The range for the category "Very Low" in terms of temperature could be $T < -5°C$ and "Low" could be $-5 < T < 5$. A day where the average temperature is -7°C would then fall under the "Very Cold" category, whereas a day averaging at 2°C falls under "Low". The program scans through the data from Meteonorm (8760 values per parameter) and places each day under the category it belongs. From each category a representative scenario is created.

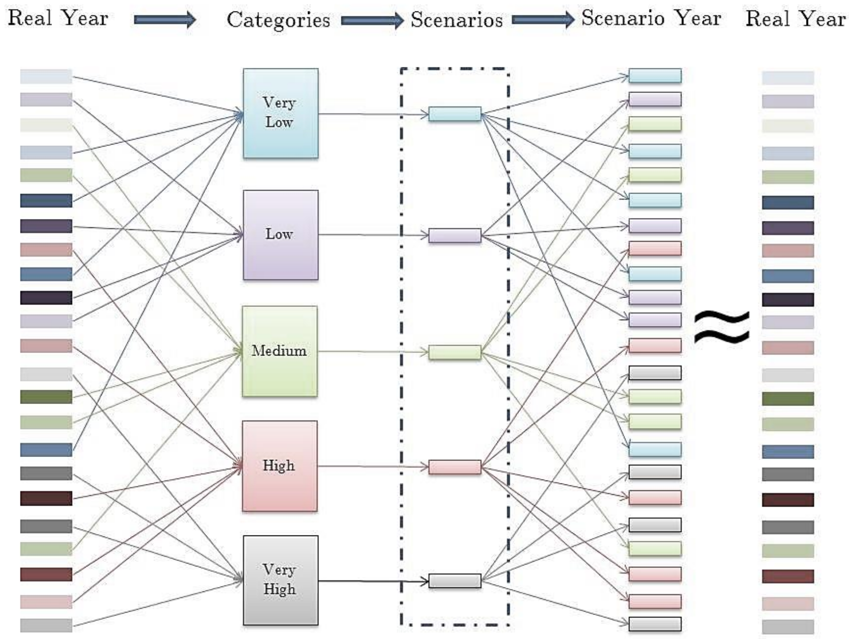

Figure 5-2 Visual representation of the data processing for one parameter. The small blocks represents days.

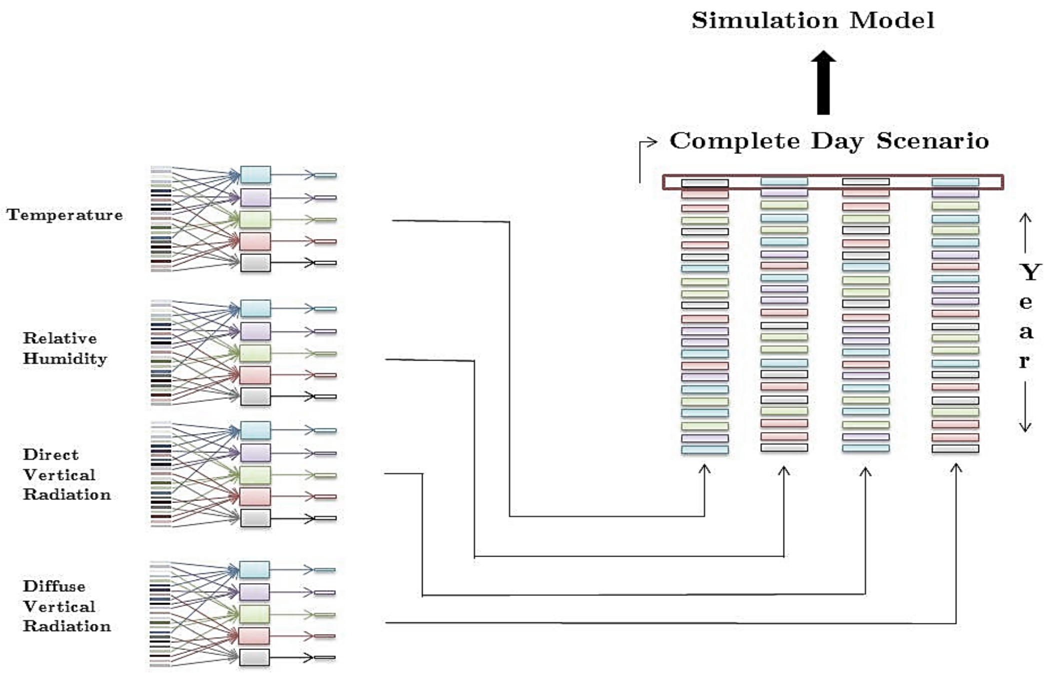

Figure 5-3 Each parameter combination completes a scenario.

5 scenarios are now created for each parameter and have to be placed in a sequence of 365 days that will match the actual data as accurately as possible. A scenario year has then been created for each parameter and it is the combination of these that create a complete day-scenario that will be used in the simulation model (Figure 5-3). The results for each parameter is a graphed with the values from the real year so the results can be visually compared. In addition, the program calculates the relative deviation between the scenario year and the real year. Every single scenario hour is compared to the corresponding real hour so that it is possible say something about the reliability of the scenarios.

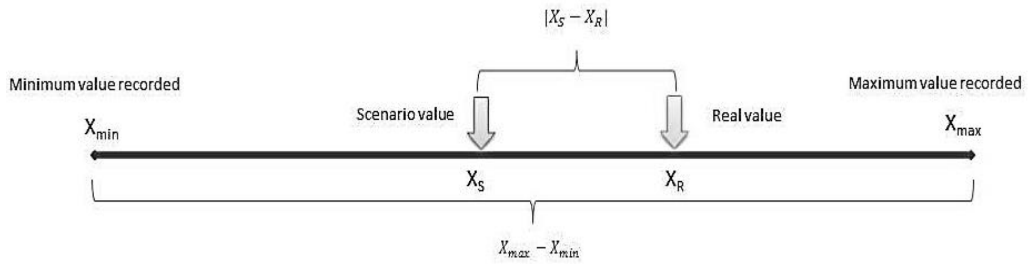

Figure 5-4 Relative deviation

$$Relative\ deviation = \frac{|X_S - X_R|}{X_{max} - X_{min}} \tag{4}$$

Example: If the highest temperature from Meteonorm was 30 ℃ and the lowest -15 ℃ and the temperature at 12.00 on the first of May was 10℃ for the scenario and 5.5℃ for the real value then the relative deviation would be

$$\frac{|10 - 5.5|}{30 - (-15)} = 0.10 = 10\%$$

Every step described take place instantaneously as the ranges for the categories are entered. That way that the graphs and deviation values can be studied while trying different ranges to optimize the scenarios.

Simplifications

Each of the parameters are divided into 5 categories and so they each get 5 possible variations. Since there are 4 parameters (Temperature, Humidity, Direct- and diffuse radiation) this would in principle add up to 5^4=625 scenarios because one temperature can have 5 variations of humidity which again can have 5 variations of direct radiation and so on. For that reason some simplifications had to be made. The first assumption is the use of only one humidity scenario. To justify this one must observe the relative humidity curves for the average day per month in Figure 5-5. Graphs of the monthly averages are produced for all parameters for a better overview and understanding of the input data. It is clear that the curves for humidity in each season are very similar, especially during summer.

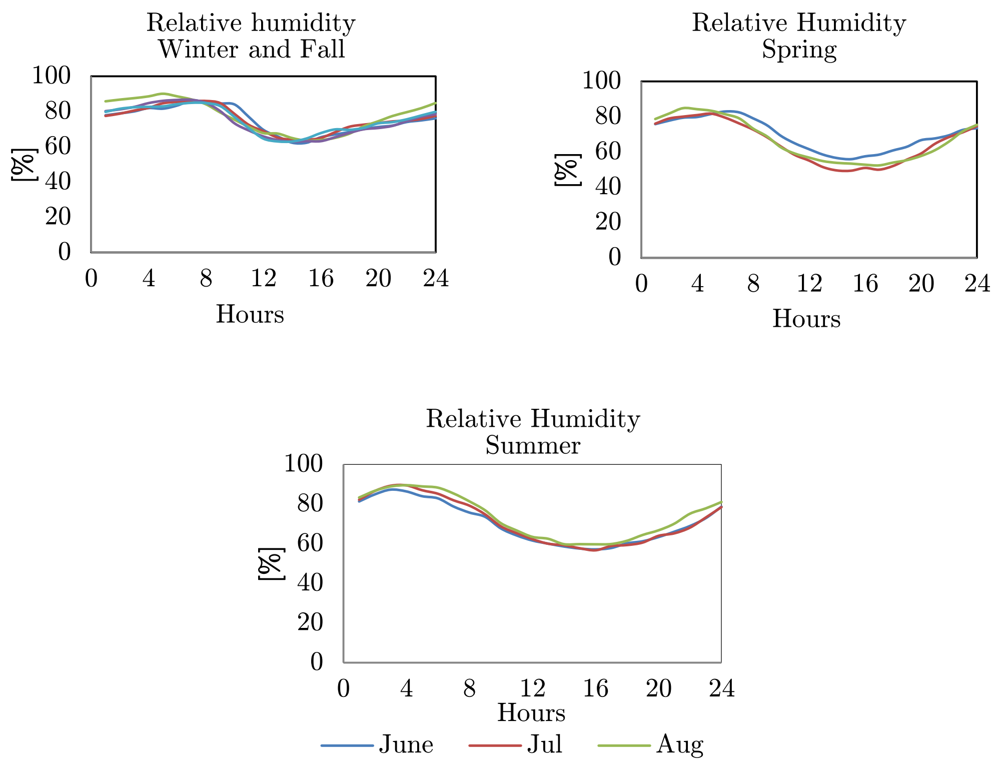

Figure 5-5 Monthly averages of relative humidity during one day

High humidity can result in frost formation on evaporators which has to be removed. Defrosting cost energy. It is during summer that the humidity is most important because this is when the relative humidity outside is closest to the relative humidity inside the supermarket. The average day of July was therefore selected.

With only one case of humidity the number of possible scenarios are reduced to $1 \times 5 \times 5 \times 5 = 125$. This is a third of a whole year, but would still require too many hours of simulation. It was therefore decided to reduce the number of variations of each of the remaining parameters; temperature, direct and diffusive vertical radiation down to only three by combining four of the five original categories into two. This gives a total of $1 \times 3 \times 3 \times 3 = 27$ possible day-scenarios reducing the simulation work with 93%.

5.3 Results

	Input Ranges		
Category	Temperature [℃]	Direct Radiation [W/m^2]	Diffuse Radiation [W/m^2]
Very High	19 and above	500 and above	250 and above
High	11−19	400−500	225−250
Medium	3−11	200−400	150−225
Low	-6 − 3	100−200	75−150
Very Low	-6 and below	100 and below	75 and below

Table 5-1 Ranges for each parameter used to create the scenarios

Table 5-1 shows the input ranges used for the parameters. The scenarios of each parameter are presented in Figure 5-6 to Figure 5-8 while the annual distribution is found in Figure 5-9 to Figure 5-11.

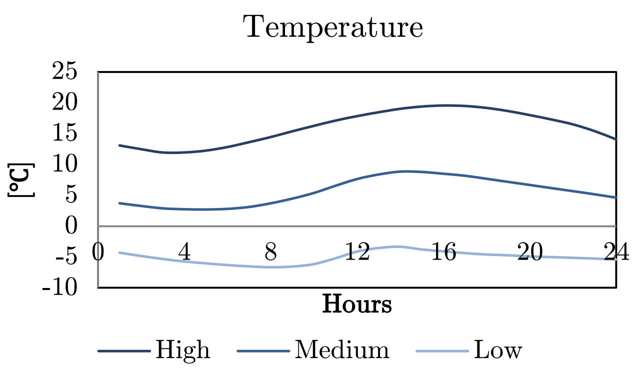

Figure 5-6 Three cases of temperature

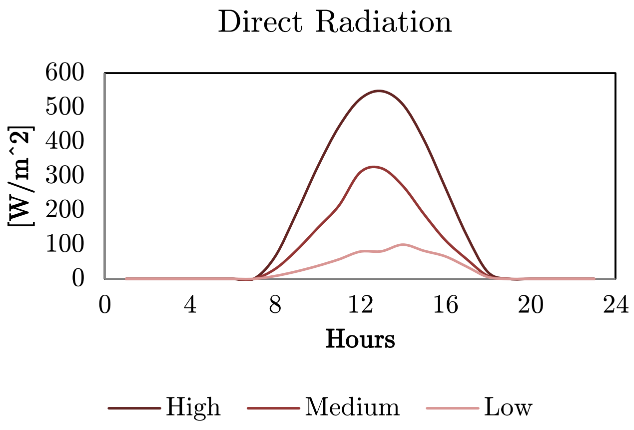

Figure 5-7 Three cases direct vertical radiation

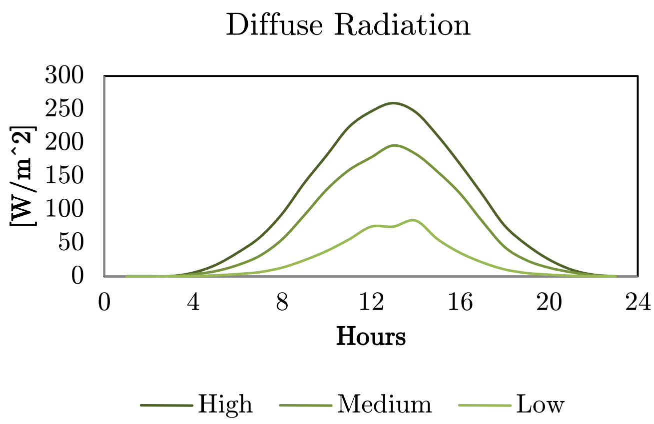

Figure 5-8 Three cases diffuse vertical radiation

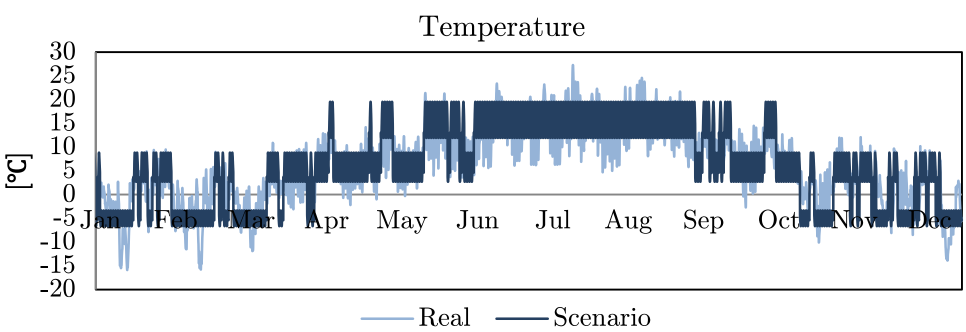

Figure 5-9 Annual distribution of temperature scenarios compared to real values

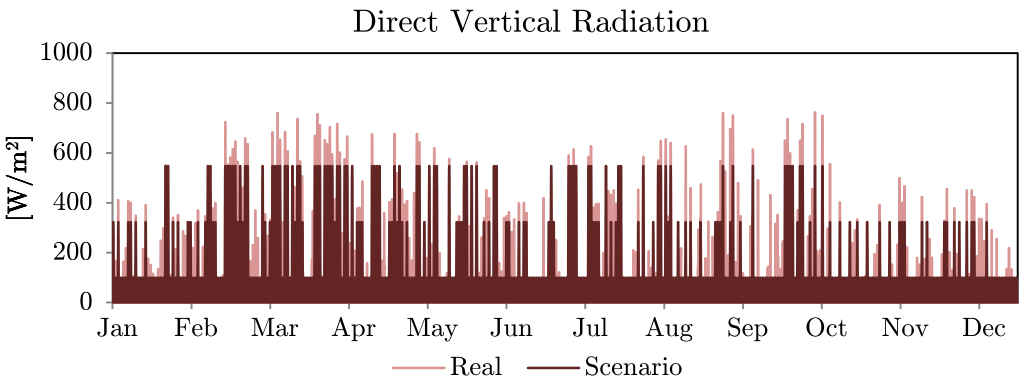

Figure 5-10 Annual distribution of diffuse vertical radiation scenarios compared to real values

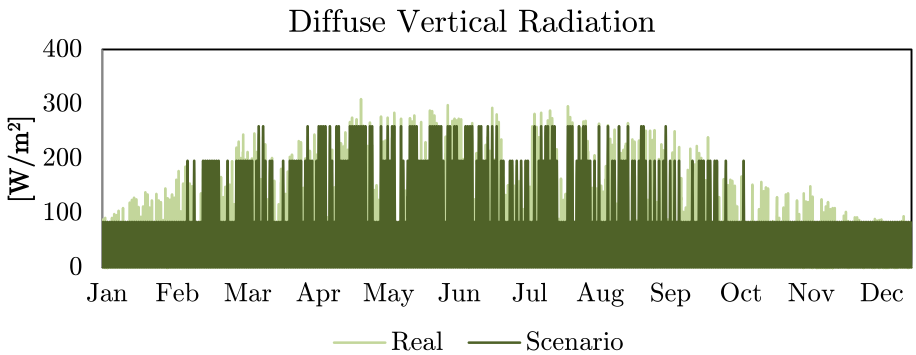

Figure 5-11 Annual distribution of diffuse vertical radiation scenarios compared to real values

The annual relative deviation for each parameter is plottet in Figure 5-12 and the monthly averages are found in Figure 5-13. Finally Figure 5-14 show the best and worst weeks of each parameter.

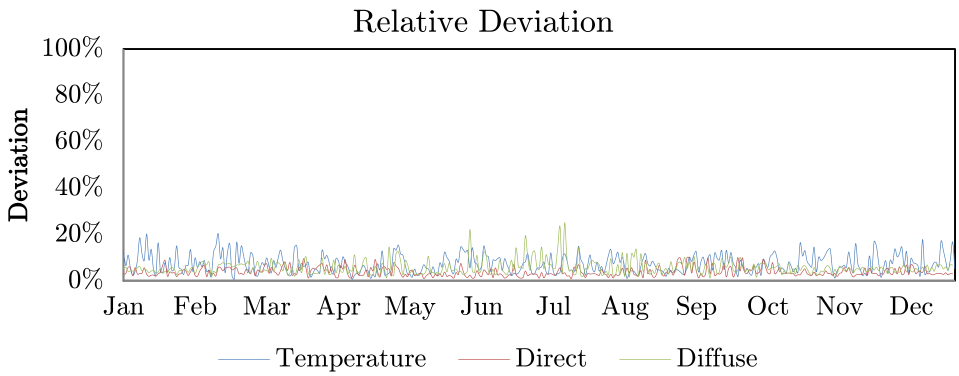

Figure 5-12 Daily average relative deviation for one year

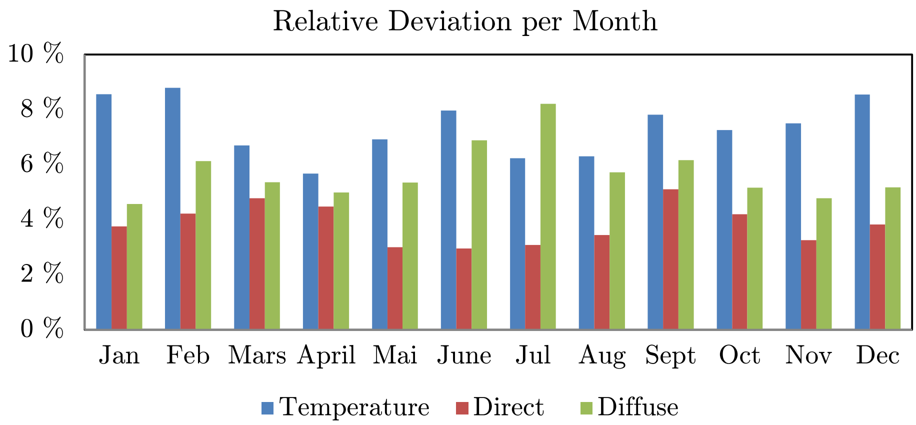

Figure 5-13 Monthly average deviation for the three parameters

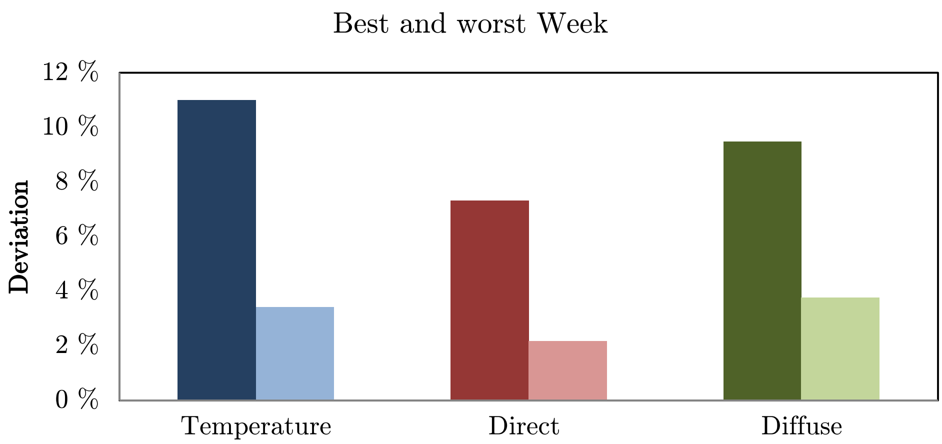

Figure 5-14 Average relative deviation for the best and worst week of each parameter.

The annual average deviation for temperature, direct and diffuse radiation was 7.35%, 3.84% and 5.71% respectively.

It is interesting to see how reducing the number of scenarios affect the results. Data using 5 variations of each parameter was therefore also created. The results for temperature, direct and diffuse radiation is presented in Figure 5-15, Figure 5-16 and Figure 5-17. The average annual relative deviation is compared to the case of 27 scenarios in Figure 5-18.

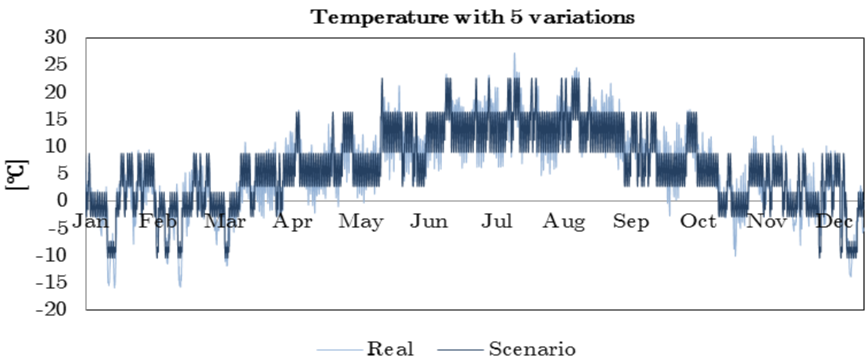

Figure 5-15 5 Scenarios vs. real values

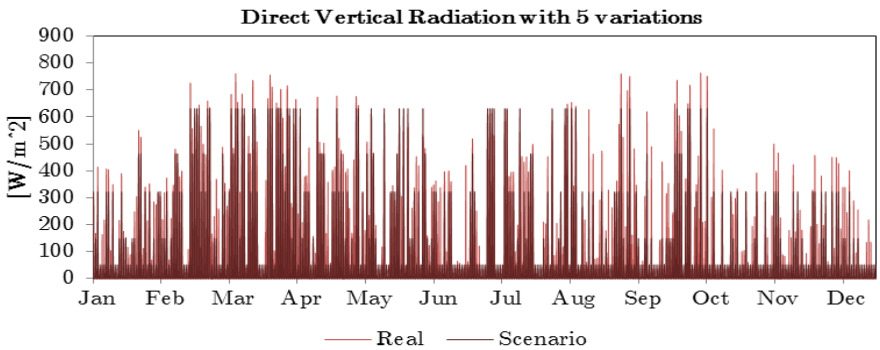

Figure 5-16 5 Scenarios vs. real values

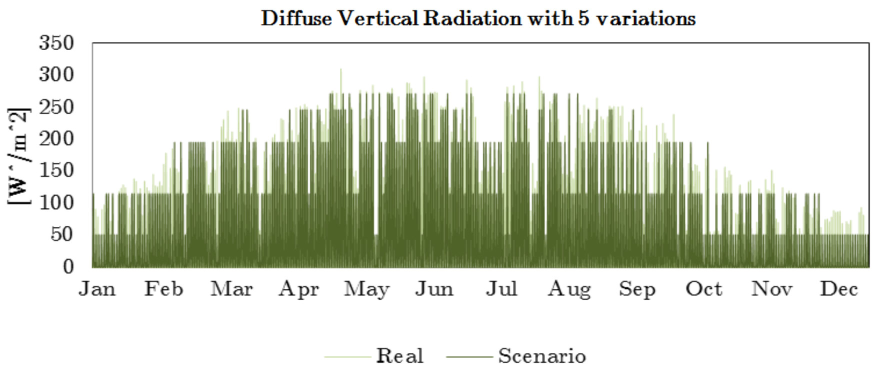

Figure 5-17 5 Scenarios vs. real values

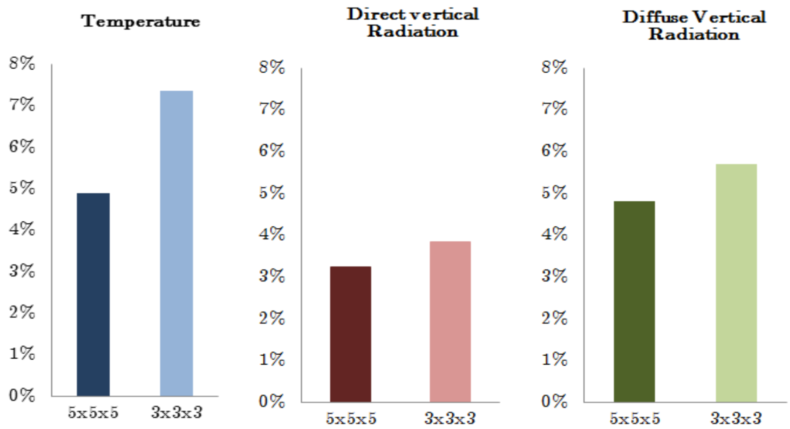

Figure 5-18 Relative deviation comparison between 125 and 27 scenarios

5.4 Results analysis

Looking at Figure 5-9 to Figure 5-11 it seems that the scenarios are representing the real case with reasonable accuracy. Undoubtedly there are some larger deviations which are expected as only three scenarios for a parameter are to cover a whole year. As the scenarios are based on averages it is natural that the extreme values will be moderated and so the really high and low real days stand out in the graphs. From the deviation analysis it was found that temperature had the largest deviation which is noticeable in the graphs. The winter months show the biggest deviation with February topping at an average of 8.8% relative deviation and a worst week of 11%. The deviation for the diffuse radiation scenarios increases during summer with July showing a maximum of 8.2%. This is a result of some very sunny days in this period which outranged the highest scenario. The relative deviation for direct diffusion is generally quite small varying between 3 and 5% with the lowest being in the summer months where the best week was only 2.2% off the real case. The penalization of going from 125 down to 27 possible scenarios is presented by Figure 5-18. The average relative deviation for temperature is raised from 4.90% to 7.35% which is an increase of 50%. The increase is only 18.5% and 18.75% for direct and diffuse radiation respectively. The numbers are calculated using the same input ranges for the categories in both cases which were optimized for 27 day- scenarios. If the input was optimized for the 125 scenario case the difference in relative deviation between the two cases is expected to be even bigger. Still, it is clear from the figures that the 125 scenario case is more accurate.

5.5 Conclusion

With optimized input ranges the excel program can create and distribute scenarios in a way that represents real values with quite good accuracy. Choosing a lower number of scenarios will increase the relative deviation, but the annual average relative deviations was only 7.35%, 3.84% and 5.71% for temperature, direct and diffuse radiation respectively when using 27 possible scenarios. Because of relatively similar humidity curves during summer only the average day for July was used as a scenario for this parameter. The maximum number of necessary simulations needed was reduced with 93%.

6 SCENARIOS FOR THE REFRIGERATION MODEL

6.1 Purpose

The data created for the ventilation model was made so that estimation on the heat *demand* of the store could be generated. The purpose of the refrigeration model is to estimate the heat *available* for recovery and the SPF. In the same manner as for the ventilation a set of scenarios will provide the information needed to generate these estimates. Variable inputs for the refrigeration model are ambient temperature and cabinet cooling demand.

6.2 Approach

Outdoor temperature scenarios were already produced as part of the weather scenarios for the ventilation model and were naturally selected for the refrigeration model as well. Heat load on cabinets was estimated using measurements from a similar plant at another REMA 1000 supermarket. The measurements were on an hourly basis over the course of one year. Because the heat load on cabinets are mainly determined by the indoor temperature the variation with seasons is small. The measurements showed however that there was a slow increase in heat load during spring which levels out during summer before decreasing during fall. It was therefore decided to create 3 heat load scenarios.

- Summer
- Winter
- Spring/Fall

An average for each hour of day for every season was created to make the three load scenarios.

6.3 Results

The three temperatures are the same as depicted in Figure 5-6 and Figure 6-1 give the curves for the cabinet cooling demand.

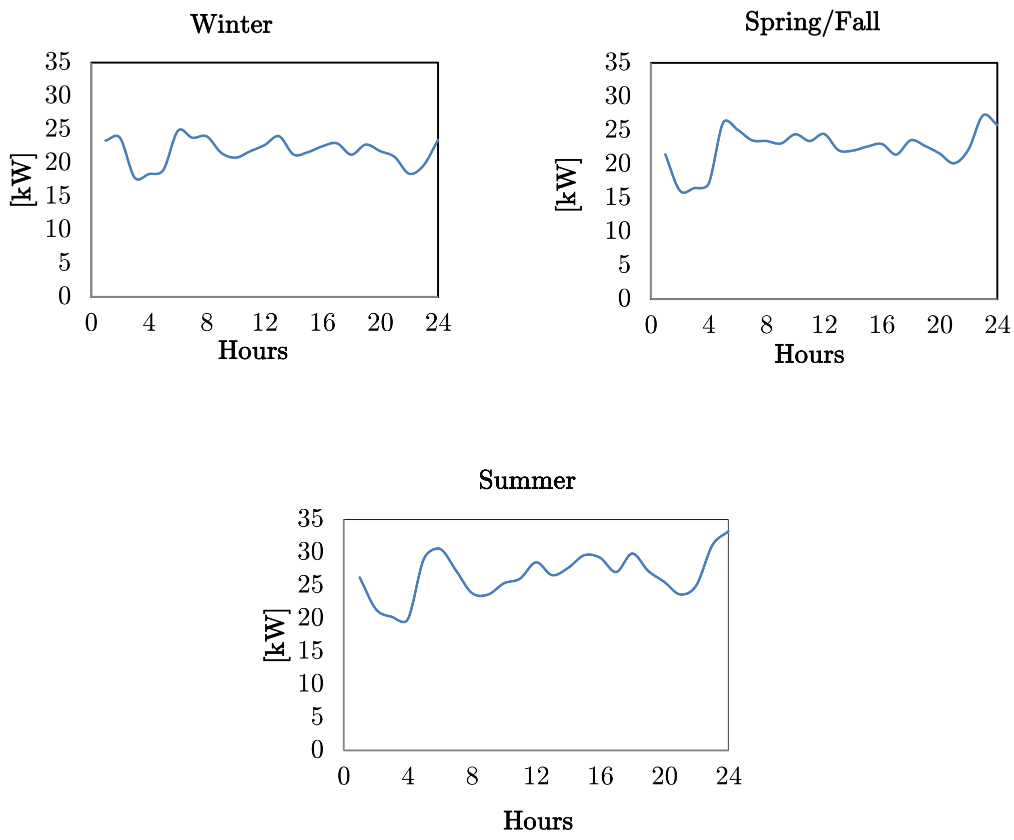

Figure 6-1 Cooling demand curves for the cabinets during winter, summer, spring and fall

7 SIMULATIONS OF THE REFRIGERATION PLANT

7.1 Purpose

The intention of the whole system at Kroppanmarka is that the heat demand is covered entirely by condensation/gas cooling heat from the refrigeration plant. The heat will be reclaimed by two heat exchangers, one dedicated to a hydronic floor heat system and the other to the heat accumulation tanks which are connected to the HVAC system. How much heat that is available for these heat exchangers at a given point is influenced by the ambient conditions and the operational mode of the plant. The amount of heat available to the floor heat exchanger is also largely dependent on the return temperature of the water. By simulating variations of these variables one can find an estimate on the heat available in each case and the corresponding efficiency of the plant. The results can be combined in such a way that they represent a whole year which in turn will be compared to the annual heating demand of the store. The goal is to see whether or not the refrigeration plant can in fact cover the entire demand without the need of an electrical heater and with good efficiency.

7.2 Approach

Variations			
Cooling Demand	Ambient Temperature	Operational Mode	Floor Return Temperature [°C]
Winter	Low	Standard (STD)	18
Spring/Fall	Medium	HighPressure (HP)	23
Summer	High	ExtraEvaporator (EE)	28

Table 7-1 Simulation variables for the refrigeration plant

Table 7-1 shows all the variations for each of the different variables. The three variations of cooling demand and ambient temperature are the scenarios described in the previous section. Since the cooling demand "Winter" always will occur at "Low" ambient temperature, and the "Spring/Fall" demand always occur at "Medium" ambient temperature, and likewise for “Summer” and “High” these combinations will from this point on only be referred to as the seasons “Winter”, “Spring/Fall” and “Summer”. Three different operational modes for the plant were selected for the simulations

- Standard Mode: This operational mode is that mode that yields the highest COP for the refrigeration plant. The condensation temperature is 12 ℃ and the corresponding gas cooler pressure is 48 bar.
- High Pressure Mode: This mode refers to the case where the gas cooler pressure is raised to 120 bar which is the maximum operational pressure. The operation is now transcritical and heat rejection occurs at gliding temperature.
- Extra evaporator: Heat is drawn from the well which is transferred through the extra evaporator. The gas cooler pressure is at 120 bar and the effect of the well is assumed to be 30 kW

Table 7-2 shows all the simulation scenarios that are performed. Each of the operational modes will be tested with varying floor return temperature making a total of 9 simulations per season.

Simulation Scenarios			
Cooling Demand	Ambient Temperature	Operation Mode	Floor return Temperature [℃]
		Standard	18
Winter	Low	High Pressure	23
		Extra Evaporator	28
		Standard	18
Spring/Fall	Medium	High Pressure	23
		Extra Evaporator	28
		Standard	18
Summer	High	High Pressure	23
		Extra Evaporator	28

Table 7-2 Combinations of variables

The simulations produce information regarding mass flow, density, temperatures, pressures, heat flows and more at any given point in the refrigeration cycle, but the main output is the heat available for the heat exchangers and the COP.

$$Refrigeration\ COP = \frac{Q_{Freeze} + Q_{Cool}}{W_{LP} + W_{HP}} \quad (5)$$

$$Total\ COP = \frac{Q_{LT} + Q_{MT} + Q_{GCI} + Q_{GCII}}{W_{LP} + W_{HP}} \quad (6)$$

$$Q_{Flow} = \dot{m}_{refrigerant} \times \Delta h_{2-1} \quad (7)$$

Available heat and COP curves for a whole year is created for each operational mode and floor return temperature with the results from the simulations. This will be generated by synchronising the simulation results with the category distribution for temperature from the excel weather scenario program. For instance, for the scenario *Standard Mode-with-18*℃ *-return-temperature* there are three variations; one for each season. In order to know which variation should be used for each day of the year it is necessary to check the nature of each day in terms of ambient conditions. This of course, is already done in the excel scenario maker program where the ambient temperature downloaded from meteonorm was analysed and each day categorized. If the first day in the program fell under the category "Low" for temperature, then the "winter case" case of *Standard-Mode-and- 18*℃*-return-temperature* is chosen for this day of the year, if the next day fell under the category "High", the summer case is selected and so on for the whole year. The available heat and COP curves for the whole year are in this way synchronized with the same conditions as for the ventilation model. This is very important. If the two models are simulated with different conditions the results would be a lot less reliable. When the ventilation model is finished and the scenarios created in this work is used the results will be compatible with the results from the refrigeration simulations in this thesis.

7.3 Results

Some of the results from the simulations are presented in this section. Because heat recovery is essential to the system the results from the winter scenarios are displayed since this is the time of largest heat demand. The rest can be found in Appendix C.

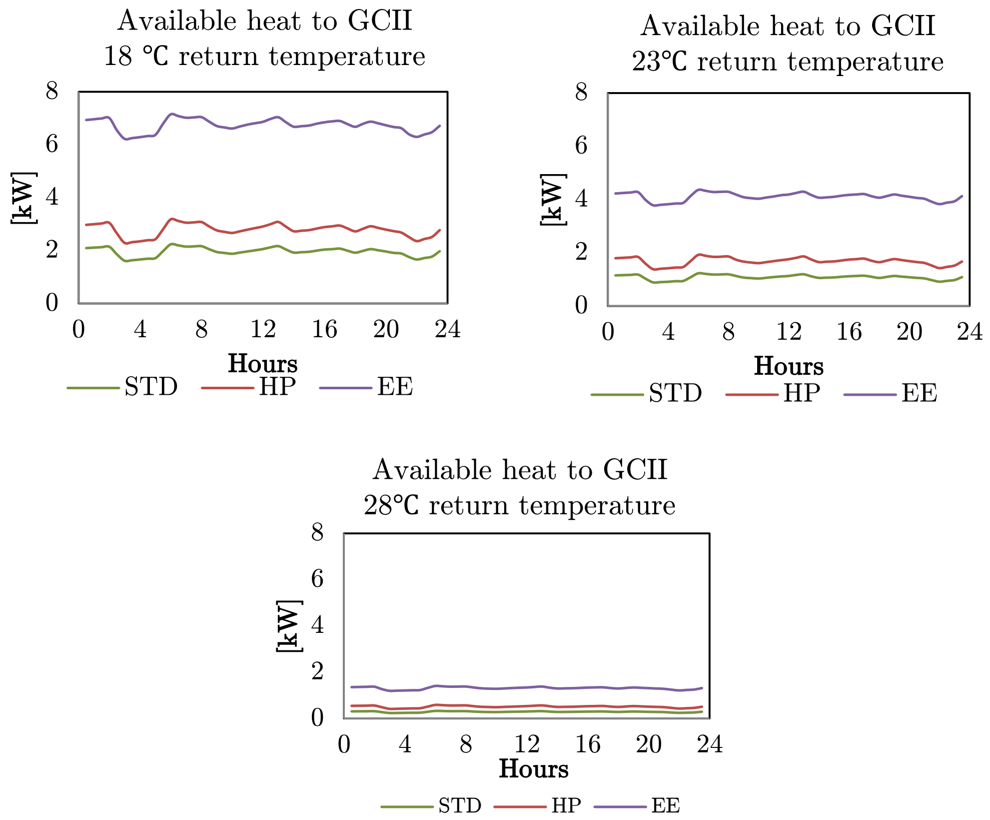

Figure 7-1 Available heat for GCII at varying floor return temperature for each mode

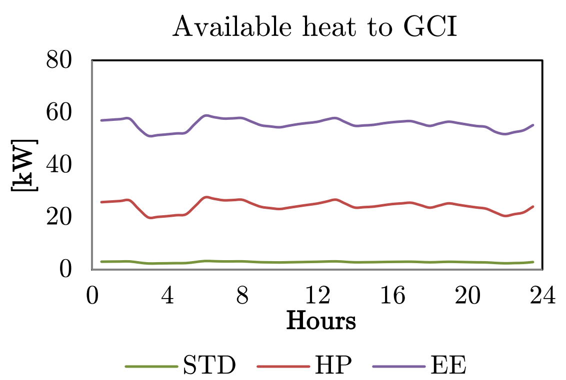

Figure 7-2 Available heat to GCI for each mode

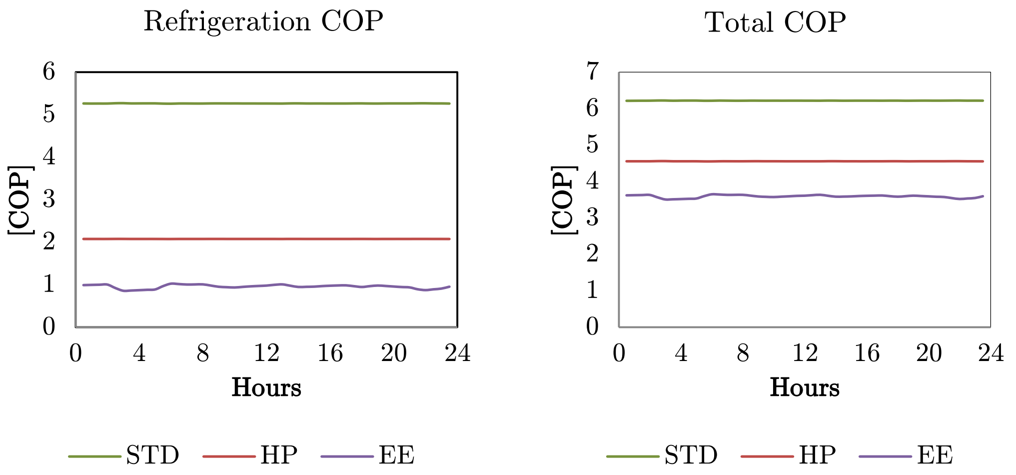

Figure 7-3 Refrigeration COP is independent of floor return temp. The total COP is in this figure at 23 °C .

7.4 Results Analysis

Standard Mode

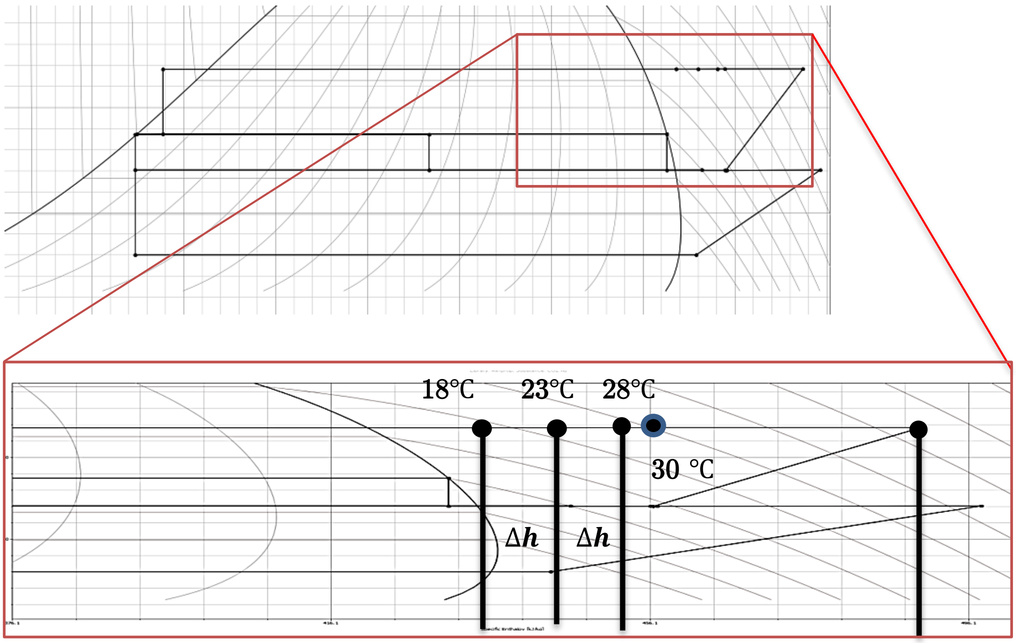

Figure 7-4 P-H diagram. Enthalpy differences as a consequence of varying floor return temperature

Figure 7-4 clearly illustrates the consequence of the floor return temperature in standard mode. Since the water-glycol inlet temperature of GCI was set to 30°C it is only heat below this temperature that is available for GCII and the floor heating system. The enthalpy difference in each case is marked in the figure by the stapled lines. This explains the big reduction in available heat for the floor as the return temperature increases in Figure 7-1. The absolute values are small but the relative differences are big. At 18℃ return temperature the available heat in standard mode is on average 2 kW during winter while at 28℃ it is only 0.3 kW. The standard mode in the simulations is at 48 bar and as Figure 7-4 shows all heat exchange in GCI and GCII take place to the right of the dew point line (despurperheating). This means that the energy well must provide the rest of the de-superheating, full condensation and finally sub cooling of the CO_2 which in standard mode is about 27 kW of cooling if the return temperature is 28℃. The total energy rejected to the well for one day is in that case 1277 kWh.

The heat available to GCI and the accumulation tank is not influenced by the return temperature of the floor heating in any mode. For standard mode the available heat during winter lies around 2.85 kW. The total heat available to both gas-coolers goes from 3.67 kW at 28℃ return temperature to about 5.10 kW at 18℃.

The refrigeration COP is also independent of the floor return temperature as long as the energy well can cool down the CO_2 down to the same temperature in each case. Figure 7-3 shows a COP of 5.27 for standard mode which is the highest refrigeration performance of any of the simulations. The total COP on the other hand depends on the floor return temperature as Q_{GCII} in equation (6) varies. The total COP is therefore largest when the return temperature is the lowest. For 18℃, 23℃ and 28℃ the total COP was 6.43, 6.22 and 6.03 respectively.

The power consumption of the compressors in standard mode averages at 4.14 kW, 4.25kW and 5.04kW for Winter, Spring/Fall and Summer respectively. The only real difference between the seasons is that the cooling demand varies. As it increases so does the mass flow rate of refrigerant. Both the compressor work and the available heat is proportional the mass flow rate (equation (7)) hence their increase is just a consequence of higher cooling demand. The cycle in the ph-diagram looks the same and so the COP values stay constant with the changing of seasons. This needs further explanation; the reason the cycle looks the same is because of the energy well. The CO_2 is conventionally condensed or sub-cooled by the ambient air in which the change of season will most definitely change the cycle. The CO_2 outlet temperature can only get as cold as the outside air. The temperature of the bed rock for the energy well, on the other hand, stays more or less constant (and is assumed constant) which is why the CO_2 is cooled down to the same state independent of season. Pressure drop in tubes and components is also assumed equal to zero and so the increased mass flow doesn't affect this either.

Average values for Standard Mode			
Floor Return Temperature [℃]	18	23	28
Power Consumption [kW]	4.14	4.14	4.14
GCI [kW]	2.86	2.86	2.86
GCII [kW]	1.96	1.08	0.29
Ref COP [-]	5.27	5.27	5.27
Tot COP [-]	6.43	6.22	6.03

Table 7-3 Summary of Standard Mode simulations

High Pressure Mode

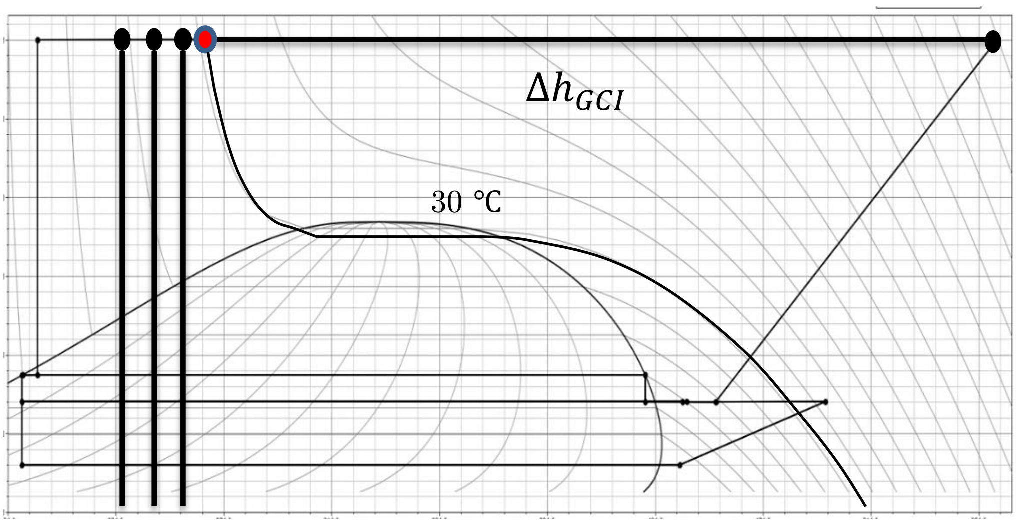

Figure 7-5 P-H diagram with HP mode cycle

As the pressure is increased to 120 bar the available heat is also increased. For GCII at 18℃ the available heat goes up 0.8 kW from standard mode. This is however a insignificant amount compared to the increase for GCI. At standard mode it was 2.85 kW, in high pressure mode it is 24.2 kW. The reason becomes apparent when investigating Figure 7-5. As the pressure is increased so is the enthalpy range between the point of discharge and the 30℃ isotherm. Compared to standard mode (48 bar) the specific enthalpy difference is 8.5 times higher. The total heat available is now dominated by the amount going to the tank heat exchanger and so the variations of floor return temperature has a relatively smaller impact.

The refrigeration COP is drastically reduced to around 2 by this high pressure operation. This is because the power consumption of the high stage compressor (W_{HP}) goes up while no additional cooling is achieved. Of course, this COP is not very relevant as it is more *heat* the cycle is meant to produce. In equation (6) both Q_{GCI}, Q_{GCII} and W_{HP} increases and the total COP at 18℃, 23℃ and 28℃ return temperature becomes 4.66, 4.56 and 4.45 respectively. It is about 1.5 points lower than for standard mode, but it delivers over 5 times more heat to the gas coolers while the power consumption is only 2.5 times higher.

Average values for High Pressure Mode			
Floor Return Temperature [℃]	18	23	28
Power Consumption [kW]	10.49	10.49	10.49
GCI [kW]	24.20	24.20	24.20
GCII [kW]	2.79	1.68	0.51
Ref COP [-]	2.08	2.08	2.08
Tot COP [-]	4.66	4.56	4.45

Table 7-4 Summary of High Pressure mode simulations

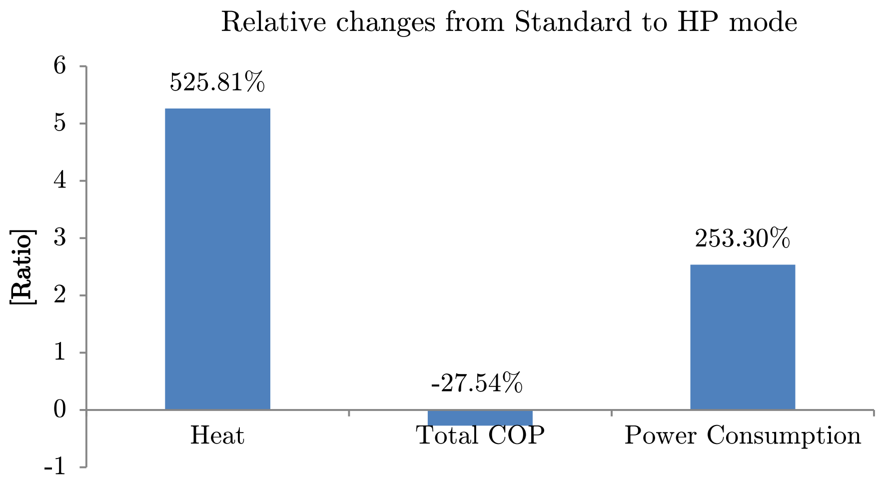

Figure 7-6 STD vs HP mode

Extra Evaporator mode

In the event where enough heat can't be provided by operating at high pressure the extra evaporator can be used. Drawing 30 kW of heat from the energy well increased the mass flow rate of refrigerant from 0.09 kg/s to 0.22 kg/s and the system delivers 62 kW to the gas coolers during winter with 18 ℃ return temperature. At 23℃ and 28℃ return temperature the available heat was 59.7 kW and 56.7 kW.

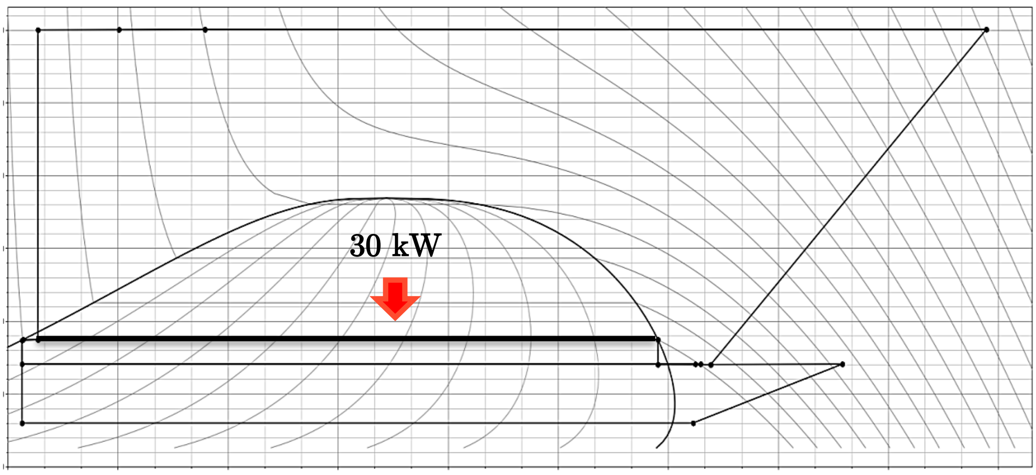

Figure 7-7 P-H diagram of the EE mode cycle

Both the heat available and compressor work is as mentioned proportional to the mass flow rate of refrigerant and so the work increases with the same factor as the available heat has. The work is now at 22.7 kW which is 5.5 times higher compared to standard mode. The system still provides the same cooling for the cabinets and so the refrigeration COP as plummeted to 0.96. In other words; the energy input of the compressors is larger than the energy withdrawn from the cabinets. It is still the total COP that matter in this case which also has decreased, but only down to 3.7.

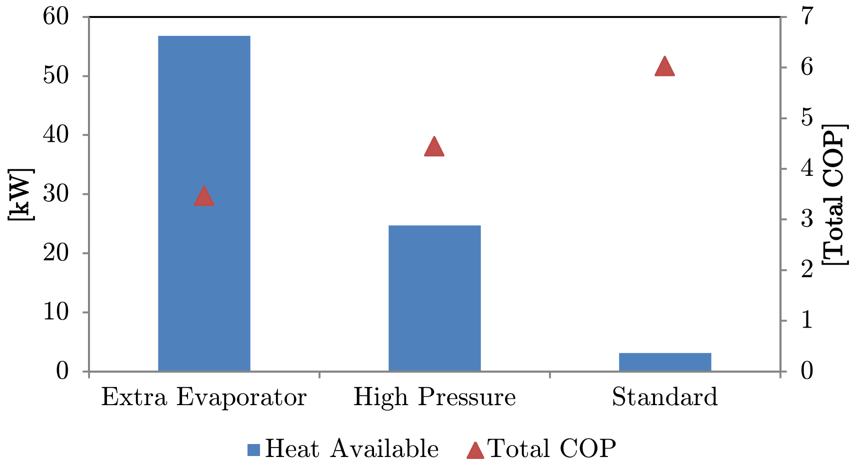

Figure 7-8 Available heat and COP for the three simulated modes of operation. All results are during winter with 28°C floor return temperature.

Simulation Results Summary Winter							
Mode	Return Temp [°C]	GCI [kW]	GCII [kW]	Total Heat [kW]	Ref. COP [-]	Total COP [-]	Compressor work [kW]
	18		1.96	4.82		6.43	
STD	23	2.86	1.08	3.94	5.27	6.22	4.14
	28		0.29	3.15		6.03	
	18		2.79	26.97		4.66	
HP	23	24.18	1.68	25.86	2.08	4.56	10.49
	28		0.51	24.70		4.45	
	18		6.74	62.20		3.71	
EE	23	55.47	4.11	59.57	0.96	3.59	22.76
	28		1.31	56.78		3.47	

Table 7-5 Summary of simulation results for all modes. Heat to GCI and the Ref. COP is independent of floor return temperature.

8 ANNUAL ESTIMATIONS

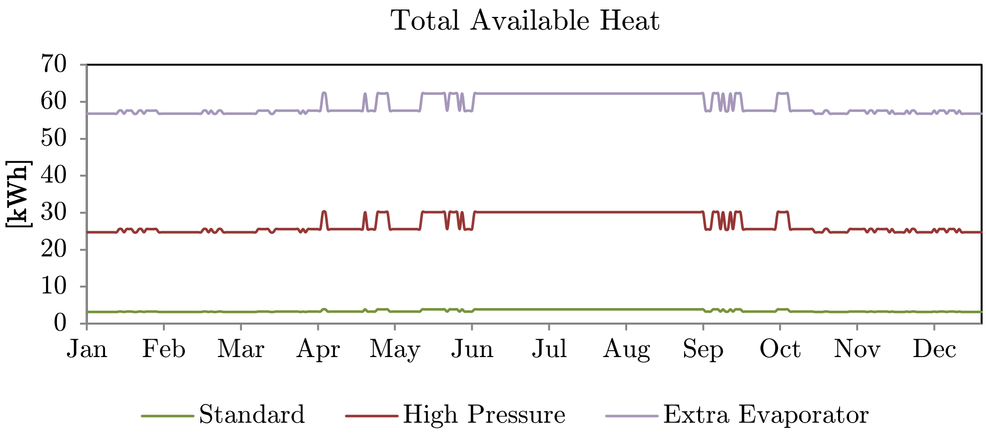

Figure 8-1 Total available heat for each of the modes during one year

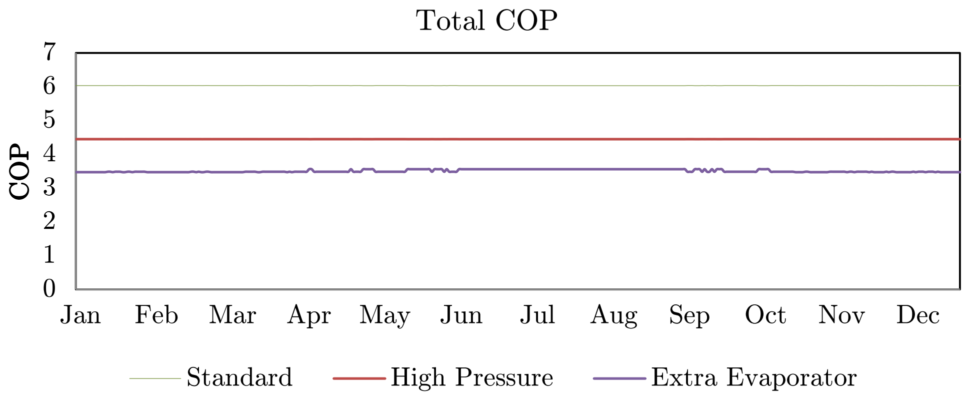

Figure 8-2 Total COP for each mode during one year

8.1 Purpose

Figure 8-1 shows annual heat available per mode at a floor return temperature of 28°C. The corresponding COP is plotted in Figure 8-2. The values plotted are the daily averages. There is slightly more heat available during summer for all the cases due to the increased heat load on cabinets. It is crucial that that the right mode is engaged at all times so that enough heat is produced at an optimum COP while keeping the compressor work at a minimum otherwise. To find this will be an important task once the ventilation model is complete and the work to integrate the two models can begin. At this point it is still possible to do some evaluations on annual available heat, control strategies, and their associated energy consumption and efficiencies with respect to the heat demand of the building.

8.2 Approach

The data from the simulations were organized into hourly values for a whole year using the weather analysis to distinguish the three variations of refrigeration load and the associated value for work, COP and available heat for each hour. This was done for all operational modes at every return temperature variation. Two reference strategies are presented without the use of accumulation tanks and four other strategies were tested with the tank included. A simplified approach for the behaviour of the accumulation tanks was used. A program was created in excel to generate the results from each of the strategies. The heat demand is inserted in a data sheet with a specific value for each hour of the year. The mode of operation and its corresponding compressor work and efficiency is decided by the control settings for each hour and depends on the heat demand of that specific hour. Only one of the three modes (STD, HP and EE) that have been simulated can operate for each hour. The specific control is explained for each strategy.

Since the heat demand from the ventilation model could not be estimated, it was decided to test two different heat demand scenarios. The first heat demand comes from measurements data for the REMA 1000 store "Dragvoll" (case 1). This is the same store from which the refrigeration duty was based on. Hourly values over the course of one year were collected. The second case comes from a heat demand simulation performed by Erwan Vaujany for Kroppanmarka with the CREATIV supermarket calculation tool in 2011 (case 2). The total heat demand of the case 1 was 95 088 kWh while the other was at 175 352 kWh.

Strategy 1

The simplest control strategy is where only one mode is operated for the whole year without the use of the accumulation tanks. In the situations where the heat available for recovery only covers a part of the demand, an electric heater will engage to deliver the remainder. For instance, if the heat demand at a certain hour is 15 kW and the available heat from the plant is 10 kW, then 5 kW of electrical power is used.

Strategy 2
Another strategy is to select the mode with the least power consumption while still providing enough heat. If the available heat at one hour is 5 kW, 22 kW and 52 kW for STD, HP and EE respectively and the demand is 24 kW, then EE will be selected. This eliminates the need for electrical heating as long as EE mode is sufficient.

With Tank

Making use of the accumulation tanks means that surplus heat can be stored and used at a later point reducing waste heat and compressor power consumption. A simplified representation of the tank is used in this study. Whenever a mode at a certain point produce more heat than the demand, the extra heat will go to the tank. If there is more heat stored than what is needed, the tank can provide the necessary effect which is then subtracted from the total available heat stored. The amount of heat in the tank at hour n is

$$Q_{Stored_{n+1}} - Q_{stored_n} = Q_{GCI_n} + Q_{GCII_n} - Q_{demand_n} \tag{8}$$

For instance, if the demand for one hour is $Q_{demand_n} = 25\,kWh$, and $Q_{GCI_n} + Q_{GCII_n} = 5\,kWh$ is produced from the plant, and the tank has stored $Q_{stored_n} = 50\,kWh$, then the available heat in the next hour will be $Q_{Stored_{n+1}} = 30\,kWh$. If the following hour $n+2$ has a demand of 15 kWh and 20 kWh is produced from the refrigeration plant, then the available heat in the tank for the next hour will be 35 kWh. For the strategies only the total heat produced is considered, it does not distinguish between the heat from each gas cooler.

Strategy 3

As long as the available heat in the tank exceeds the heat demand Standard mode will be engaged. If the heat demand is bigger, then the mode with lowest power consumption that can cover this demand on its own will be engaged. For example;
$Q_{stored_n} = 20\,kWh$,
$Q_{demand_n} = 35\,kWh$,
Standard mode will give 5 kWh,
HP mode gives 25 kWh,
EE gives 55 kWh,

Since the $Q_{demand_n} > Q_{stored_n}$ standard mode will not be selected. $Q_{HP_n} < Q_{demand_n}$ so HP won't be chosen either which means EE mode is engaged. $Q_{Stored_{n+1}}$ then becomes 40 kWh. If the demand for the next hour stays the same, then standard mode will engage.

Strategy 4

This strategy is the same as the previous except now the added sum of the available heat in the tank and the heat from the plant has to be exceeded by the demand before a higher mode is engaged. In other words if $Q_{stored_n} + Q_{standard} < Q_{demand_n}$ then try $Q_{stored_n} + Q_{HP} > Q_{demand_n}$, if this is not true then use EE mode. This means that it takes more before EE is engaged.

Strategy 5

This approach utilizes the tank until the available heat is less than the demand in which EE mode is engaged immediately. EE mode delivers on average about 30 kW more heat than HP mode and so STD mode, which yields the highest COP, is allowed to run for more hours after EE mode is finished compared to HP mode. HP mode is never used in this strategy. Whenever the tank can cover the demand ST mode is operational.

Strategy 6

EE mode is the operational mode with the lowest total COP. Strategy 5 is about eliminating the use of this mode by setting a minimum value for the available heat in the tank. If the tank falls below 200 kWh, then HP is engaged to recharge the tank. The goal is that there will always be enough heat available in the tank so EE is not engaged. Whenever the tank can cover the demand STD mode is operational.

8.3 Results

The results from the strategies and different heat demands are presented with 28 °C floor return temperature to show the situation in the worst case scenario. More results can be found in Appendix D.

Strategy 1:
Figure 8-3 show the three operational modes over the course of one year compared to the heat demand of case 1. The same can be found for case 2 in Figure 8-4. The main results are summarized in Table 8-1 and Table 8-2.

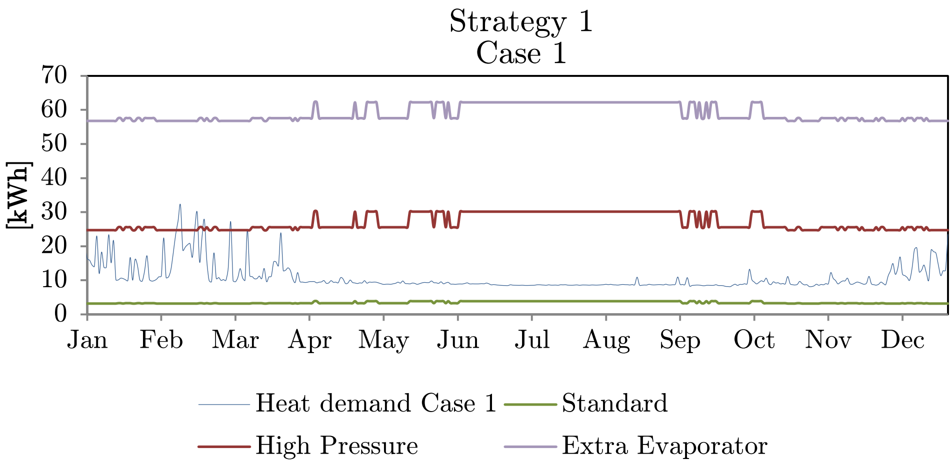

Figure 8-3 Case 1 with the three modes

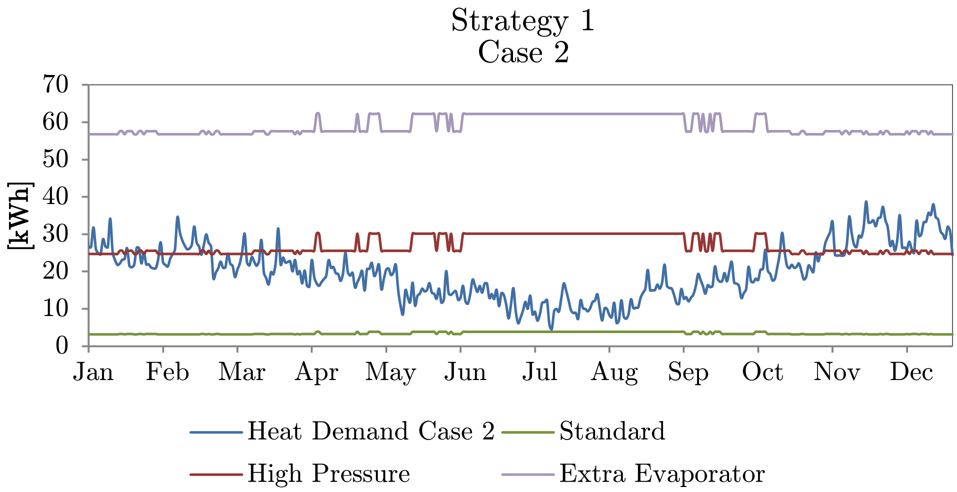

Figure 8-4 Case 2 with the three modes

Case 1			
[kW]	STD	HP	EE
Heat Available	30 085	235 925	516 774
Electric Heating Needed	65 003	2 952	0
Excess Heat	0	140 837	421 686
Compressor work	39 421	99 832	207 501
Total Consumption	104 424	102 784	207 501

Table 8-1 Case 1 summary

Case 2			
	STD	HP	EE
Heat Available	30 085	235 925	516 774
Electric Heating Needed	147 091	17 856	0
Excess Heat	0	60 582	341 431
Compressor work	39 421	99 832	207 501
Total Consumption	186 512	117 688	207 501

Table 8-2 Case 2 summary

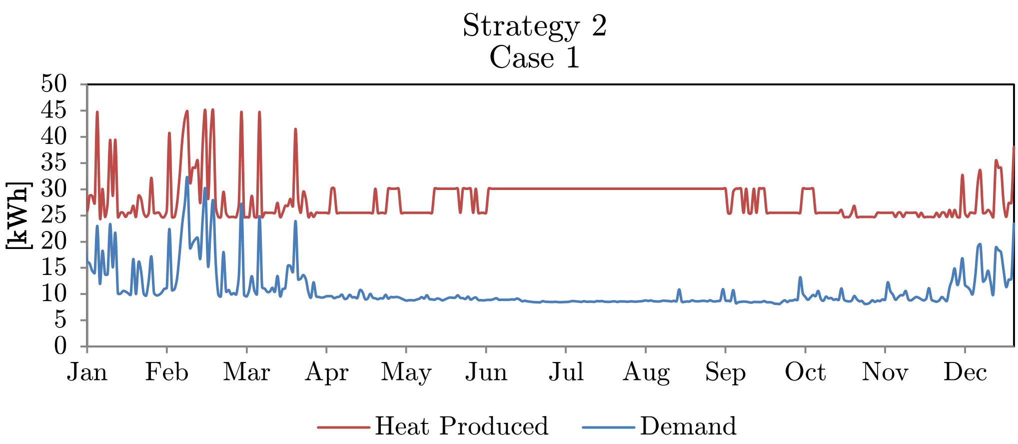

Figure 8-5 Heat curves for strategy 2 case 1

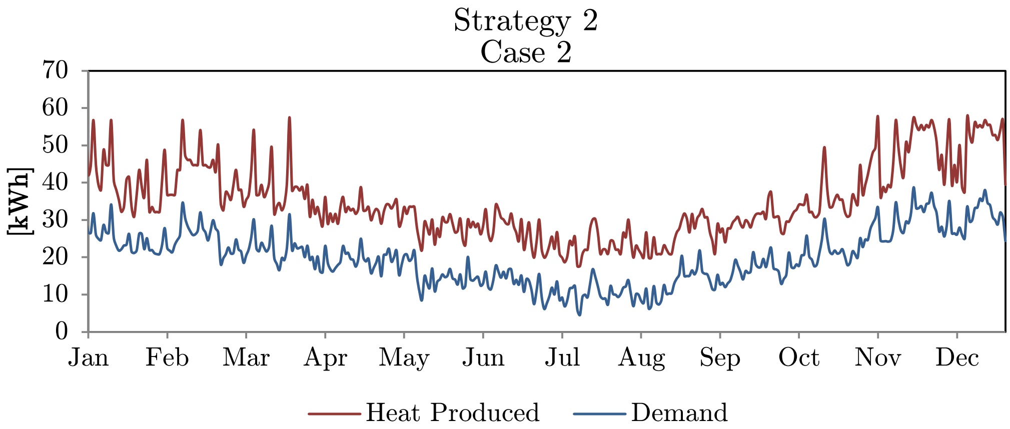

Figure 8-6 Heat curves for strategy 2 case 2

Strategy 2			
		Case 1	Case 2
Energy [kWh]	Work	104 256	127 718
	Heat Available	247 597	305 433
	Heat Wasted	152 509	130 090
Efficiency [-]	SPF	2.90	2.99
Mode Frequency	Standard	0.0 %	7.4 %
	HP	95.8 %	61.6 %
	EE	4.2 %	30.9 %

Table 8-3 Strategy 2 summary for each case

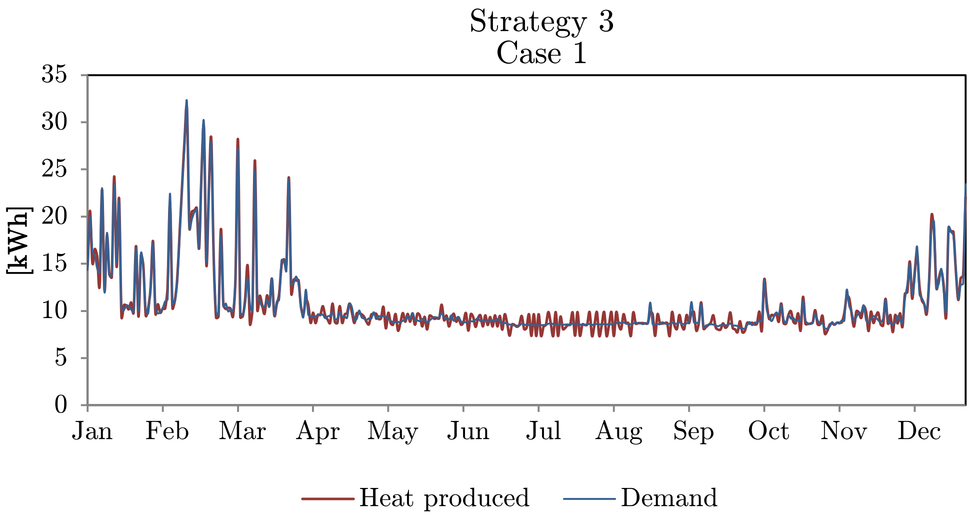

Figure 8-7 Heat curves for strategy 3 in case 1

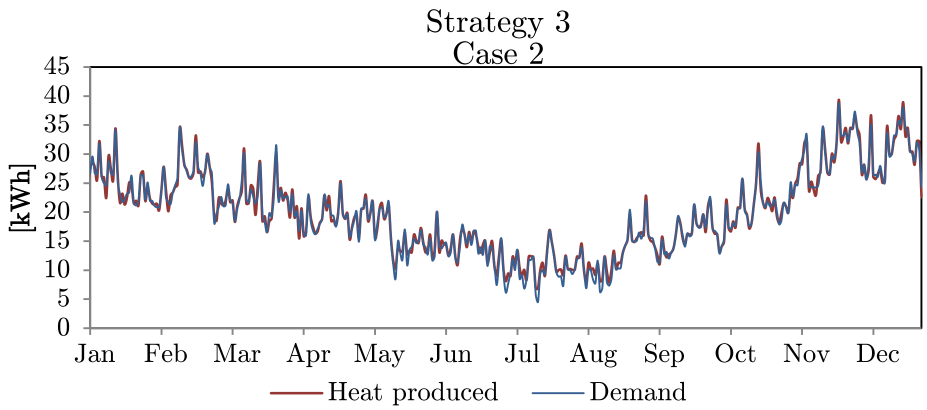

Figure 8-8 Heat curves for strategy 3 in case 2

Strategy 3			
		Case 1	Case 2
Energy [kWh]	Work	59 192	86 374
	Electrical Heat Needed	0	0
	Heat Available	95 102	175 352
	Heat Wasted	13	10
	Total Power Consumption	59 192	86 374
Efficiency [-]	SPF	5.11	4.43
Mode Frequency	Standard	71.7 %	52.5 %
	HP	25.6 %	29.8 %
	EE	2.7 %	17.6 %

Table 8-4 Summary for strategy 3

Strategy 4			
		Case 1	Case 2
Energy [kWh]	Work	58 838	84 042
	Electrical Heat Needed	0	0
	Heat Available	95 105	175 354
	Heat Wasted	17	11
	Total Power Consumption	58 838	84 042
Efficiency [-]	SPF	5.14	4.55
Mode Frequency	Standard	69.5 %	38.6 %
	HP	29.2 %	53.1 %
	EE	1.3 %	8.3 %

Table 8-5 Summary for strategy 4

Strategy 5			
		Case 1	Case 2
Energy [kWh]	Work	61 775	89 513
	Electrical Heat Needed	0	0
	Heat Available	95 125	175 392
	Heat Wasted	37	50
	Total Power Consumption	61 775	89 513
Efficiency [-]	SPF	4.89	4.27
Mode Frequency	Standard	86.4 %	69.6 %
	HP	0.0 %	0.0 %
	EE	13.6 %	30.4 %

Table 8-6 Summary for strategy 5

Strategy 6			
		Case 1	Case 2
Energy [kWh]	Work	58 620	79 544
	Electrical Heat Needed	232	8 655
	Heat Available	95 196	166 665
	Heat Wasted	108	-8 678
	Total Power Consumption	58 852	88 198
Efficiency [-]	SPF	5.14	4.34
Mode Frequency	Standard	67.7 %	30.8 %
	HP	32.2 %	69.2 %
	EE	0.0 %	0.0 %

Table 8-7 Summary for strategy 6

8.4 Results Analysis

Except for in standard mode strategy 1 represents an unrealistic way of controlling the plant, but shows how the different modes perform over the course of a year with the two heat demands. For Case 1 the energy consumption of standard mode is only 39 041 kWh, but since the heat it provides is nowhere near the demand, over 60 000 kWh of electric energy have to be used. One can see from Figure 8-3 that the available heat line for standard mode is always below the heat demand line. Still, this would be the way to operate the plant if noe heat recovery was considered. In HP mode there is in total 140 837 kWh more heat produced than needed. Due to some very cold winter days the electric heater had to be used for for almost 3000 kWh nonetheless. The extra evaporator mode is able to provide far more heat than is necessary at all times and after one year about 516 000 kWh of energy is made available. This has come at the expense of much compressor work which for this mode was at 207 501 kWh a year. The HP mode had the lowest total consumption with 102 784 kWh, about 2000 kWh less than STD mode would claim.

The heat demand estimation of Case 2 is not only nearly twice as high as Case 1, but the shape of the curve is also different. It is therefore quite interesting to see how the different modes fan out in the two cases. For strategy 1 the available heat curves are the same and only the demand curve differs. Because the demand is so much higher, the standard mode needs even more electric heat assistance which in this case becomes 147 091 kWh. This makes the total consumption 186 512 kWh over one year in standard mode, which is only 21 000 kWh less than EE mode claims. EE mode also provides 341 431 kWh more than necessary. HP mode needs a little less than 18 000 kWh from the electrical heater, but is still the best solution of the modes with a total consumption of 117 688 kWh. Without any heat recovery and with the plant operating in standard mode the total power consumed for heating and refrigeration would be 214 773 kWh. This shows that even with the simplest strategy control of heat recovery one could save 214 773 -117 688 = 97 085 kWh per year for case 2.

Strategy 2 involved always trying to provide the necessary heat to the store directly from the plant. No electric heat had to be used for any of the heat demands because EE was always sufficient. The problem however is that since this evaluation only has the three simulated modes available the heat delivered is sometimes a lot higher than necessary for this strategy. For instance, if the demand is 1 kWh higher than what HP mode can provide then EE mode is engaged which provides about 28 kWh more than necessary in this case. This results COP's below 1 and this is why the results from strategy 2 is worse than HP mode in strategy 1.

The first two strategies were tested so that the results can be used as references for the strategies which include the tank (strategy 3-6). This is interesting because the accumulation tanks are one of the key features of the new supermarket system designed by SINTEF.

With strategy 3 the total power consumption in Case 1 has dropped to 59 192 kWh (Table 8-4). This is a reduction of 42% compared to the best case of strategy 1 and 2. Figure 8-7 shows how the red available-heat-line is swaying above and below the demand line. This is especially clear in the figure between May and September. The times when the red line is above represents situations where more heat is generated than consumed and so the surplus is stored in the tank. When the red line is below the demand line it means that heat is taken from the tank which allows the plant to operate at standard mode and so less heat is produced in these periods, hence the red line is below. For strategy 2 standard mode was never operational, but with the tank it was able to run for 72% of the time. The high total COP of the standard mode contributes to a good seasonal performance factor (SPF) which was 5.11. The use of the tank also means almost no available heat is wasted.

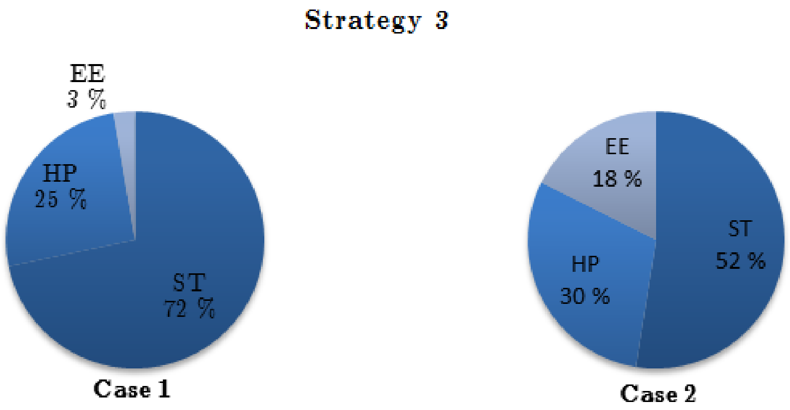

Figure 8-9 Frequency of modes in the two cases with strategy 3

The heat demand for Case 2 is as mentioned much higher and as a result standard mode is not as often operational as in Case 1. In strategy 2 it was running for 7.4 % of the time, but with the accumulation tanks it is up to 52% for strategy 3. The power consumption is reduced by 27 % compared to the best case of strategy 1 and 2 and the SPF is 4.43.

The control of Strategy 4 allows a mode to be operational for a broader range before a higher mode is initiated. In other words, ST mode and HP mode should be allowed to run more often. The results show however that only hours of HP mode went up while standard mode was reduced. The frequency of standard mode is quite related to the frequency of EE mode. It was found by investigating all the operational hours of the year that in some periods after, one hour in EE mode, Standard mode was able to run for up to 6 hours before more heat had to be produced.

Strategies limiting EE mode can then also limit ST mode. For Case 1 the difference was small going from strategy 4 from 3, but Case 2 there was a big shift. EE mode went down 10 % points and ST dropped to 39 % leaving HP mode as the dominant mode at 53%. Maybe a little surprisingly the energy consumption went down for both cases compared to strategy 3. The reduction was only less than 1% for the Case 1, for Case 2 the reduction was 2.7% or 2332 kWh.

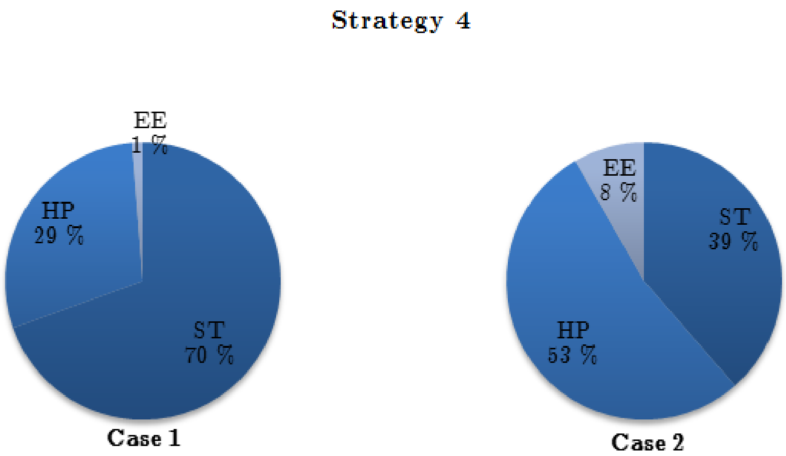

Figure 8-10 Frequency of modes in the two cases with strategy 4

Strategy 5 eliminates HP mode and aims to increase STD mode. Although this has been accomplished it has resulted in higher energy consumption than strategy 3. For Case 1 the SPF is now 4.89 and Case 2 is at 4.27. The energy consumption increased by 4.36% and 3.6 % respectively compared to strategy 3.

The last strategy that was tested to see wether or not EE mode is required to avoid electric heating. For Case 1 the electric heater had to help out with 232 kWh only and so the strategy did almost accomplish the goal. The total energy consumption was almost exactly the same as for Strategy 4 at 58 852 kWh. HP mode was operational for 32% of the time. For Case 2 this strategy failed to deliver. Due to longer periods with high demand the HP mode did not manage to create the necessary heat and as a result the heater had to produce 8 655 kWh to the store. The total consumption was then 88 198 kWh a year which is 5 % more than for strategy 4.

After the results from the different strategies were evaluated a seventh strategy was created based on the results and tested. It works as a combination of strategy 4 and 6. Standard mode is only engaged when the tank capacity is over 200 kWh. If this is not true and there is less available heat in the tank combined with the heat provided by HP mode, only then will EE mode be engaged. Otherwise HP mode will run. This ensures that electric heat is never used and EE mode is only engaged when absolutely necessary.

Strategy 7			
		Case 1	Case2
Energy [kWh]	Work	58 634	82 889
	Electrical Heat Needed	0	0
	Heat Available	95 178	175 413
	Heat Wasted	90	71
	Total Power Consumption	58 634	82 889
Efficiency [-]	SPF	5.15	4.61
Mode Frequency	Standard	67.9 %	30.8 %
	HP	32.0 %	66.0 %
	EE	0.1 %	3.1 %

Table 8-8 Summary for strategy 7

This strategy had the least power consumption for both cases. For the Case 1 it was only marginally less than strategy 4 (0.35%) and for Case 2 it was 1.4% lower. For Case 2 the EE mode was operational for 3.1% of the time and HP mode was dominating with 66 %. For Case 1 EE was almost never used and standard mode dominated with 67.9%.

It is interesting to see that despite the big differences between the two heat demands the reaction was quite similar with respect to the total consumption for the different strategies. As Figure 8-11 shows, strategy 4 and 7 had the lowest consumptions for both cases and strategy 5 was the worst. This was true despite the fact that the total operational time for each mode is very different for the two cases.

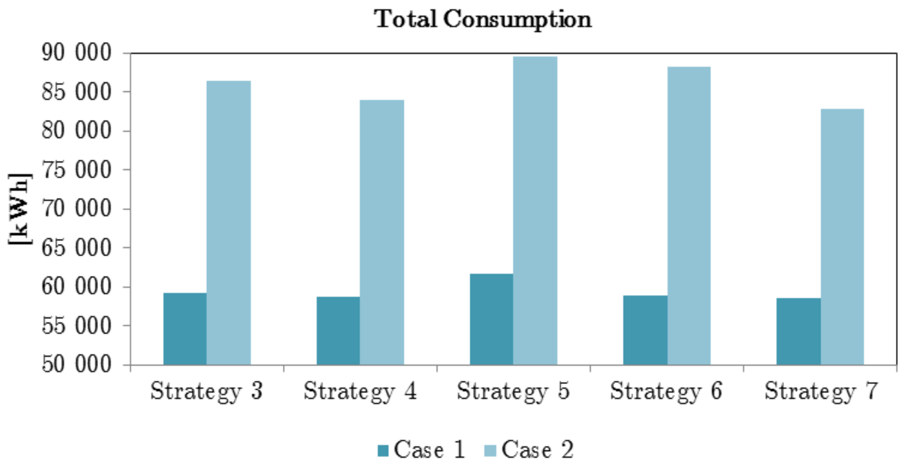

Figure 8-11 Strategy comparison of total consumtion for each case

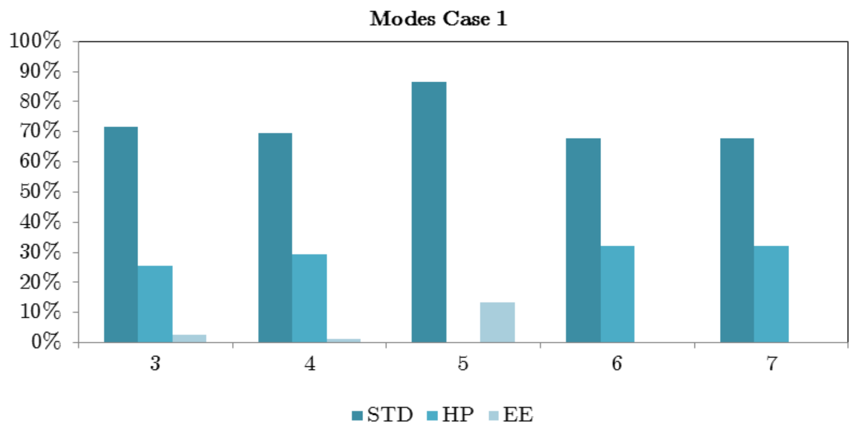

Figure 8-12 Strategy comparison in terms of mode frequency in case 1

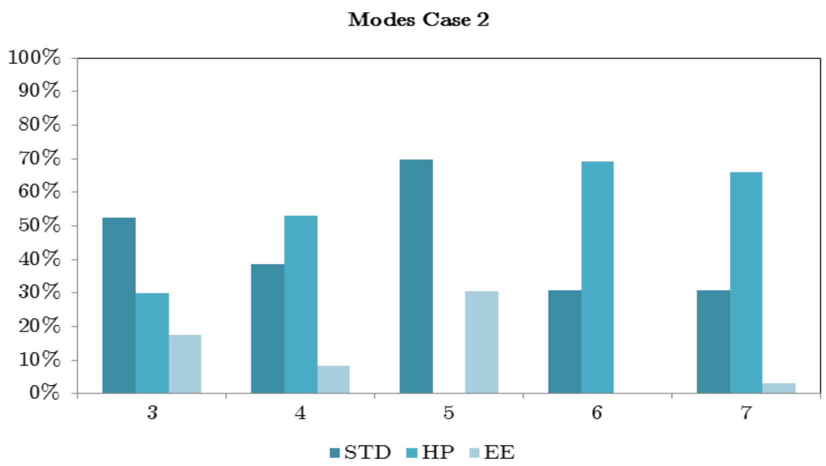

Figure 8-13 Strategy comparison in terms of mode frequency in case 2

There was less variation in the demand from the Dragvoll measurements which led to very small differences between that various strategies. This is apparent when looking at Figure 8-12. Only strategy 5 stands out. In the heat demand estimated for Kroppanmarka however, there were bigger changes. Even though the total consumption of strategy 3 and 7 was low and not too far apart the plant operated very differently as can be witnessed in Figure 8-13. The three modes behaves quite uniquely, as shown in strategy 1, so one could imagine that the outcome of strategy 3 and 7 should be more distinctive. The main reason that this is not the case is because of the accumulation tanks. How heat is provided becomes less important as everything is stored in the tank. It does not matter if 50 kWh of heat is produced when only 22 kWh is needed because the extra amount is stored and not wasted. Still, the results show that if the heat is provided using EE mode rather than HP mode, the SPF goes down since EE yields the poorest COP.

The SPF of Case 1 is for all strategies higher than for Case 2, which is expected. Higher demand require higher modes which have lower COP. The best SPF for the two cases of demand was 5.15 and 4.61 for Case 1 and Case 2 respectively. This of course occurred under strategy 7. Compared to a situation where there is no heat recovery the savings are 75 875 kWh (56.4%) for Case 1 and 131 874 kWh (61.4%) for case 2.

The same strategies were also tested with the floor return temperature of 18℃. For this case more heat is available for GCII. This led to a lower frequency for the high energy modes in all strategies and the total consumption was reduced. How the different strategies performed compared to each other was not changed by the new return temperature, except for strategy 6 in Case 1 that reduced the electric heating so much that the total consumption was marginally less than strategy 7. For strategy 7 the energy consumption was 8.7 % lower for case 1 and 7.5 % for case 2 compared the results with a floor return temperature of 28℃. The total energy saved was 60.2% for Case 1 and 64.3% for case 2 compared to no heat recovery.

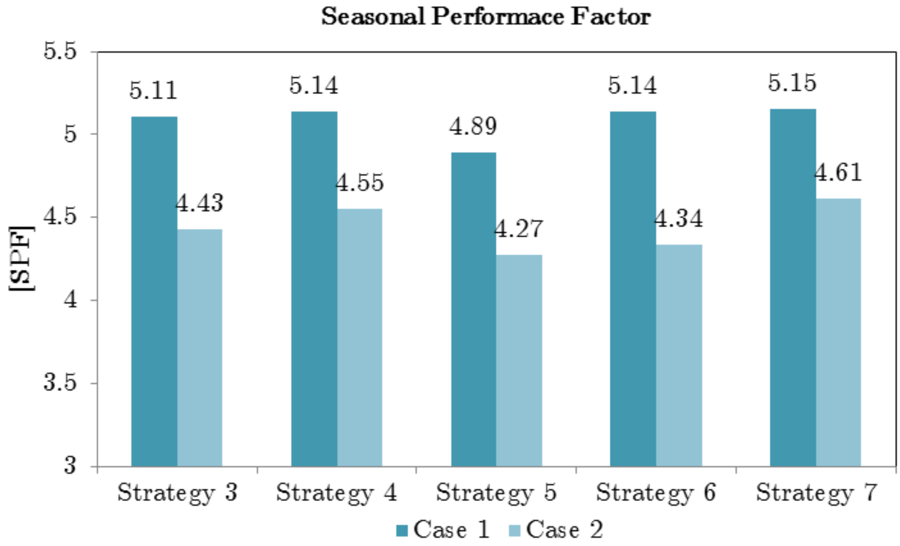

Figure 8-14 SPF of all strategies for each case

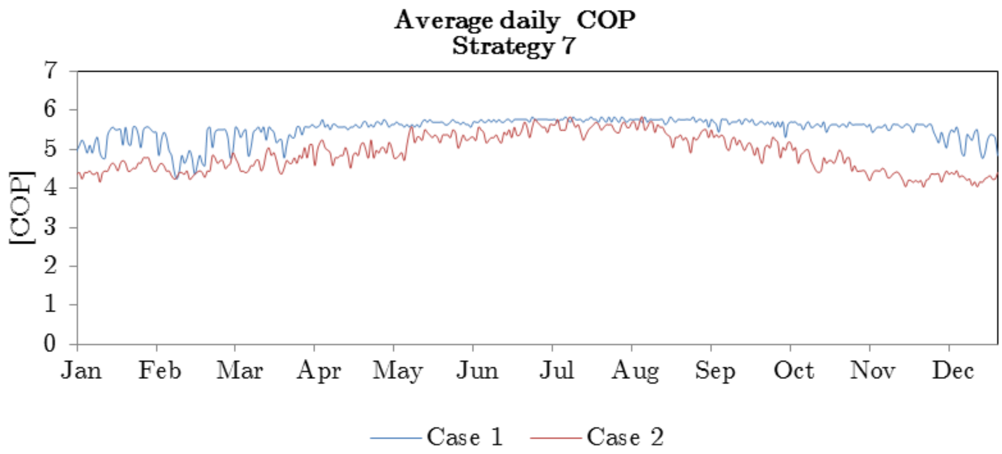

Figure 8-15 Annual COP curves for strategy 7 in both cases

8.5 Conclusion

There is a lot of energy to save by reclaiming heat from the refrigeration plant. Even with the simplest controls not including the accumulation tanks the plant reduced the consumption with 24% in case 1 and 45% in case 2. The plant is capable of providing large amounts of heat when necessary. With the two cases of heat demand the extra evaporator mode was always able to meet the demand which suggests that the system in fact does not have to rely on electric heating. The energy consumption depends on which modes are operated, but the accumulation tanks makes the system more forgiving as surplus heat is never wasted. Different strategies lead to different performance. A higher demand tend to result in lower SPF. Even though the two cases of heat demand were far from equal the same strategies had similar effect on the total consumption. Of the seven strategies tested in this evaluation it was strategy 7 that gave the lowest energy consumption for both cases. Compared to no use of heat recovery the reduction was 56.4% for case 1 and 61.4% for case 2 in the worst case of floor return temperature.

9 DISCUSSION

The results in this paper have been produced with models that uses certain simplifications and assumptions. For instance, the refrigeration plant assumes a near ideal situation for a lot of thermodynamic processes and the very idea of using scenarios is based on simplifications. What kind of impact this has had on the simulation results is discussed in this section along with some other comments and remarks.

Weather Scenarios

The ventilation model was unfortunately not ready for the work of this thesis. Nevertheless, the scenarios created for it are ready, and it will be very interesting to see the estimates of the annual heat demand of the store. For this model it will also be necessary to test different control strategies, and hopefully this will be possible to do within the simulation model as well. The control strategies of the HVAC will have an influence on the strategies for the refrigeration plant which is why joining these two models will be particularly interesting. Perhaps will the ventilation model also one day be ready for whole year simulations. This will remove the need for scenarios completely, and give the most accurate results. The results analysis of the weather scenarios showed that there was a pretty close match between the scenario year and the real/measured year. This means that once the ventilation model is ready the results should be quite reliable. There are however a couple of things that should be discussed.

Temperature

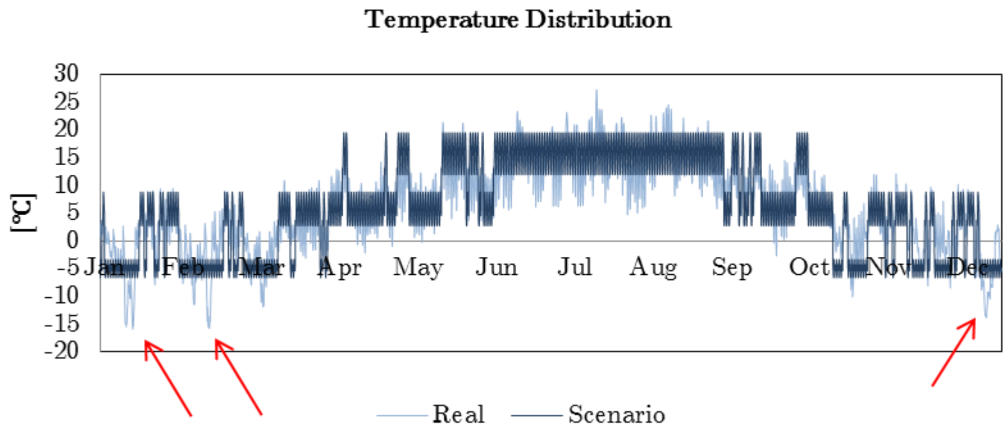

Figure 9-1 Extreme weather situations are neglected/not covered by the scenarios

Looking at Figure 9-1 it is apparent that in some cases the scenarios are quite far off. Temperature was the parameter with the highest average relative deviation at 7.35% over one year. There are some periods where this is more important than others. Because the scenarios of the parameters are based on the average values per hour for each day the scenario-hours become moderated versions of the input. Extreme values that do not occur very often is sort of washed out by the other "normal days". One period in January, one in February and one in December is particularly cold (red arrows). The "Low" scenario only reaches around -7°C and so these periods will not be simulated in the model. This is unfortunate because it is during these periods that the refrigeration plant really has to step up. It was found in the annual estimation analysis, using other sources for heat demand, that under conditions like this the extra evaporator is absolutely necessary to avoid electric heating. This is valuable information and it would therefore be wise to perform simulations for these extreme weather conditions as well. The same can be said for the hot periods in July and August where the free cooling system of the energy well will need to perform. With only three scenarios of temperature these cases are also neglected.

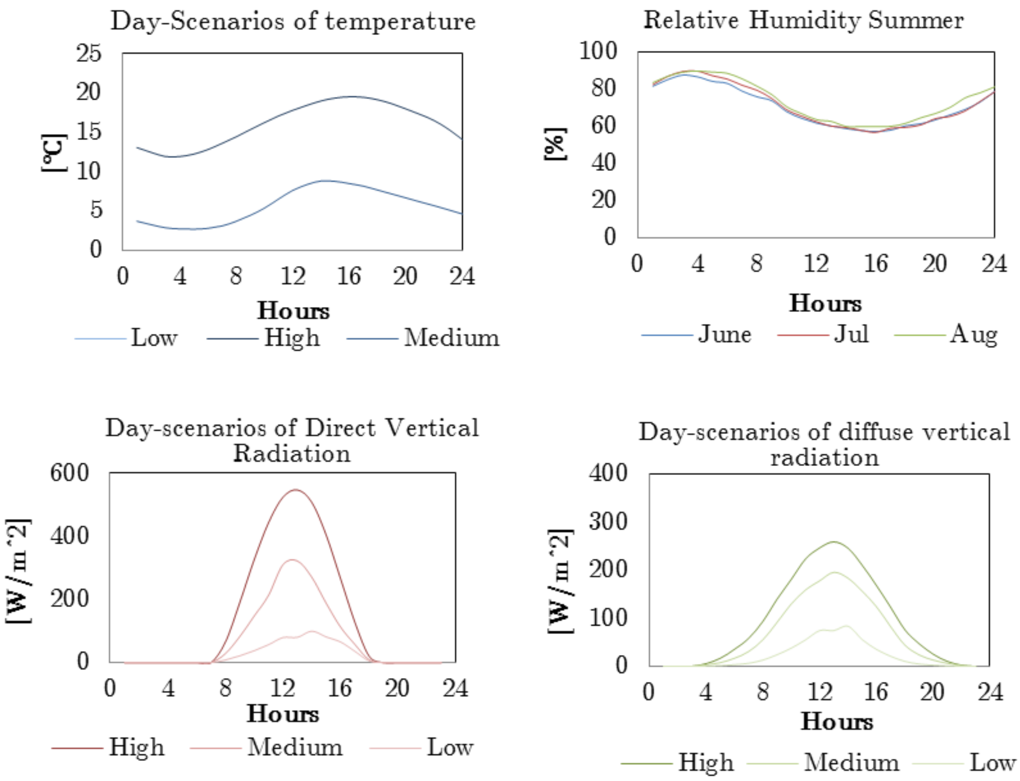

Figure 9-2 Average values smoothens the curves of the scenarios

The daily distribution curves (Figure 9-2) are also very smooth as a result of the averaging. But not all days look like this. In Trondheim the weather can change in a matter of minutes and then change back again moments later. A blue sky is suddenly clouded and the air is filled with rain cooling down the temperature quickly. How the system will respond to changes like this is very interesting, but this will not be experienced in the simulations with the scenarios created. Once again, perhaps extreme situations like this should be simulated as well.

COP curves

One of the tasks that were to be performed was: *"Create curves of available heat and COP for a whole year for each of the combinations of operational mode and return temperature".*

It is desirable to operate the refrigeration plant with the highest possible total COP at all times. This means the combination of refrigeration and heat is delivered with the least amount of work. When only one mode is operated for a whole year and compared to another doing the same it is possible to compare the performance by studying the COP curve. Still, the COP does not always tell the whole truth, and can in fact be misleading. Figure 9-3 shows the COP over the course of one year with strategy 7 and strategy 5 for Case 2. It is clear that strategy 5 has a higher daily average COP than strategy 7. One would then be tempted to select strategy 5 as the best solution. This would however lead to a higher cost as the annual estimation analysis showed (Figure 9-4). The same amount of heat and refrigeration was provided by both strategies.

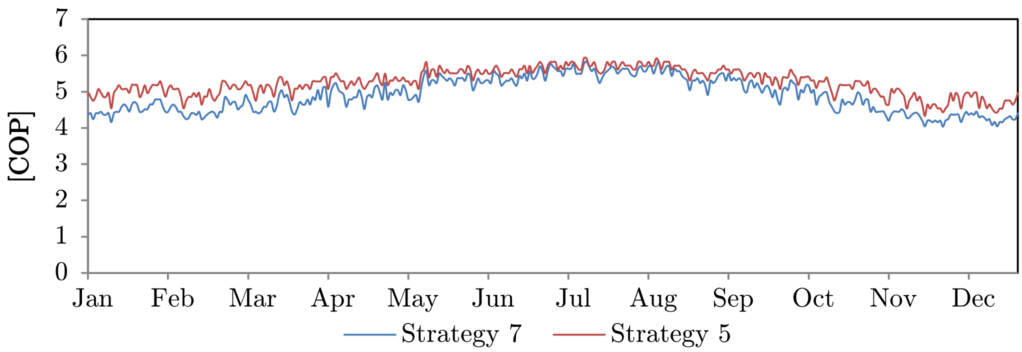

Figure 9-3 COP curves for strategy 5 and 7 from daily averages

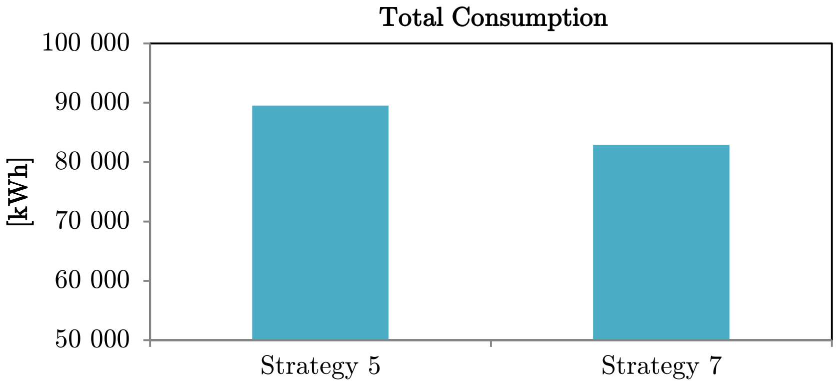

Figure 9-4 Total consumption of the best and worst strategy

This seems confusing and illogical, but can be explained by a simple example: The plant runs for 6 hours with two strategies. The same amount of heat and refrigeration is provided. The data used is from the simulations that where performed.

Strategy A			
Mode	Heat Provided [kW]	Work [kW]	Total COP [-]
EE	61.85	22.45	3.46
EE	61.85	22.45	3.46
STD	3.52	4.45	6.03
STD	3.52	4.45	6.03
STD	3.52	4.45	6.03
STD	3.52	4.45	6.03
Sum	137.78	**62.7**	31.04
Average			**5.17**

Table 9-1 Example strategy 1

Strategy B			
Mode	Heat Provided [kW]	Work [kW]	Total COP [-]
HP	26.73	11.24	4.44
HP	26.73	11.24	4.44
HP	26.73	11.24	4.44
HP	26.73	11.24	4.44
HP	26.73	11.24	4.44
STD	3.52	4.45	6.03
Sum	137.17	**60.65**	28.23
Average			**4.7**

Table 9-2 Example strategy 2

Strategy A has the highest average COP, but the work spend is higher than strategy B. The reason is due to the fact that the work associated with EE mode has a bigger relative impact on the total amount of work than its corresponding COP has on the added COP. In other words, the low COP has less of a negative effect on the average COP than the high work input has, which gives the impression that strategy A is the most effective by looking at the COP curve when in fact the opposite is true. The real efficiency is calculated by dividing the combined heat and refrigeration delivered, with the total amount of work, and so strategy B is the best solution. The difference seems small in this example, but when there is 8760 hours in one year and a plant may operate for 20 years it does matter.

This explains why the curves in Figure 9-3 don't match the results in Figure 9-4, or rather, why they actually do. The conclusion is that COP curves like this can't be used to draw conclusions about the efficiency of a plant or a strategy.

Simulation of the refrigeration plant

One of the main tasks given in this thesis was to simulate three operational modes of the refrigeration plant under varying conditions and analyze the results. The simulations were performed at the technical university of Braunschweig after the varying condition scenarios were received, before the results were sent back to Trondheim. The three modes could not be changed and so all the results from the analysis are affected by the same settings and assumptions made for the different modes.

Standard Mode

The operational mode referred to as "standard mode" in this thesis was a mode representing the case of least compressor consumption i.e. the mode yielding the highest refrigeration COP. It can then be thought of as an option far down on the scale of possible operations while Extra Evaporator mode would be one at the other end. From the annual estimation analysis it was shown that because of the accumulation tanks the standard mode was able to operate for a high percentage of the time, especially in Case 1. The high COP of the mode helped reduce the annual power consumption substantially. However, in real life this is perhaps not very likely. There are indeed some issues with this mode that should be addressed.

Figure 9-5 show the standard cycle in a p-h diagram. As mentioned in the refrigeration plant analysis the energy well (GCIII) has to get rid of all the heat remaining after the CO_2 has exited GCII. This amount is highlighted with the blue line for a floor return temperature of 28 ℃. This corresponds to an effect of 27 kW that must be divided by the four wells.

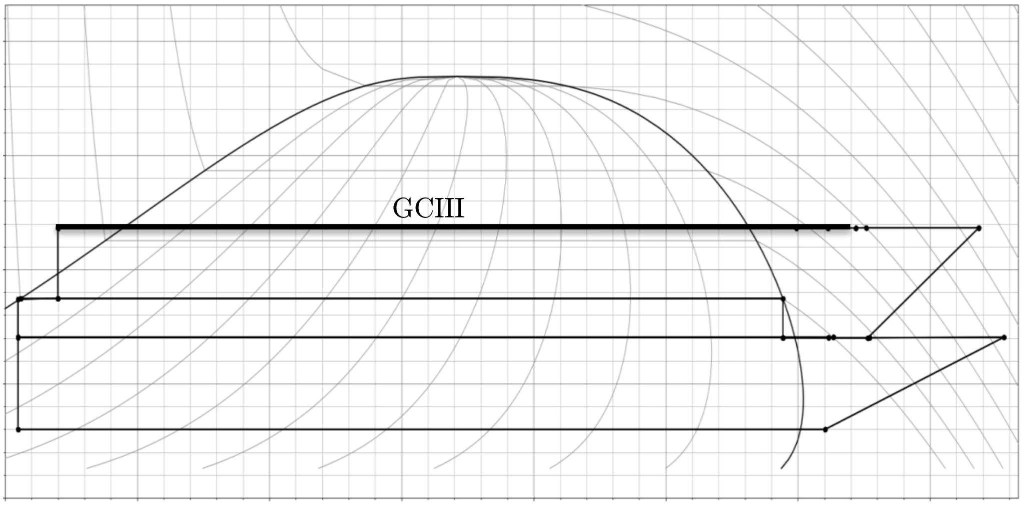

Figure 9-5 Heat rejection to GCIII in standard mode

In the simulation model the glycol is entering the sub-cooler (GCIII) at 5℃ and exiting at 28℃ while the CO_2 does the opposite. This of course is not possible as it would take a heat exchanger with infinitely large surface area. But this is not the biggest assumption. What is of more concern is the fact that the glycol in the simulation is always able to dispose of the heat to the bedrock and return at 5℃. In reality when the glycol is pumped back into the well at 28 ℃ hour after hour it is highly unlikely that it would be able to return at only 5℃. Not only because one needs a temperature difference here as well, but also because the sorounding bedrock temperature would gradually increase. The temperature of the glycol returning to the gas cooler would increase and the refrigeration capacity of the plant would suffer because of higher CO_2 temperature before throttling. CO_2 mass flow rate would have to increase to provide the same refrigeration duty and the energy consumption would be proportionally higher resulting in reduced SPF and higher cost. The “standard mode” can therefore not be used like this for long periods of time in reality which should be considered when looking at the results.

There is however a simple solution to the problem. The standard mode operates at gas cooler pressure of 48 bar. By increasing the pressure ratio slightly the majority of the heat could be transferred through GCII instead. The new gas cooler pressure will of course depend on the floor return temperature. Not only would the energy well by able to sub-cool the CO_2, but the total COP would also be much higher assuming the heat is needed.

Another simulation was performed to investigate this. The MT cabinet load was assumed constant at 18.5 kW and LT at 4.5 kW. The floor return temperature was held constant at 14°C while the pressure was increased from 50 bar to 120 bar. After reaching 120 bar the floor return temperature was then increased to 16°C while the high pressure was reduced gradually back to 50 bar. This was done all the way up to a floor return temperature of 28°C so that an optimum gas cooler pressure giving the highest total COP could be found for each case. Figure 9-6 show the results from the simulation.

At 14°C floor return temperature the optimum pressure was 50.6 bar which gave a total COP of 10.57. The reason for the high value is that at 14°C and 50.6 bar, the full condensation now takes place in GCII while the high stage pressure ratio is still small. Pressure is only increased slightly at 16°C floor return temperature to achieve almost the same performance. As the floor return temperature increases so does the optimum high pressure. At a floor return temperature of 28°C the optimum pressure is at 71 bar and the total COP is in this case 6.28.

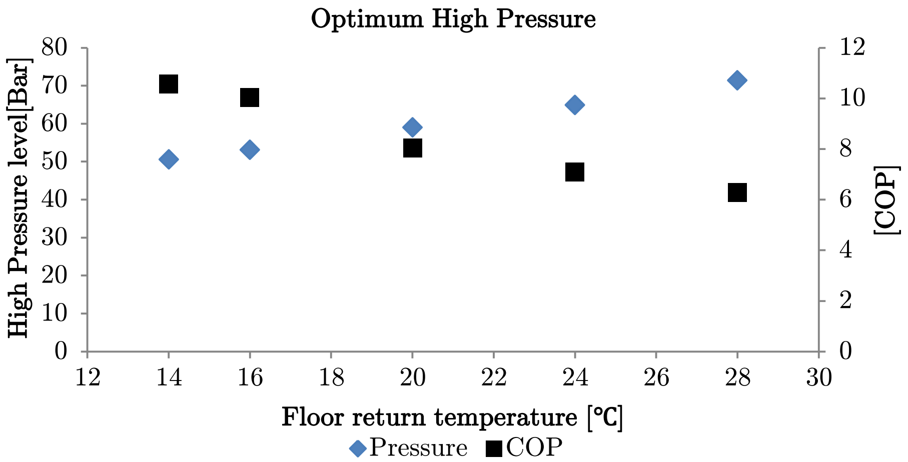

Figure 9-6 Optimum high pressure and corresponding COP for increasing floor rturn temperature

If these modes were used instead of the "standard" mode in the simulation there is reason to believe that the results would be even better. In the two cases of heat demand in the annual estimation analysis the standard mode was almost never able to provide enough heat by itself. The available heat in each case of floor return temperatures from the simulation at optimum COP is displayed in Figure 9-7. Even at 28°C, where the optimum high pressure is 71 bar over 20 kW of heat is available. That could be used instead of HP mode in a lot of situations in the annual estimation analysis and with a higher COP.

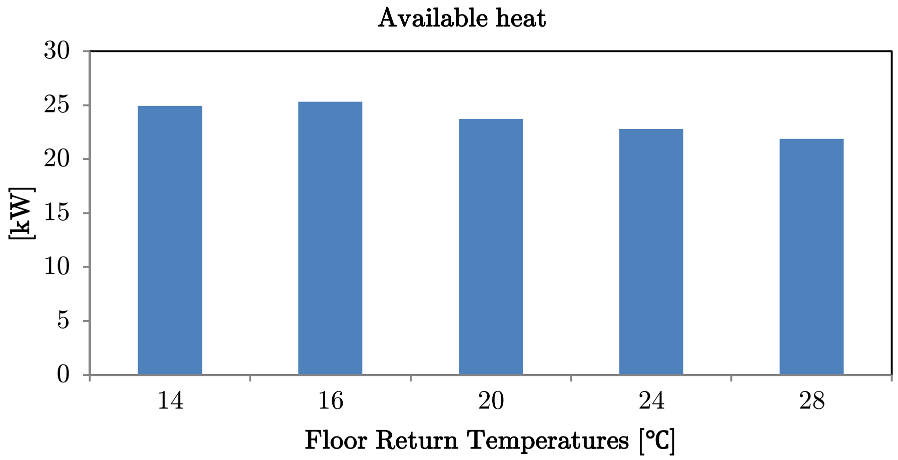

Figure 9-7 Available heat at optimum high pressures for increasing floor return temperature

High Pressure Mode

Standard mode and Extra Evaporator mode represents as mentioned the extremes or boundaries of plant operation. One is well aware of the fact that these aren't modes that necessarily are going to be operated that often in reality, but it gives sense of what is possible and it is therefore quite interesting to look at the results of such cases. HP mode as it is in the simulations is perhaps not that interesting. The point of using the extra evaporator is that the plant will sometimes not be able to provide enough heat with only the cabinets as heat source. The high stage pressure ratio can be raised to a certain point where, if more heat is needed, using the extra evaporator will be most economical. Up until this point increasing the pressure ratio has been the most efficient solution, which means that this is sort of a turning point. It would therefore be interesting to simulate the situation which is just before the extra evaporator has should kick in. This situation is *not* the HP mode used in the simulations. The HP high pressure is at 120 bar which is far beyond the turning point.

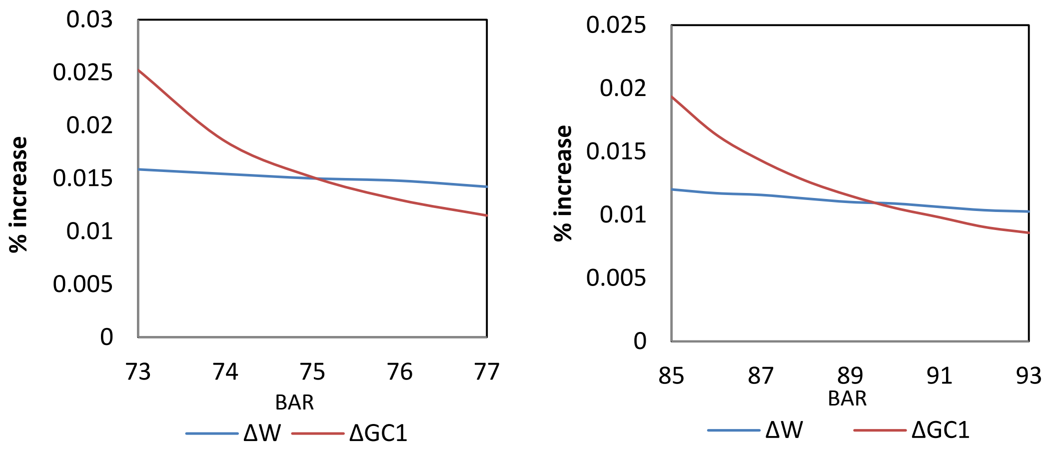

Figure 9-8 Turning points for CO_2 outlet temperature of 30℃ (left) and 35 ℃ (right).

These turning points were investigated in the preceding project work of this thesis. With a CO_2 outlet temperature of 35℃ from GCII the turning point was found to be at about 90 bar. If the temperature was 30℃ then the point is around 75 bar. It is perhaps these situations that should be simulated instead. The reason there exists such turning points can be understood from the shape of the isotherms. When the isotherms are relatively flat (horizontal) then a pressure increase will result in a big enthalpy difference. As the pressure is further raised the isotherms become steeper and steeper until the incremental gain in enthalpy difference across the compressor is larger than the incremental enthalpy gain in the heat exchanger. From this point on an increased mass flow rate will be more efficient which the extra evaporator will provide.

Another issue with the HP mode is of course that it operates at the highest possible pressure ratio (so does EE mode) which over time would be quite stressful for the compressors. The only special thing with HP mode is that it represents the mode which gives the most heat without the use of an extra evaporator. As long as this extra evaporator works there will be no circumstances under which one would want to use HP mode as it was defined in these simulations because it would be less economical.

Extra Evaporator Mode
This represents the maximum heat possible to provide from the plant. In the simulations it was assumed that 30 kW was provided by the evaporator. Depending on what kind of bedrock we have and what the temperature difference is, the heat effect from a well can vary between 20 and 80 W/m [20]. If we assume that it is 20 W/m and that this is valid for the whole length of the collector tubes then we can estimate the effect. There are four wells each 170 m deep and heat transfer occurs as the fluid travels up and down

$$\dot{Q} = 170m \times 4 \times 2 \times 20 \frac{W}{m} = 27.2\ kW \qquad (9)$$

This shows that the assumption of 30 kW seems quite reasonable, but the problem is when the mode is operated for a longer period of time. It is the same situation as with the sub cooling in standard mode only now the return temperature of the glycol will become too *low* instead. The CO_2 is at -8°C during the evaporation and this becomes the temperature of the glycol since perfect heat transfer is assumed. The surrounding temperature decreases as cold glycol is continuously pumped back into the well and it won't be possible to get the 30 kW after a while.

Another issue with extra evaporator mode is that the increased pressure drop has not been considered in the simulation model. CO_2 systems have low pressure drops compared to other systems due to high vapour density as discussed in the literature study. Nevertheless, it is something that will affect the performance.

EE mode was also tested at standard pressure, but because of the already large amount of heat going to GCIII the cycle broke down. There is of course no point in using the extra evaporator mode at standard pressure, but it shows that the cycle relies on the ability to cool down the CO_2 in order to function properly.

Pressure Drop
When the extra evaporator is used the mass flow of refrigerant will increase to provide the same refrigeration to the cabinets. Figure 9-9 show the difference in mass flow between standard/HP and EE mode. The mass flow increased from an average of 0.091 kg/s to 0.22 kg/s which is 2.4 times higher.

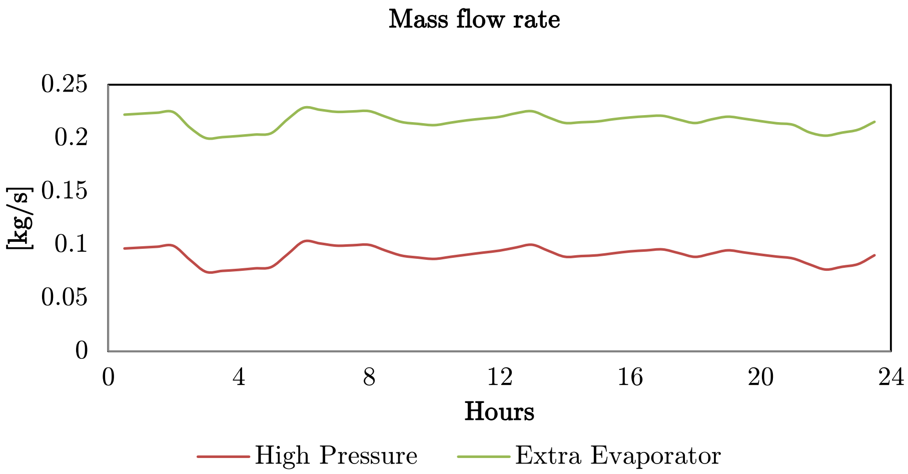

Figure 9-9 Mass flow rate of refrigerant in HP and EE mode

The mass flow rate is defined as

$$\dot{m} = \rho \times A \times u \tag{10}$$

And the pressure drop per meter is

$$\Delta p = \lambda \times \frac{1}{D} \times \frac{\rho \times u^2}{2}, \quad \left[\frac{Pa}{m}\right] \tag{11}$$

The pipe friction factor λ for *turbulent* flow provided by Colebrook

$$\frac{1}{\sqrt{\lambda}} = -2 \times log\left[\frac{\epsilon/D}{3.7} + \frac{2.51}{Re \times \sqrt{\lambda}}\right] \tag{12}$$

And for *laminar* flow

$$\lambda = \frac{64}{Re} \tag{13}$$

$$Re = \frac{\rho u D}{\mu} \tag{14}$$

If we for the sake of simplicity assume laminar flow for both modes then the pressure drop is found as a function of mass flow rate by inserting (10), (13) and (14) into (11)

$$\Delta p = 32 \times \frac{\dot{m} \times \mu}{\rho \times A \times D^2}, \quad \left[\frac{Pa}{m}\right] \tag{15}$$

Going from HP to EE mode will then increase the pressure drop proportionally to the mass flow rate. It will in other words be 2.4 times higher assuming the other properties stay constant.

Heat network

Even though the results from the extra evaporator mode are perhaps a bit too optimistic it seems clear that it is capable of providing more heat than necessary in many situations. An interesting thought would be to integrate the heat system of the supermarket with nearby domestic buildings. This supermarket is a pilot project and as most pilot projects it is much more expensive than the usual supermarket to build. A goal is that this supermarket will prove not only environmentally friendly but also very efficient so that it can be the new standard for new supermarkets. The simulation results showed that if the store is to operate without the need of electrical heating it has to use the extra evaporator. That means that energy wells or a similar substitute is necessary. Wells are expensive and it can be difficult to have new supermarkets investing in these. The solution can perhaps be a heat network. The requirements for annual space heating demand of a normal Norwegian home is 80 kWh/m^2 according to the Norwegian building codes of 2007 (TEK2007). For a low energy house it is 58 kWh/m^2. The domestic hot water heating is 25-30 kWh/m^2 for both cases. [21]. If we assume that the extra evaporator could be used all year as it was simulated, then with the highest heat demand (Case 2) the extra heat produced was 341 431 kWh/year. This could in theory provide space heating for 28 normal homes of 150 m^2 or 39 Low-Energy houses. Even if the extra heat was only a fourth of this it would still be enough for 10 low-energy houses. 78 % of the energy consumption in Norwegian homes is electric energy [22]. The COP of the extra evaporator mode is 3.5 and so heating of these houses would be more efficient than with electricity. This could perhaps be an incentive for government support on the refrigeration system and drilling of wells, or the supermarket could in some way "sell" the extra heat.

Such an arrangement would be easiest to implement if both the supermarket and an apartment block were built together. Since the plant would have to operate a lot as a heat pump it would be important with thermal charging of the well. One solution could be to use wastewater from the apartments which would then be heat reclaim of DHW.

Compressor Efficiency

Another assumption in the simulation model is a constant isentropic and volumetric efficiency of 70%. The efficiency of a compressor is related to the pressure ratio. 70 % is an ok estimation at standard mode where the pressure ratio is just below 2, but in HP and EE mode the ratio is 4.3 and one must expect efficiencies around and below 60%. This leads to higher compressor energy consumption and a lower COP as well as increased discharge temperatures. For HP mode in the discharge temperature is already at 150 °C according to the simulation results.

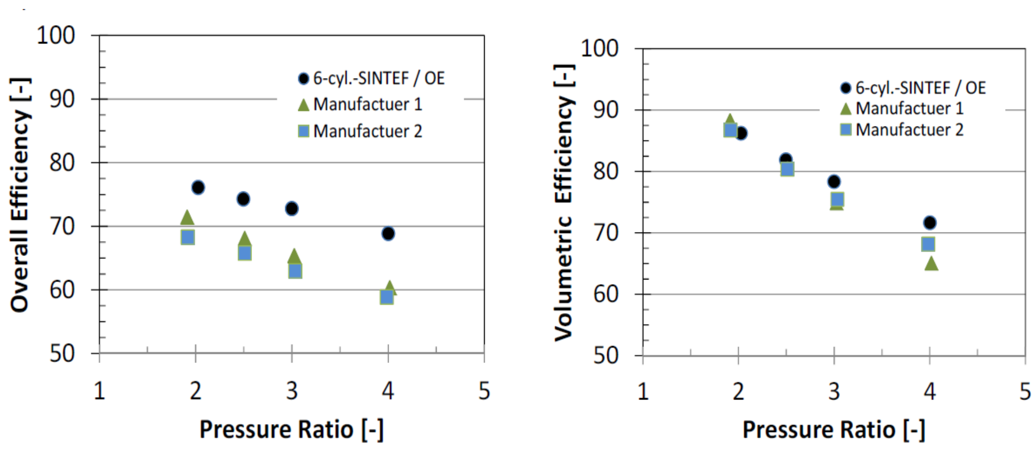

Figure 9-10 Compressor efficiencies at varying pressure ratio [23]

COP

The total COP was defined as

$$Total\ COP \quad = \quad \frac{Q_{LT} + Q_{MT} + Q_{GCI} + Q_{GCII}}{W_{LP} + W_{HP}} \tag{16}$$

One of the things that ensure efficient operation of the refrigeration plant is the sub-cooling of the CO_2 by the energy well. This temperature was set to 5°C. It is more realistic that the CO_2 will be cooled down about 10°C since there must be some a temperature difference GCIII and the fact that the bedrock can be at temperatures of up to 8 °C. However, this is all part of the situation being looked upon as ideal. What perhaps is more objectionable is the fact that the use of the energy well seems free. In reality there is a pump working to circulate the glycol providing cooling. The same goes for the situation where the energy well operates as an extra evaporator. The power input of the pump should also be included in the calculation of the total COP since the sub-cooling also comes with a cost. There is also a pump circulating the glycol in the tank and the water in the floor heating circuit. This is all energy consumption associated with delivering the heat from the plant. If the efficiency is to be compared with alternative electric heating with COP=1, then all parts of the system should be included in the equation. It is therefore suggested that the operation of these pumps should be included in the simulation model. Whether or not fans for the ventilation system should be included is debatable since they would have to operate regardless of how the heat is provided.

$$Total\ COP = \frac{Q_{LT} + Q_{MT} + Q_{GCI} + Q_{GCII}}{W_{LP} + W_{HP} + W_{pumps}} \qquad (17)$$

Accumulation Tanks

The results from the annual estimation analysis show that the accumulation tanks make a huge difference for the total energy consumption. Storing of surplus heat allow high COP modes solely for refrigeration to operate for long periods. The properties of the tanks were simplified a great deal in the analysis. As for the energy well, the tank is not simulated in the refrigeration model. The model only uses fixed boundaries such as a constant return temperature of 30°C. In the actual supermarket that is currently under construction the situation will be more complicated. First of all the accumulation tanks function as a link between the refrigeration plant and the ventilation system. These two only communicate through the tanks. From the HVAC point of view the tanks are just a heat reservoir which is available at all times. The refrigeration plant operates according to temperature signals from the tanks, always making sure that they don't fall under a specific level. There are three tanks connected in series. Each tank will have ha temperature distribution with the highest temperature in the top of the tank and the lowest in the bottom. The average temperature of the first tank will be higher than the second which again is higher than the third, when they are connected as depicted in Figure 9-11.

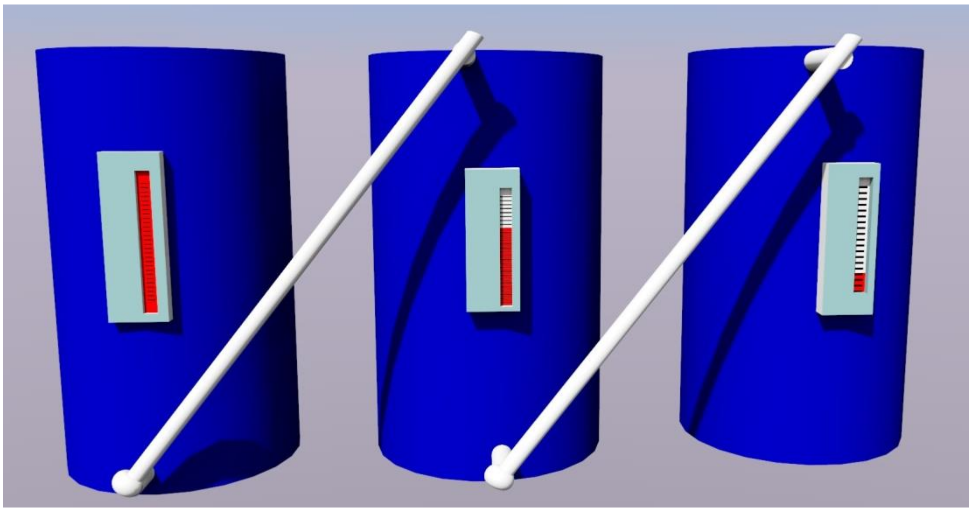

Figure 9-11 Model of accumulation tanks connected in series

The hot glycol exiting GCI enters the top of the first tank. As the temperature increases in the tanks the return temperature will also increase making it harder to transfer more heat unless the glycol is cooled somehow. The glycol loop can exchange heat with an air curtain at the entrance, or provide ground heat at the HSE area. There is also the possibility of dumping the heat through dry coolers when there is no heat demand and the CO_2 just need cooling to maintain an efficient refrigeration cycle. The temperature available from the tank will influence the necessary mass flow rate of glycol to provide the required effect to the heating coil in the ventilation system. There are in other words a lot that can affect the state of the tanks which is neglected in the simulations. Once the ventilation model is ready it will be important to join the two models with a third model simulating the behaviour of the accumulation tanks.

The big picture

There are roughly 4000 supermarkets in Norway according to a number of sources [24] [25] [26]. The average supermarket energy consumption according to the statistics of Enova was 514 kWh/m^2 in 2008 [27]. This number was based on the measurments from 204 stores with temperature and location corrections. Heating and refrigeration account for around 50 % of the total consumption of a supermarket. The average REMA 1000 store was 800 m^2 in 2007 [28]. If we assume that these numbers are representative for the country and that most supermarkets have little or no heat recovery we can look at the benefit of implementing the new supermarket system on a national level. The annual estimation analysis using the simulation results showed that we can save between 50 and 60 % by using the new system compared to a system with no heat recovery.

$$National\ consumption\ in\ heating\ and\ refrigeration =$$
$$50\% \times 514 \frac{kWh}{m^2} \times 800\ m^2 \times 4000\ stores = 822\ 400\ 000\ kWh$$

If the new system could reduce this amount with 50%:

$$Energy\ saved = 411\ 200\ 000\ kWh$$

The average European CO_2 release is approximately 0.6 kg per kWh of electrical energy produced [6].

$$CO_2\ emission\ reduced = \ 0.6 \frac{kg}{kWh} \times 411\ 200\ 000\ kWh$$
$$= 246.7\ million\ tonnes\ CO_2$$

This is an incredible amount, but of course a very rough estimate. Because of the hydro power available in Norway the emission rate of 0.6 kg/kWh is actually 0.0kg/kWh, so the emission reduction is not correct for Norway spesifiaclly. Denmark on the other hand, a country of similar size in population, the number is approximately 0.84 kg/kWh [1]. There are only about 5 million living in Norway and Denmark, but 750 million for Europe in total. Although these two countries are not representative for the rest of Europe it is without any doubt that we can say these smart heat recovery systems have huge economical and ecological potential. This rough estimate was also not including the benefit of using CO_2 as a refrigerant opposed to the HFC's with thousands of times stronger impact on global warming. This is therefore an very interesting field for the future and how much can be accomplished will to a large extent probably rely on the political initiative to foster smart and ecological systems. There most certainly should not be a lack of incentive.

10 CONCLUSION

Supermarkets are energy intensive buildings responsible for large amounts of greenhouse gas emissions. The last couple of years have led to an increased interest in natural refrigerants with low environmental impact. The literature study shows that CO_2 based refrigeration plants and heat pumps can operate with high efficiency under the right conditions. A new supermarket tailor made for a CO_2 booster system with heat recovery has been designed by SINTEF and is currently under construction.

A program was created in Excel to construct scenarios describing the ambient conditions in Trondheim over one year for a ventilation model of the new supermarket. The program organizes the scenarios in such a way that they represent a base year with high accuracy. The relative deviation analysis shows that the program was able to create a scenario year for the weather in Trondheim where the parameters temperature, direct and diffuse vertical radiation deviated only by 7.35%, 3.84% and 5.71% respectively from the base year. The maximum simulation work was reduced by 92.6%.

Another set of scenarios was created for an ideal simulation model of the refrigeration plant at the new supermarket to investigate the energy consumption and effectiveness at varying conditions and control strategies. Simulation results suggest that there is a lot of energy to save by reclaiming heat from the refrigeration plant. Annual estimations of the plant tested against two cases of heat demand shows that even with the simplest control strategy, not including the accumulation tanks, the consumption is reduced by 24% in case 1 and 45% in case 2. Results also suggest that the plant is capable of providing large amounts of heat when necessary by use of the extra evaporator, eliminating the need of an additional electric heater. The energy consumption depends on how the refrigeration plant is operated, but the accumulation tanks makes the system more forgiving as surplus heat is never wasted. Of the seven strategies tested in this thesis the same strategy gave the lowest energy consumption for both cases. Compared to no use of heat recovery the reduction was 56.4% for case 1 and 61.4% for case 2 in the worst case scenario. Results also show that the energy savings are only 3% better at the most favourable case of floor return temperature due to the fact that the total heat reclaimed is dominated by GCI. The models should be further improved to reduce the number of simplifications and assumptions of ideal thermodynamic processes to enhance the credibility of the results.

11 PROPOSAL FOR FURTHER WORK

The refrigeration model was able to produce valuable data about the performance and possibilities of the heat recovery system at Kroppanmarka. The modes given for this thesis to evaluate can be regarded as extremes and so the results represent the boundaries of the system. As mentioned in the discussion section it would be interesting to also simulate the point in which the pressure is increased no further and the extra evaporator should start to engage, referred to as the turning point. There are also several aspects of the model itself than can be improved. A more advanced model of the energy well that not only includes the temperature changes as function of cooling at heating demand, but also the associated pump work should be developed.

When the ventilation is ready the scenarios created can be implemented and these results must be evaluated. Simulations should also be performed for the extreme weather situations that have not yet been included in the scenarios. Both the refrigeration and ventilation model will then have interesting results, but a supermarket is a complicated building with many interactive sub-systems and evaluating them separately will overlook important aspects. A third accumulation tank model should be developed to unite the ventilation and refrigeration model since this is the link between the two systems. Testing different control strategies will be important once this is done.

Some of the more non-technical aspects of the new supermarket should perhaps also be investigated. Is it possible to create a heat sharing network with low energy buildings and homes? Could the supermarket share and sell energy so that the repayment period of the investment costs are reduced making these environmentally friendly solutions more attractive for investors?

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12 APPENDIX

A. Draft for a scientific paper

Introduction

Supermarkets are one of the most energy intensive buildings in the commercial sector. In industrialized countries the power consumption represents 3-5% of the total annual consumption [1]. With huge cooling demands for food storage the industry applies great amounts of refrigerant. It is estimated that supermarkets are responsible for 28% of the world's refrigerant consumption [29]. Annual average leakage is in the range of 15-30% [3], with Europe being in the lower part of the range. The most common refrigerants in Europe are the HFC gases R134a and R404a, both of which have a high global warming potential [4]. In countries like Norway and Denmark HFC gases have a CO_2 equivalence tax of 0.225 kr/kg [5]. Climate change, environmental damage and depletion of natural resources followed by more laws and regulations has forced engineers to come up with smart solutions for supermarkets that are both energy efficient and environmentally friendly. This has increased interest for natural working fluids. CO_2 is a safe and green refrigerant with special properties that can be utilized in refrigeration and heat recovery. A new supermarket is being built in Trondheim in collaboration with SINTEF and their CREATIV project. A transcritical CO_2 booster system has been selected to provide both refrigeration and heating for the whole store. The complete system is carefully designed to optimize efficiency and reduce environmental damage.

CO_2 in Supermarkets

By the 1950s CO_2 had almost completely disappeared from refrigeration mainly due its high operational pressure and low critical temperature being unfavourable for the usual applications. Now, because stricter regulations for HFCs combined with regained focus on CO_2 as refrigerant, improved equipment has been manufactured specifically designed for its unique properties. This has enhanced the positive sides and moderated the negative making it an increasingly competitive alternative to HFC systems.

High operating pressure leads to high volumetric heating/cooling capacity and a smaller required compressor volume. It also means higher density which result in lower fluid velocity and therefore small pressure losses. The saturation pressure-temperature curve is also very steep which leads to even lower temperature drop per pressure drop. Required sizes for tubes and components are smaller and so material costs and radiation losses are also reduced. The minimization of tubes and components also leads to easier installation of cabinets and

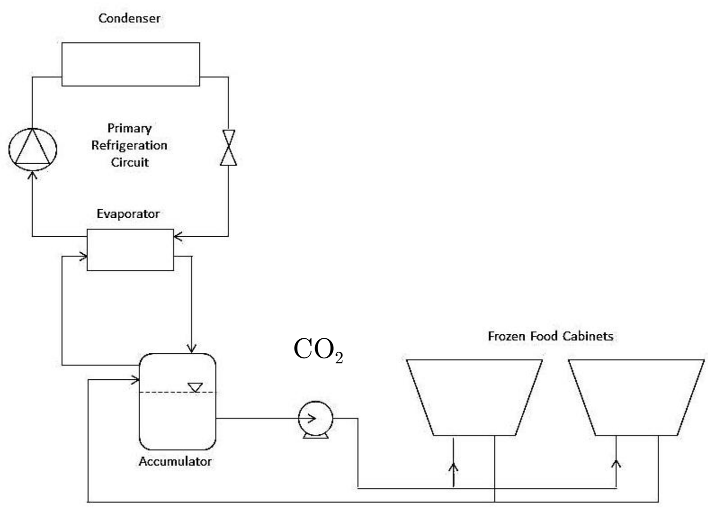


Fig. 1

lower refrigerant charge [12]. With high operational pressure the compressor pressure ratio is small resulting in high compressor effectiveness [13] .The first commercial applications of CO_2 were in indirect systems for freezing temperatures(Fig. 1). The primary refrigerant circuit, which is typically R404A, evaporates by drawing heat from the CO_2 returning from the freezing cabinets via an accumulating vessel. At these temperatures the working pressure of the CO_2 is quite reasonable (11 bar at -37℃). Another system where CO_2 is becoming more competitive is in the cascade solution. CO_2 is used in the low temperature stage, but can also do the job in the medium temperature. This way it is possible to provide medium and low temperatures using only one heat exchanger.

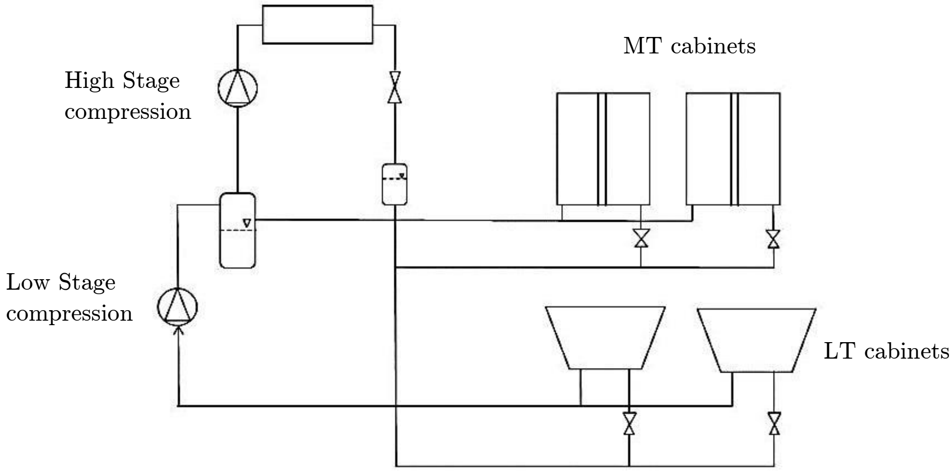


Fig. 2

A third option is the so called booster system (Fig. 2). In this case CO_2 is the only refrigerant and so this is an attractive solution from an environmental point of view. Double compression and double expansion assures two evaporation levels for medium and low temperature applications. The discharge pressure of the low compression stage matches the pressure in the cabinets at the medium temperature level and so both levels joins before the high stage compression. The high stage working pressure in a Norwegian climate will usually be between 40 and 60 bar under normal operation which is much higher than it would be if R404A was used. For this reason it can be difficult for the CO_2 booster system to achieve as high values of COP as conventional working fluids. Although this is a widely applied system in supermarkets it has been mainly limited to countries with colder climate due to the importance of cooling down the CO_2 and reducing the high stage working pressure.

The fact that supermarkets in colder climates often require substantial heating to the building as well as cooling for the sales products creates an opportunity for integrating systems. The idea of using some of the condensation heat from the refrigeration plant to cover the heat demand of the building has the potential to reduce the energy consumption considerably [29]. CO_2 is especially suited for heating of water because of excellent heat transfer capabilities and great temperature fit in transcritical operation [13]. The CO_2 is cooled rather than condensed which means that heat rejection occurs at gliding temperature. Supermarkets using hydronic heating systems are therefore good candidates for CO_2 refrigeration plants. Such a system has been investigated in this study and is presented in the following sections.

Case Study Kroppanmarka

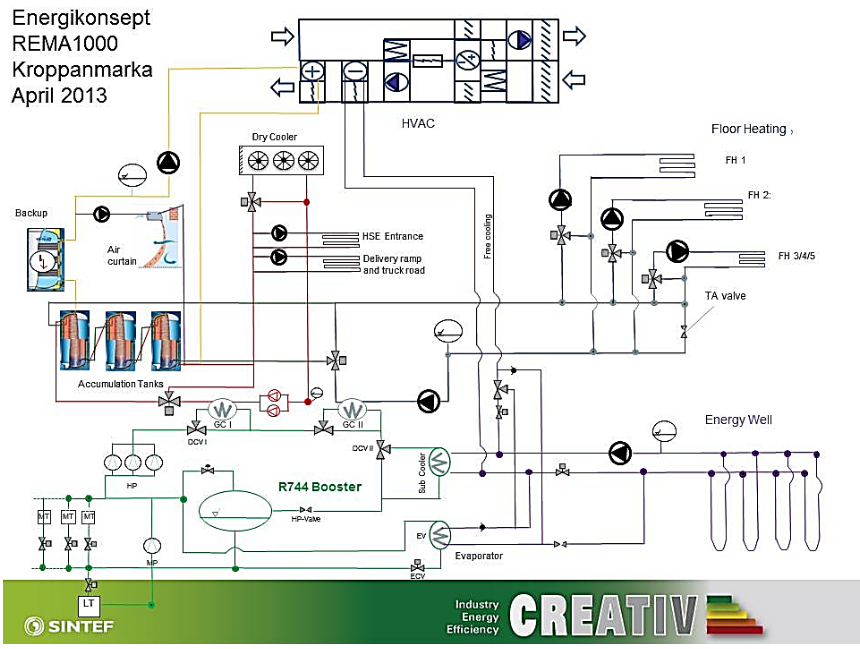


Fig. 3

The franchise REMA 1000 is building a new supermarket in collaboration with SINTEF and their CREATIV project. There are two special things about this supermarket. The first is that the whole building is built from scratch and the sub-systems are carefully designed to match one another. A challenge with most supermarkets is that there is a conflict of interest between the merchant, the franchise and the owner of the building. The ventilation system is the responsibility of the owner and as long as rules and regulations are followed he will most likely buy the cheapest one he can get. The franchise on the other hand is responsible for the refrigeration plant, and investment costs are also a decisive factor here. As a result, integration between the two systems may prove difficult due to the fact that they were never meant for each other in the first place. Finally the merchant is left with the electrical bill and a relatively inefficient system which also have a bad effect on the environment. With this new project this challenge will not be an issue. The second special thing about the supermarket is that it uses a refrigeration system solely on CO_2 and aims to use heat recovery to meet the whole heating demand. Simulations of the refrigeration plant have been performed to investigate the available heat from the plant and the associated COP at various operational modes and external conditions. Weather scenarios representing a whole year has also been created for a model being developed to simulate the behaviour and heat demand of the ventilation system at the store.

System Description

Fig. 3 shows the entire principle for heating, cooling and refrigeration in the supermarket. This is a highly integrated system consisting of 5 main parts.

- Refrigeration plant
- Accumulation Tanks
- Floor heating system
- HVAC
- Energy Well

The heart of the system is the refrigeration plant (green circuit in Fig. 3). It is a transcritical R744 booster system delivering both medium and freezing temperatures to cabinets and cold rooms. Smart control of the operation is crucial as this is the only source of heat for the building.

Heat demand and heat availability does not necessarily coincide which is why the system uses heat accumulation tanks to store the heat from the first gas cooler (GCI) to be utilized when a demand arises. The heat is then brought trough a heating coil in the ventilation system which feed the store with warm air. To maintain good air quality fresh air needs to enter the building. This air must be heated or cooled to the desired indoor temperature.

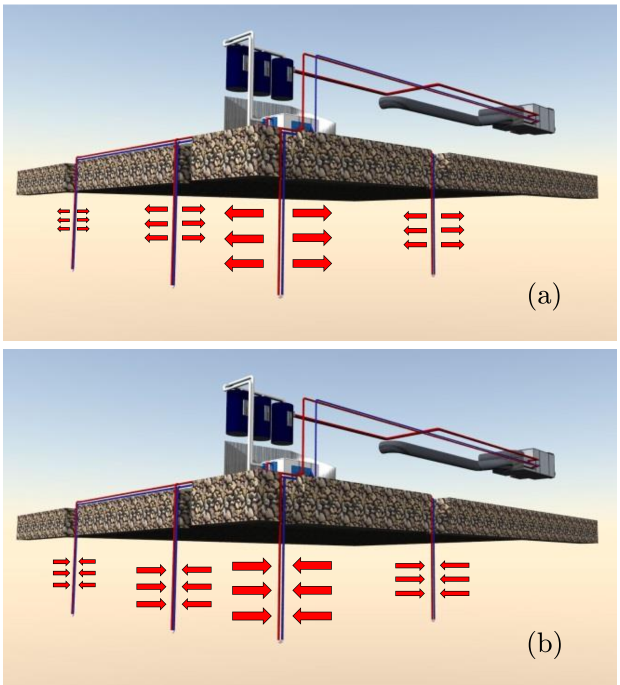


Fig. 4 The energy well can provide both cooling (a) and heating (b) to the supermarket

Free cooling is available from the energy well at no other cost than the associated pump work (Fig. 4). Four wells can provide the necessary cooling independent of ambient temperature. The wells are also there to feed an extra evaporator for the refrigeration plant in the case of increasing heat demand. Sub-cooling for the refrigeration cycle is also provided by the well. Heat is reclaimed to a hydronic floor heating system through the second gas cooler (GCII) which is connected in series after GCI. Water is heated in GCII and will be circulating in tubes under the concrete floor of the store. In case of an additional floor heat demand a spiral heat exchanger in the accumulation tanks can deliver more effect. There are four different loops in the store with individual control to maximize efficiency and comfort.

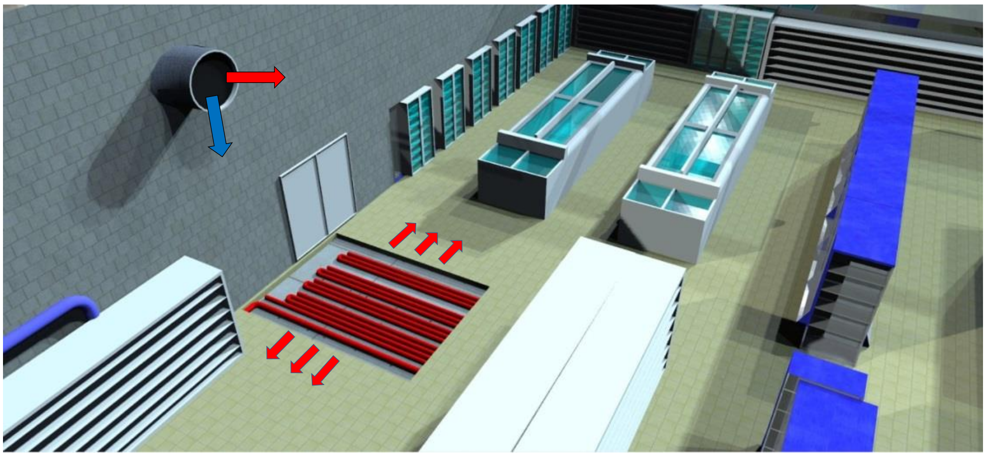


Fig. 5

Data for the ventilation model

Because of the important role of ambient conditions weather data for a year in Trondheim was downloaded using metenorm7. The data is in the form of hourly values for a number of parameters over the course of one year based on measurements from a weather station at Værnes. The simulation program performs 24 hour simulations and so in order to obtain annual estimations a set of day-scenarios were created to represent a whole year. This was done to reduce the amount of simulations as one in principle should simulate all 365 days. A program in excel was created to produce the scenarios so that the full year of scenarios would match the real one as accurately as possible while holding the number of scenarios to a minimum. The parameters for the simulation model were temperature, relative humidity and direct- and diffuse vertical radiation. 27 combinations of scenarios were created using the excel program. Only one case of relative humidity was used while the parameters; temperature and direct- and diffuse vertical radiation came in three variations. The scenario year was verified by comparing each of the parameters to the real data.

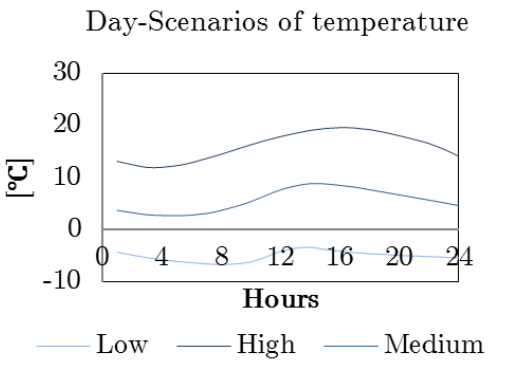
Results

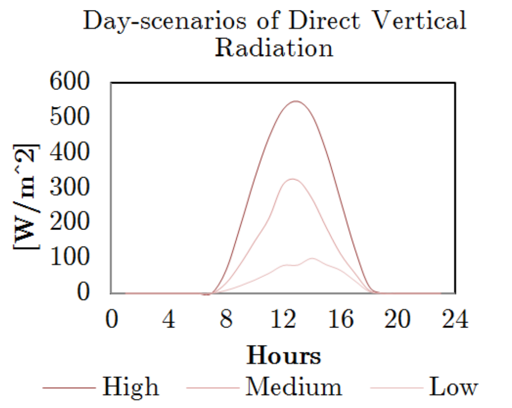
The day-scenarios of each parameter are shown to the right and their distribution over one year is in Fig. 6 - Fig. 8 where they are also compared to the actual year they are representing. From these results the excel program seemed to work very well which was confirmed by the results from the relative deviation analysis where each day of the scenario year was compared to the actual days.

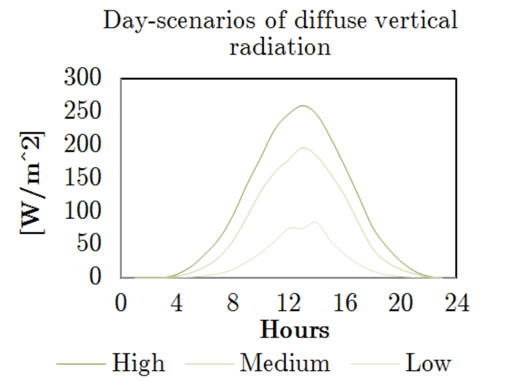
$$Relative\ deviation = \frac{|X_S - X_R|}{X_{Rmax} - X_{Rmin}}$$

The relative deviation was defined as the difference between the value for the scenario and value for the real case at each hour, divided by the total range of the real values within the parameter. Temperature was the parameter most difficult to cover with only three scenarios. The annual relative deviation was 7.35%, while the direct vertical radiation had a deviation of only 3.84% on average. The excel scenario creator program can be used for any type of climate so that simulations for supermarkets in other parts of the world are possible as long as data such as the ones from Meteonorm is provided.

Annual Relative Deviation	
Temperature	0.0735
Direct Vertical Radiation	0.0384
Diffuse Vertical Radiation	0.0571







Simulations of the ventilation behaviour and heat demand can begin as soon as the model is completed.

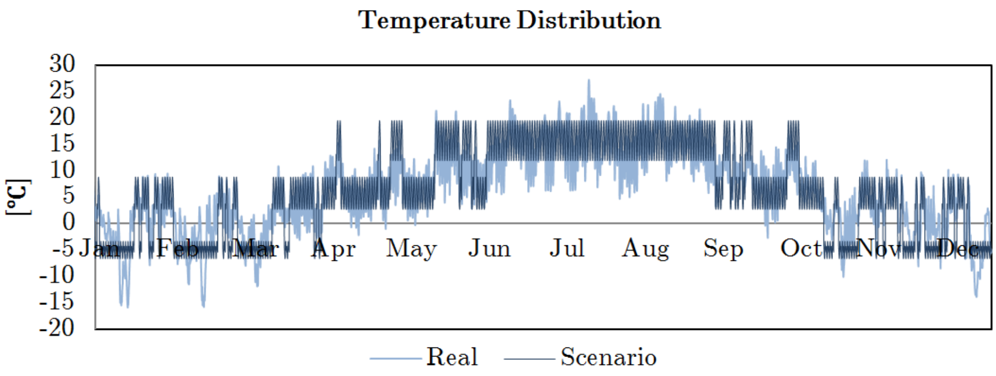


Fig. 6

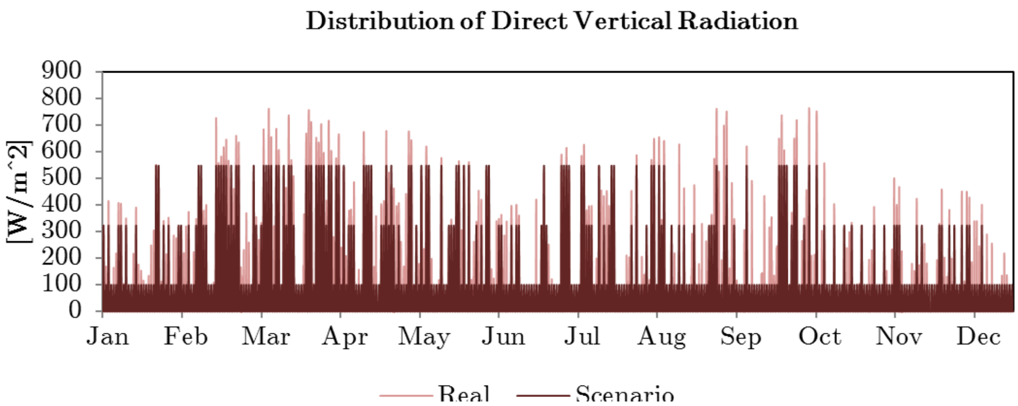


Fig. 7

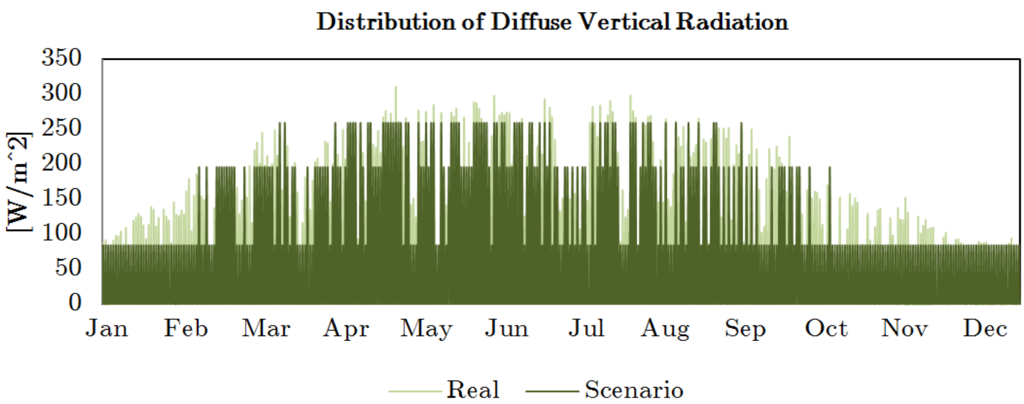


Fig. 8

Simulations of the Refrigeration Plant

The same scenarios for temperature were used in the refrigeration model along with cabinet heat load data based on measurements from another supermarket with a similar refrigeration system and size. Three cases of refrigeration capacity were considered; one for winter, one for summer and one for spring/fall. These are related to the ambient temperature.

Variations of the variables			
Refrigeration	Temperature	Operational Mode	Floor Return Temperature
Winter	Cold	Standard	18℃
Summer	Medium	High Pressure (HP)	23℃
Spring/Fall	Hot	Extra Evaporator	28℃

Table: 1

Three operational modes for the refrigeration plant were used

- *Standard Mode:*

This operational mode is that mode that yields the highest COP for the refrigeration plant. Condensation takes place at 12 ℃ and 48 bar.

- *High Pressure Mode:*

This mode refers to the case where the gas cooler pressure is raised to 120 bar which is the maximum high pressure possible. The operation is now transcritical and heat rejection occurs at gliding temperature.

- *Extra evaporator Mode*:

The energy well is used as an extra evaporator (30 KW) to increase the heat to the gas coolers. This is done at 120 bar and therefore represents a maximum of what can be produced by the plant.

The amount of heat transferred through the second gas cooler (GCII) depends on the inlet temperature of the water returning from the floor heating circuit. The lower the temperature is the bigger the enthalpy difference Δh across the heat exchanger becomes and the more heat is transferred. Since this return temperature is an unknown 3 cases at; 18, 23 and 28 ℃ was tested to investigate the outcome. The heat available for GCI is independent of the floor return temperature. The return temperature from the tanks was assumed constant at 30℃. The simulation model produces values for the available heat for both gas coolers and the total COP of the plant defined as

$$Total\ COP \quad = \quad \frac{Q_{LT} + Q_{MT} + Q_{GCI} + Q_{GCII}}{W_{LP} + W_{HP}} \tag{18}$$

Results

The simulation results for the winter case is presented in Fig. 9 to Fig. 11.

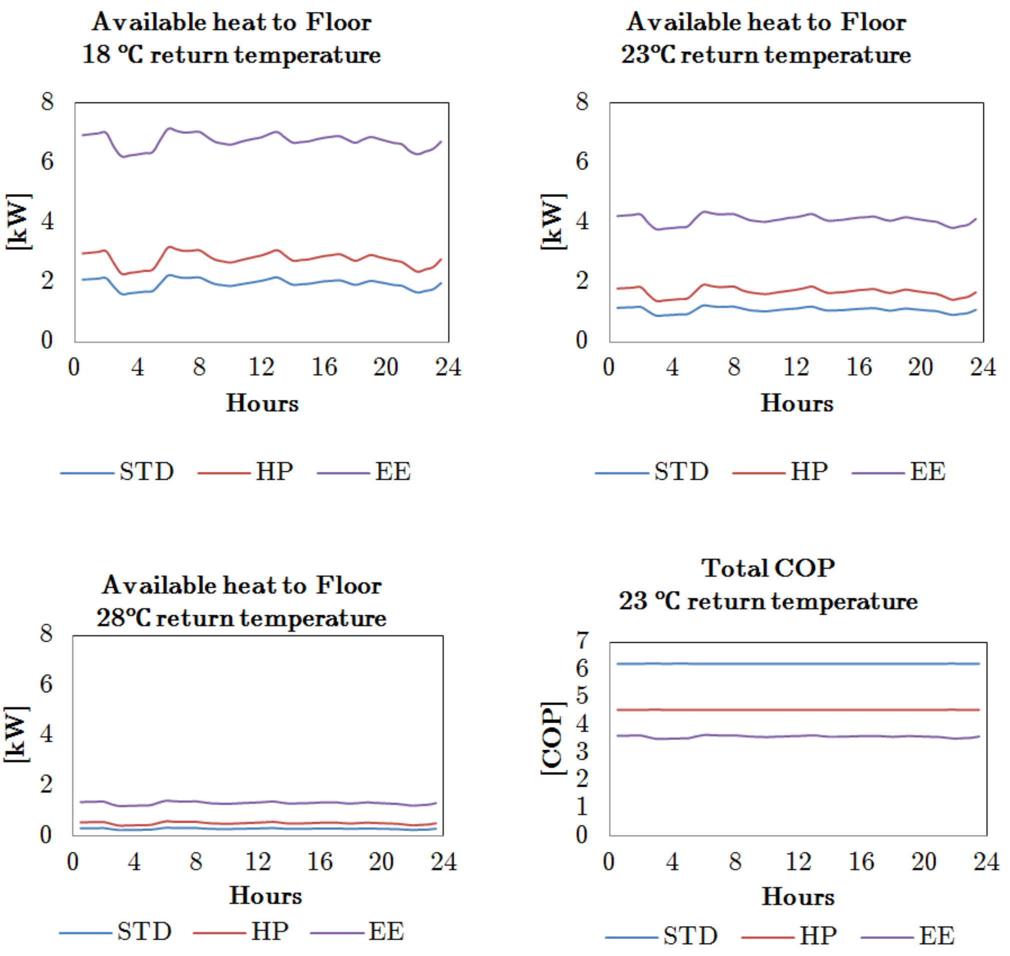


Fig. 9

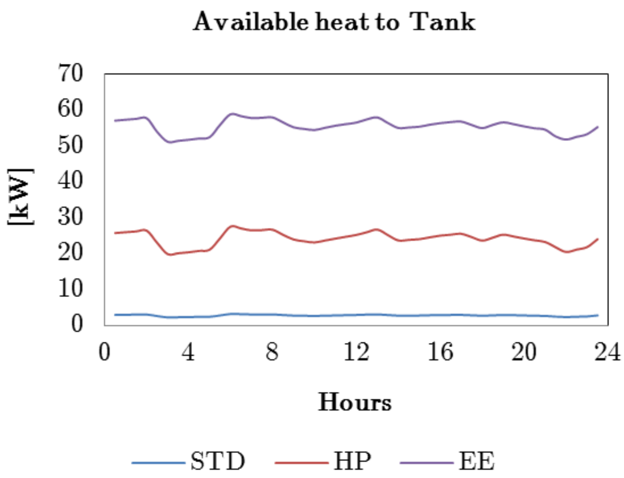


Fig. 10

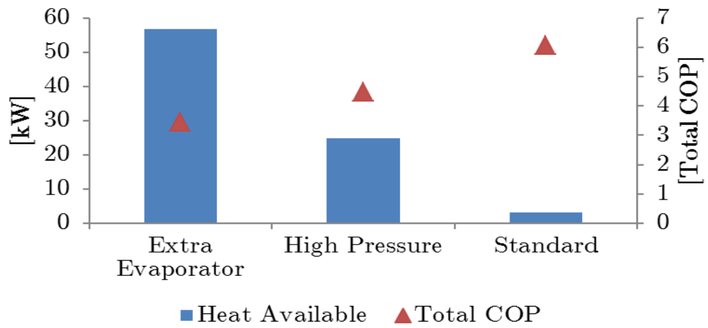


Fig. 11

The results show how different the outcome of each mode is. The standard mode showed a total COP of 6.02, but delivers very little heat to the gas coolers. At only 48 bar condensation takes place in GCIII so all this heat is dumped to the wells. Extra Evaporator mode delivers 56.7 kW in the worst case (28 ℃ return temperature) at a total COP of 3.5 on average. In EE mode the compressors use 5 times more energy than in standard mode. High pressure mode delivers 24.7 kW with a COP of 4.45 with an energy consumption that is about have of that in EE mode.

Annual energy consumption

The data from the simulations were organized into hourly values for a whole year using the weather analysis to distinguish the three variations of refrigeration load and the associated value for work, COP and available heat for each hour of the year. This was done for all operational modes at every return temperature variation. Seven different control strategies were tested for the plant. A simplified approach for the tank properties was used. A program created in excel generated the results from each of the strategies. The mode of operation for each hour and its corresponding compressor work and efficiency is determined by the control settings which will depend on the heat demand of that specific hour. Two sets of annual heat demand values were tested. The first heat demand came from measurements data for the REMA 1000 store "Dragvoll" (case 1). The other came from a heat demand simulation performed by Erwan Vaujany in the CREATIV supermarket calculation tool (case 2) for a previous thesis. The total heat demand of Case 1 is 95 088 kWh while the other was at 175 352 kWh. The first two strategies did not include the accumulation tanks and the results were only used as a comparison. The other five strategies (3-7) are presented. All the strategies are based on the three operational modes simulated. The heat available in the tanks for each hour n is given by

$$Q_{stored_{n+1}} - Q_{stored_n} = Q_{GCI_n} + Q_{GCII_n} - Q_{demand_n}$$

Strategies

- *Strategy 3:*

As long as the available heat in the tanks Q_{stored_n} exceeds the heat demand Q_{demand_n}Standard mode is engaged. If the heat demand is bigger, then the mode with lowest power consumption that can cover this demand on its own will be engaged.

- *Strategy 4:*

This strategy is the same as the previous except now the added sum of the available heat in the tank and the heat from the plant at that hour has to be exceeded by the demand before a higher mode is engaged.

- *Strategy 5:*

This approach utilizes the tank until the available heat is less the demand in which EE mode is engaged immediately. EE mode delivers on average about 30 kW more heat than HP mode and so ST mode, which yields the highest COP, is allowed to run for more hours after EE mode is finished compared to HP mode. HP mode is never used in this strategy. Whenever the tank can cover the demand ST mode is operational.

- *Strategy 6:*

If the tank falls below 200 kWh, then HP is engaged to recharge the tank. The goal is that there will always be enough heat available in the tank so EE is not engaged. Whenever the tank can cover the demand ST mode is operational.

- *Strategy 7:*

Standard mode is only engaged when the tank capacity is over 200 kWh. If this is not true and there is less available heat in the tank combined with the heat provided by HP mode, only then will EE mode be engaged. Otherwise HP mode will run. This ensures that electric heat is never used and EE mode is only engaged when absolutely necessary.

Results

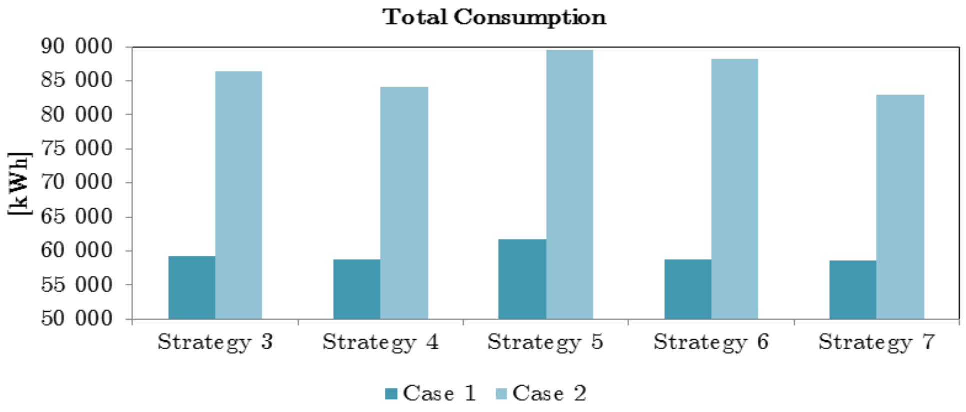


Fig. 12

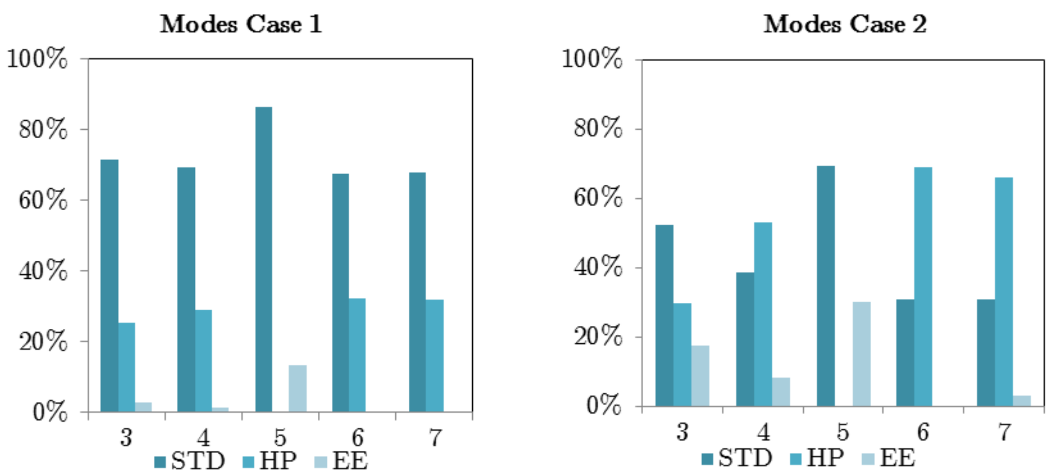


Fig. 13

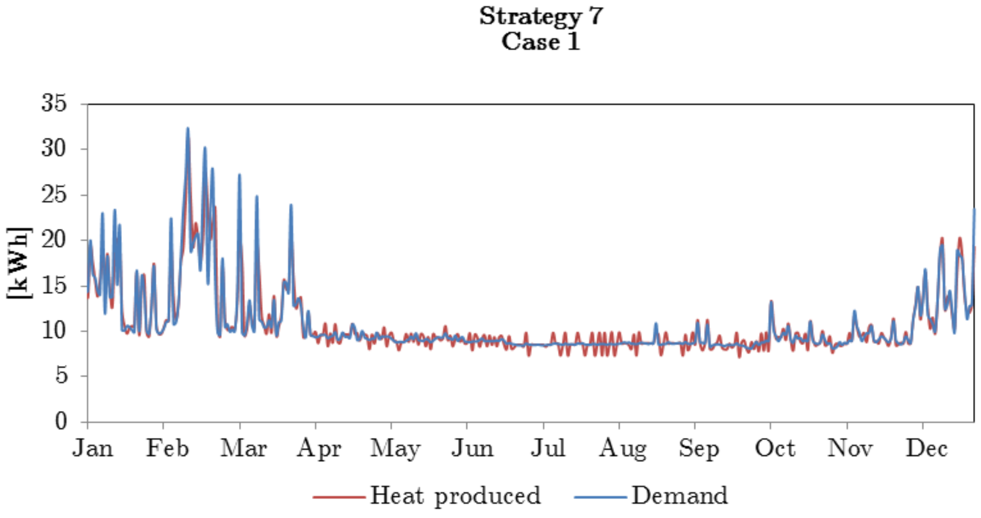


Fig. 14

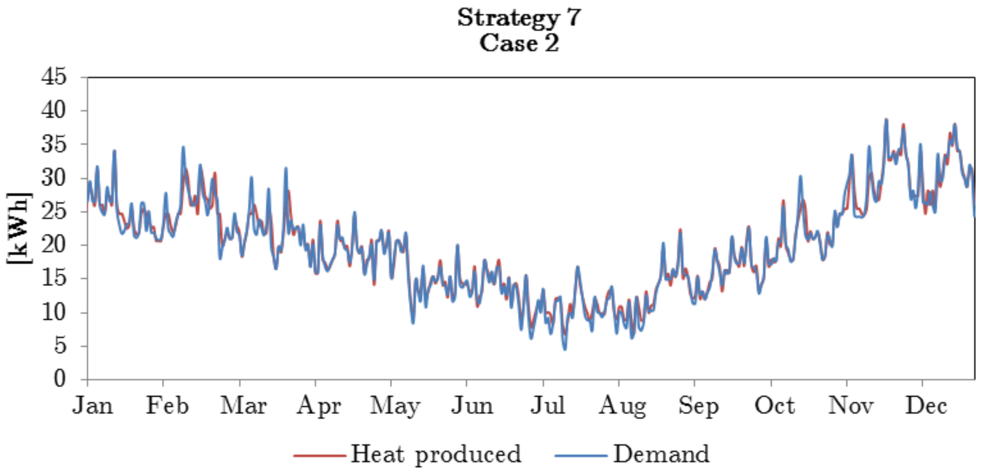


Fig. 15

There is less variation in the heat demand of Case 1 which led to very small differences between that various strategies. The demand is also quite low reducing the need for EE mode. There is bigger changes and in the Case 2 demand. Even though the total consumption of strategy 3 and 7 was low and not too far apart the plant operated very differently. The three modes behave quite uniquely, so one could imagine that the outcome of strategy 3 and 7 should be more distinctive. The main reason this is not the case is because of the accumulation tanks. How heat is provided becomes less important as everything is stored in the tank. It does not matter if 50 kWh of heat is produced when only 22 kWh is needed because the extra amount is stored and not wasted. Still, the results show that if the heat is provided using EE mode rather than HP mode, the SPF goes down since EE yields the poorest COP. The effect of the tank can be witnessed quite clearly during summer in Fig. 14. As the red line is above the blue it means that more heat is produced (made available) from the gas coolers than needed. Following is a period where the red line is below the blue by the same margin. This is the graphical representation of heat being stored in the tanks and utilized at a later point letting no available heat be wasted. The same behaviour occurs all year for both cases of course, but is not as noticable in the graphs during higher demand varaitions.

The SPF of Case 1 is for all strategies higher than for Case 2, which is expected. Higher demand requires higher modes which have lower COP. The best SPF for the two cases of demand was 5.15 and 4.61 for Case 1 and Case 2 respectively. Compared to a situation where there is no heat recovery and the plant only operates in standard mode the savings are 75 875 kWh (56.4%) for Case 1 and 131 874 kWh (61.4%) for case 2.

Discussion

The model simulates a near ideal situation. There is no pressure drop in the system, heat exchange is considered 100% efficient, the energy well is infinite and the compressors operate with 70 % efficiency regardless of the pressure ratio. Pumps for the well and the floor water circuit and the tanks have not been included in the calculation of total COP either. These are all simplifications that make the results more optimistic than they should be. However, the point is not to get exact figures, but to compare different scenarios and solutions. Moreover, the modes that are simulated represent extreme cases for the refrigeration plant. It is not intended that these modes will be operational very often when the supermarket is finished. There are a lot of ways in between these modes to operate the plant which will optimized for every situation and so performance is expected to be higher. By simulating these extreme cases one can get a sense of the limits of the system and the affect it has on the energy consumption.

Another simulation was performed to investigate a optimum standard mode at different floor return temperatures by varying the gas cooler from 50 to 120 bar for each case. The results (Fig. 16 and Fig. 17) strengthens the belief that the performance of the plant could be higher than the simulated 3 extreme modes indicate.

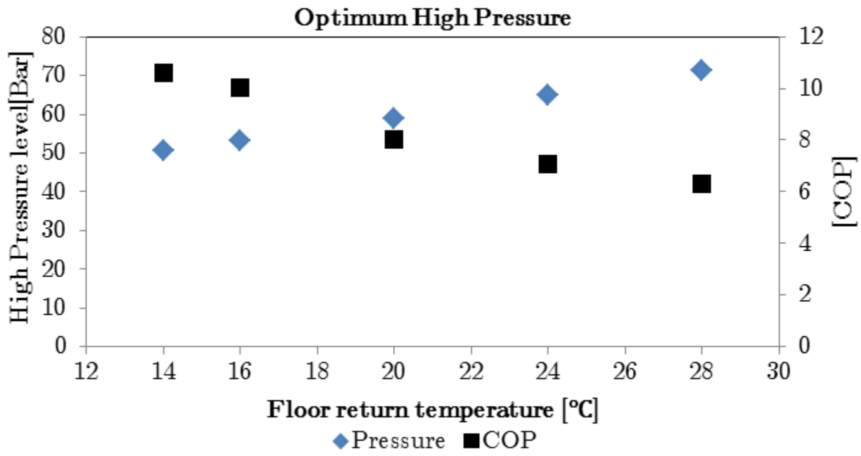


Fig. 16

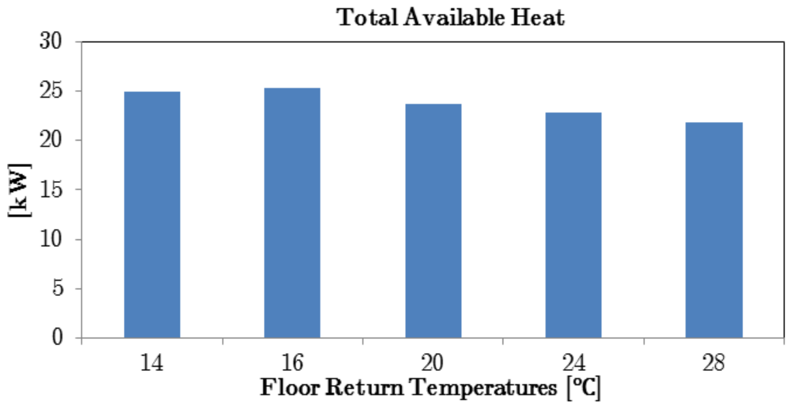


Fig. 17

Conclusion

Supermarkets are energy intensive buildings which are responsible for large amount of greenhouse gas emissions. The last couple of years have led to an increased interest in natural refrigerants with low environmental impact. The literature study shows that CO_2 based refrigeration plants and heat pumps can operate with high efficiency under the right conditions. A new supermarket with a system tailor made for a CO_2 booster plant with heat recovery has been designed by SINTEF and is currently under construction. Simulations of the refrigeration plant suggest that the whole heat demand of the building can be provided by heat recovery and with a high efficiency. Two cases of heat demand have been evaluated for 7 different strategies of plant operation. The results show that the best strategy can save between 50 and 60 % of the total energy consumption for refrigeration and space heating compared to a system without heat recovery.

B. Scenarios, Meteonorm, Data processing, HVAC modes

Scenario creater program

The first task during the work in this thesis was to create day scenarios for the weather in Trondheim. The scenarios come in the form of hourly values for temperature, humidity, direct radiation and diffuse radiation. The goal was to use this data for a simulation model of the ventilation system at Kroppanmarka. This model can produce valuable information on the store heating and cooling demand over the course of one day (day simulations) which will have an influence on the operation of both the ventilation and refrigeration system. It all comes down to estimating the power consumption and consequently the cost of operation. The simulations are done with a program called Dymola which uses the coding language Modelica.

Since the heating and cooling demand will be depend on the ambient conditions one would in principle need a simulation for each day of the year to get the best annual results. This is possible and indeed the intention for a future version of the simulation model, but as the model was when this work started it would be far too time consuming. Instead the solution was to create the day scenarios. By letting the scenarios represent typical days during the year the amount of simulations could be reduced from 365 to the number of scenarios chosen.

A program to produce the scenarios was created in excel. The scenarios that are created are based on weather data from Meteonorm7 which uses measurements gathered from weather stations across the globe to provide the information needed. Variables such as temperature, humidity and radiation in addition to many more can be directly downloaded from anywhere in the world. The data used for this thesis was recorded at a weather station in Værnes in 2010. This is found in the sheet called “Real Values”.

C	D	E	F	G	H	I	J
month	day	hour of day	Accumulated	T_ambient	Relative	Direct	Diffusive Radiation
				Temp	RH	Direct Vertical radiation	Diffusive radiation vertical plane
				°C	%	W/m^2	W/m^2
1	1	1	1	0.4	84	0	0
1	1	2	2	0.7	77	0	0
1	1	3	3	0.6	85	0	0
1	1	4	4	0.1	88	0	0
1	1	5	5	-0.5	86	0	0
1	1	6	6	0.4	78	0	0
1	1	7	7	0.1	80	0	0
1	1	8	8	0.7	77	0	0
1	1	9	9	0.6	87	0	0
1	1	10	10	0.8	97	0	0
1	1	11	11	0.3	94	0	19
1	1	12	12	0.3	100	91	80
1	1	13	13	0.1	94	70	73
1	1	14	14	0.6	91	0	22
1	1	15	15	-0.1	98	0	0
1	1	16	16	0.1	98	0	0
1	1	17	17	-0.1	99	0	0
1	1	18	18	0.2	96	0	0
1	1	19	19	0.2	97	0	0
1	1	20	20	0.1	95	0	0
1	1	21	21	0.6	84	0	0
1	1	22	22	0.3	83	0	0
1	1	23	23	-0.3	91	0	0
1	1	24	24	0.4	77	0	0
1	2	1	25	-0.4	81	0	0
1	2	2	26	-0.6	83	0	0
1	2	3	27	-0.4	83	0	0
1	2	4	28	-1.1	98	0	0

Fig. 18

The best way to describe how this program works is with an example. To get a better picture of the process Fig. 24 should be used with Fig. 25 while reading. This example will be using temperature, but everything works the exact same same way for the other parameters as well. The data from Meteonorm is in the form of hourly values for a whole year (8760 values) in one single column per parameter. The first thing the program does is to reorganize the data so it can be processed. One row represents one day and each hour of these days are in 1 to 24 columns (Fig. 19). This makes the data easier to work with. This is found in the sheet called Temperature. There is a corresponding sheet for the other parameters called *Humidity, Direct* and *Diffuse.*

F	G	H	I	J	K	L	M	N	O	P	Q	R	S	T	U	V	W	X	Y	Z	AA	AB	AC	AD	AE	AF
Month	Day	Hour	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
1	1		0.4	0.7	0.6	0.1	-0.5	0.4	0.1	0.7	0.6	0.8	0.3	0.3	0.1	0.6	-0.1	0.1	-0.1	0.2	0.2	0.1	0.6	0.3	-0.3	0.4
1	2		-0.4	-0.6	-0.4	-1.1	-1.2	-1.4	-1.5	-1.4	-1.6	-1.2	-0.7	-1.2	-1.9	-1.1	-1.5	-1.1	-1.3	-1.7	-2	-1.9	-2.3	-2	-2.4	-1.8
1	3		-2.1	-2	-2.2	-2.1	-2.1	-1.8	-2.1	-1.9	-1.8	-1.6	-1.5	-1.8	-2.1	-2	-3.1	-2.9	-3.3	-2.7	-3.1	-3.7	-3.7	-5.3	-4.7	-4.6
1	4		-4.3	-6	-7.4	-7	-7	-7.5	-8.2	-9.3	-8.1	-7.8	-8.2	-7.7	-8.5	-8.1	-8	-7.7	-7.1	-6.5	-6.4	-5.8	-4.8	-4.7	-4.9	-4
1	5		-4	-3.9	-3.9	-3.5	-3.5	-3.3	-3.9	-3.8	-3.6	-3.8	-3.4	-3.1	-2.5	-3	-5.4	-5.5	-1.8	-1.5	-0.8	-0.2	0.5	0.4	0.3	0.6

Fig. 19

In order to create the scenarios it was necessary to divide the weather data into categories. The categories were named *very high, high, medium, low* and *very low* referring to the magnitude of the parameter. How to separate the categories is done in the INPUT sheet. Fig. 20 show the inputs for the temperature categories. The category *high* is in this example defined as a day where the average temperature is above 10°C and below 17°C. This means that every day from the meteonorm7 weather data that had an average temperature between 10 and 17°C will fall under the category *high*. The category *Very High* ranges in this example from an average temperature of 17°C and above, while *Very Low* ranges from -6°C and all values below. In the case of radiation the categories are divided by average values of W/m^2 and relative humidity as percentages. For temperature *Low* is sometimes called *Cold* and *Hot* is the same as *High.*

Category	Range
Very High	
	17
High	
	10
Medium	
	3
Low	
	-6
Very Low	

Fig. 20

After all the categories have been defined the program goes through all the data from metenorm and registers which category each day of the year belongs to. Fig. 21 displays the first two weeks of January and the distribution of categories in this period. It shows that there were three *very cold* days while the rest was *cold* based on our definitions of each category. 1000 means "yes" and is used for the sake of simplicity as it is easier to spot than for instance 1. This was very useful and quite necessary during the writing of codes and functions for the program, making mistakes much easier to spot. A table for every month and the whole year in total is also produced to give a better overview of the category distribution. From Fig. 21 one can see that there were 45 *very cold* days and 20 of these occurred in December. This can found in the Temperature-sheet. (low = cold, and high=warm, for temperature.)

	Very Hot	Hot	5<T<15	Cold	Very cold
1	0	0	0	1000	0
2	0	0	0	1000	0
3	0	0	0	1000	0
4	0	0	0	0	1000
5	0	0	0	1000	0
6	0	0	0	1000	0
7	0	0	0	1000	0
8	0	0	0	1000	0
9	0	0	0	1000	0
10	0	0	0	1000	0
11	0	0	0	0	1000
12	0	0	0	1000	0
13	0	0	0	1000	0
14	0	0	0	0	1000

Fig. 21

	Very High	High	Medium	Low	Very Low
Jan	0	0	0	28	3
Feb	0	0	1	18	9
Mars	0	0	4	27	0
April	0	1	15	14	0
Mai	1	8	16	6	0
June	1	18	11	0	0
Jul	14	16	1	0	0
Aug	11	19	1	0	0
sept	2	12	16	0	0
okt	0	8	12	11	0
nov	0	0	2	15	13
dec	0	0	0	11	20
Tot	29	82	79	130	45

Fig. 22

The 45 *very cold* days should be represented by 1 *very cold* -scenario, and the 130 *cold days* should be represented by 1 *cold*-scenario and so on.
The program now takes the average temperature of each hour of each day within a category and creates with this one day to represent them all. For example from Fig. 23 one can see that for all the 45 days that fell under the category *very cold* the average temperature at 01.00 was -7.361°C, which is why this is the temperature for the scenario representing *very cold* days at this hour. The procedure naturally occurs for every hour, for every category, for every parameter. This is found in the *Scenario*-sheet

	Temperature											
Hours	1	2	3	4	5	6	7	8	9	10	11	12
Very Hot	15.605	15.235	14.915	14.885	15.24	15.855	16.645	17.4	18.22	19.055	19.885	20.615
Hot	10.566	9.7211	8.9734	9.0073	9.2606	9.8055	10.584	11.495	12.531	13.471	14.345	15.057
Hot+very Hot	13.086	12.478	11.944	11.946	12.25	12.83	13.615	14.448	15.376	16.263	17.115	17.836
Medium	3.7167	3.2732	2.8862	2.742	2.7043	2.8051	3.1174	3.7087	4.4681	5.3819	6.5507	7.6413
Cold + very cold	-4.299	-4.833	-5.32	-5.725	-6.028	-6.298	-6.517	-6.655	-6.548	-6.148	-5.206	-4.112
Cold	-1.238	-1.623	-2.008	-2.323	-2.529	-2.724	-2.863	-2.893	-2.64	-1.903	-0.623	0.6475
Very Cold	-7.361	-8.044	-8.633	-9.128	-9.528	-9.872	-10.17	-10.42	-10.46	-10.39	-9.789	-8.872

Hours	13	14	15	16	17	18	19	20	21	22	23	24
Very Hot	21.325	21.98	22.45	22.675	22.66	22.4	21.91	21.255	20.55	19.73	18.44	16.805
Hot	15.603	16.071	16.336	16.435	16.302	15.932	15.406	14.743	14.038	13.266	12.377	11.329
Hot+very Hot	18.464	19.025	19.393	19.555	19.481	19.166	18.658	17.999	17.294	16.498	15.409	14.067
Medium	8.3804	8.8428	8.7891	8.4971	8.1717	7.6783	7.1761	6.6768	6.1725	5.6754	5.1659	4.6246
Cold + very cold	-3.537	-3.337	-3.775	-4.069	-4.367	-4.594	-4.712	-4.915	-5.01	-5.134	-5.248	-5.405
Cold	1.37	1.7213	1.3063	0.8238	0.41	0.095	-0.169	-0.43	-0.686	-0.969	-1.229	-1.504
Very Cold	-8.444	-8.394	-8.856	-8.961	-9.144	-9.283	-9.256	-9.4	-9.333	-9.3	-9.267	-9.306

Fig. 23

The scenarios now have to be placed in a sequence that combined represents a whole year. The program will therefore check which of the categories are “true” for each day of the actual year. One can see that the first three days were true for the category *Cold* (Fig. 21). As a result the first three days of the scenario-based year will be filled with the *cold-scenario*. A whole year is created like this using the 5 scenarios for each parameter.

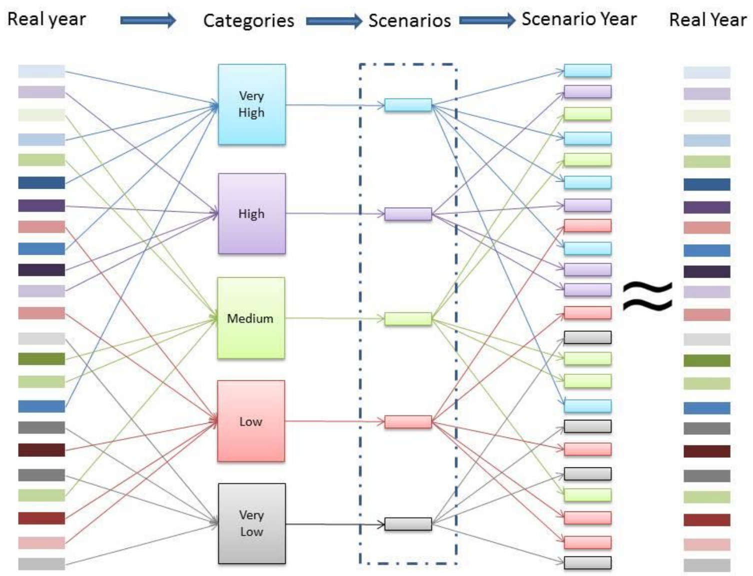


Fig. 24

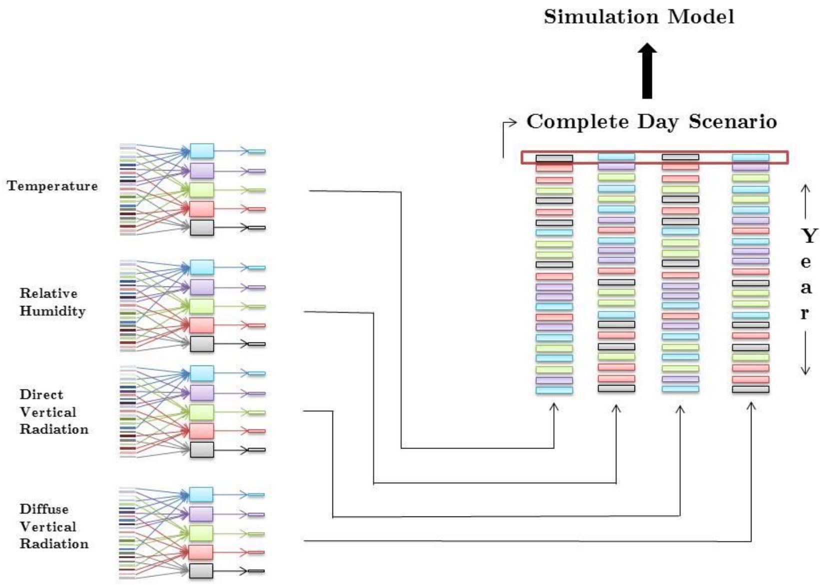


Fig. 25 Each of the parameters creates one scenario for each day that combined becomes a complete day-scenario.

The simulation model in Dymola uses all the variables in one simulation meaning that a representative day of temperature, of humidity, of direct vertical radiation and one of diffuse vertical radiation is used. It is the combination of each parameter that creates a day-scenario and. If all these parameters have 5 different variations one can potentially have $5 \times 5 \times 5 \times 5 = 625$ different day-scenarios. This means that one should perform 625 simulations which make no sense. If constant humidity is assumed, meaning only one representative day for the whole year it is $1 \times 5 \times 5 \times 5 = 125$ simulations. By letting the categories *Low* and *Very Low* merge into one scenario called *Low* and the same with *High* and *Very High,* each parameter is reduced to 3 scenarios and the total amount of day-scenarios become $1 \times 3 \times 3 \times 3 = 27$ (Fig. 26). The program creates data both for the case where 125 scenarios are used and for the case with 27. The sheets are called 125Days and 27Days. It is in these sheets that the scenario year is created and where all the values are reorganized back to the same original form from Meteonorm with each parameter in one single column.

Fig. 27 show a part of the 27Days sheet where one can see what kind of day scenario is created. The first day in this figure is a day where the temperature will be covered by the medium scenario and where both direct and diffuse radiation will be covered by the low scenario. This is scenario 18 as showed by Fig. 28.

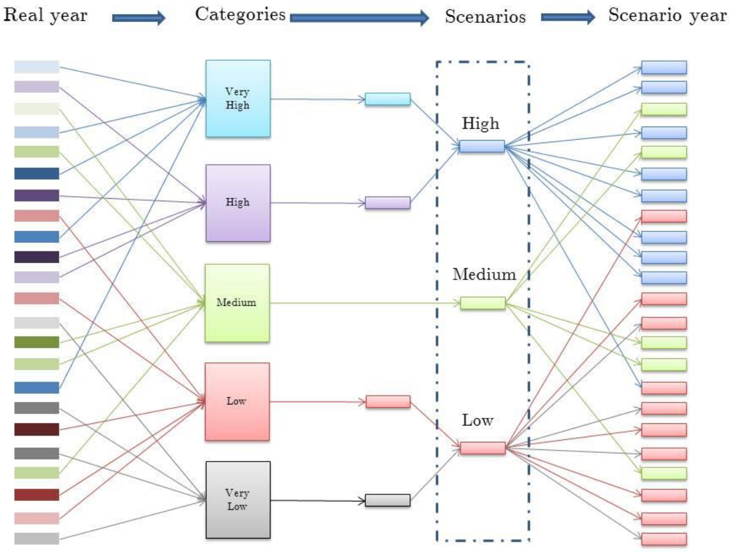


Fig. 26

Temperature			Direct			Diffuse		
High	Medium	Low	High	Medium	Low	High	Medium	Low
0	1000	0	0	0	1000	0	0	1000
0	0	1000	0	1000	0	0	0	1000
0	1000	0	0	0	1000	0	0	1000
0	0	1000	0	1000	0	0	0	1000
0	0	1000	0	0	1000	0	0	1000
0	0	1000	0	0	1000	0	0	1000
0	0	1000	0	0	1000	0	0	1000

Fig. 27

In the Scenario Distribution sheet it is possible to see how often each scenario is used during the whole year. The reason for this is because the results that come from the simulation of a specific scenario must be multiplied with the amount of occurrences it had. That way an annual estimation can be made. Simulation results of Scenario number 8 will for instance be multiplied with 29 (Fig. 29) while scenario number 6 doesn't have to be simulated at all.

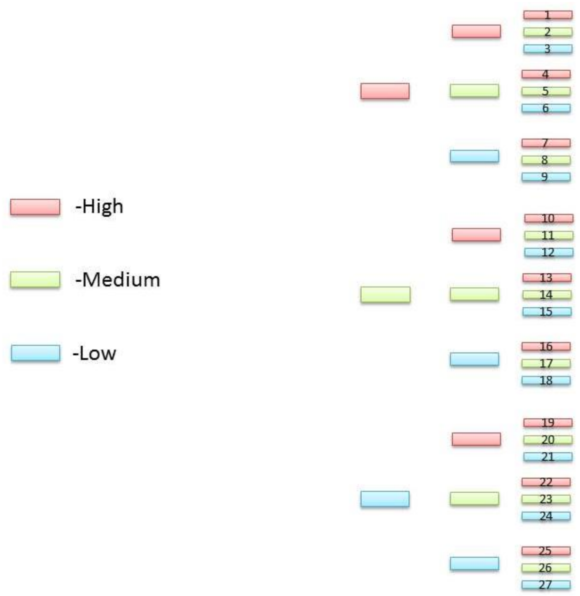


Fig. 28

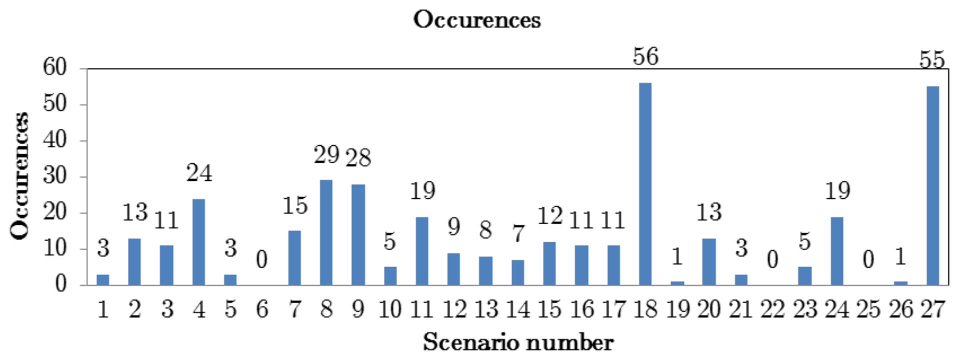


Fig. 29

A full year is at this point created for each of the parameters and it is plotted and compared to the real values that they are representing. Fig. 30 shows how 5 scenarios of temperature (blue) match the real days (red) of the course of one year. From this graph it looks like the scenarios match the real case pretty closely, but for a more definite comparison one can look in the sheet called Relative Deviation.

In this sheet the difference between the scenario values and real values are calculated. It measures, for all the parameters, how far of each month are, each week, each day and even each hour. Every scenario hour is in other words judged on accuracy. Far more interesting is to see the weekly and monthly average relative deviations as one hour alone can't say much about the year in total. The program calculates the relative deviation both for the 125 day scenario and for the 27 day scenario so that it is possible to see how much one is penalized by reducing the number of day-scenarios.

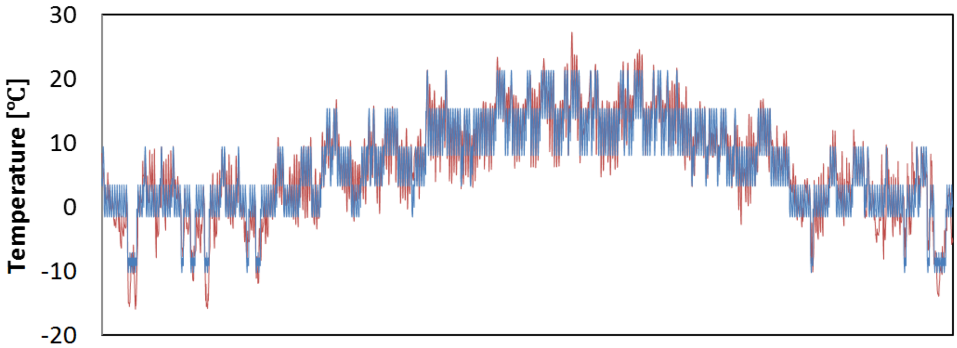


Fig. 30

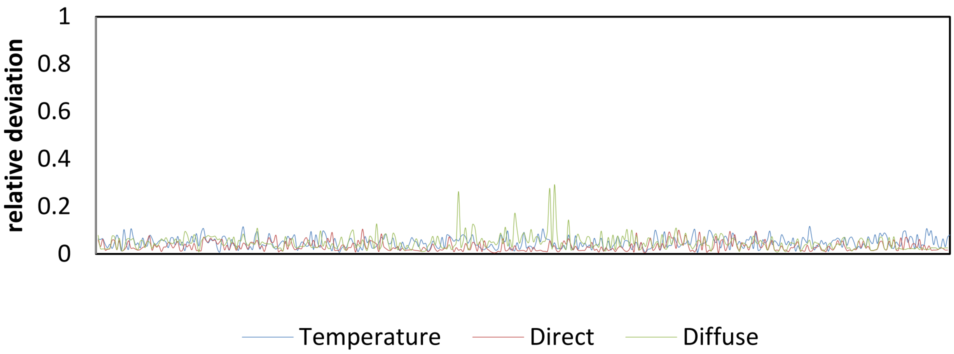
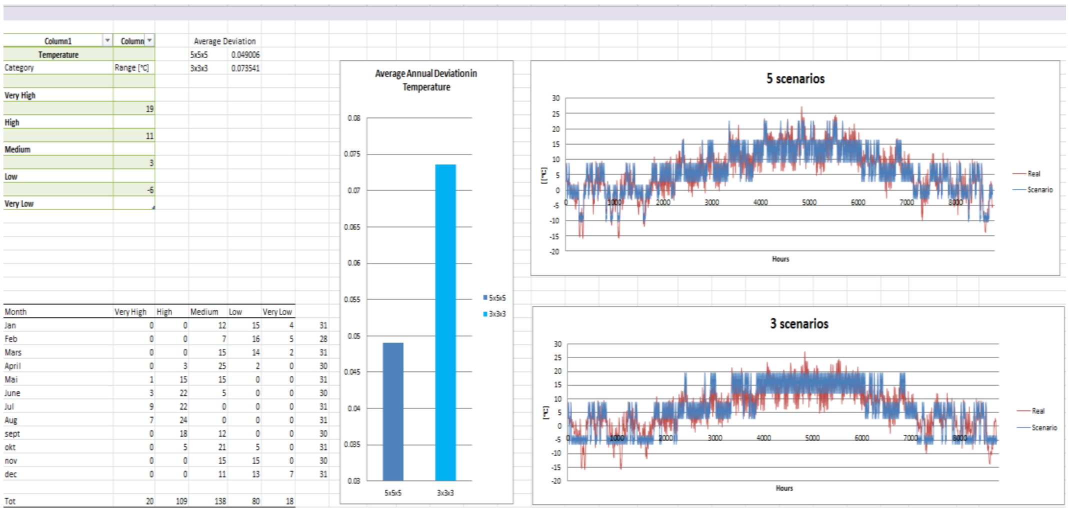


Fig. 31

Although the steps described in this section actually occur in that order explained it is all done in a matter of seconds. The only thing that needs to be done by the user is to insert the values from meteonorm in the REAL VALUES sheet, and enter the ranges for each category in the INPUT sheet. The ranges of one parameter will not affect those of another which means that they can be optimized individually. Located next to the table of input ranges are both graphs and numbers on relative deviation as well as the visual comparison between the real and scenario values for both the 27 day-scenario case and the 125 day-scenario case. This lets the user immediately after entering a range see whether or not this has led to a better fit with the real values which makes optimization easier.



Column1	Column
Temperature	
Category	Range [°C]
Very High	
	19
High	
	11
Medium	
	3
Low	
	-6
Very Low	

Average Deviation	
5x5x5	0.049006
3x3x3	0.073541

Month	Very High	High	Medium	Low	Very Low	
Jan	0	0	12	15	4	31
Feb	0	0	7	16	5	28
Mars	0	0	15	14	2	31
April	0	3	25	2	0	30
Mai	1	15	15	0	0	31
June	3	22	5	0	0	30
Jul	9	22	0	0	0	31
Aug	7	24	0	0	0	31
sept	0	18	12	0	0	30
okt	0	5	21	5	0	31
nov	0	0	15	15	0	30
dec	0	0	11	13	7	31
Tot	20	109	138	80	18	

Fig. 32

Summary

All that is needed to produce a set of scenarios are hourly data from meteonorm or similar programs. Once the input values have been entered for the categories everything is calculated. The graphs and table can be viewed while trying different inputs to optimize the scenarios. It can in principle be used for any place in the world, but the input values always have to be carefully selected to get the best results. If other properties than temperature, humidity and the two types of vertical radiation is wanted one simply inserts this data instead. To illustrate this the excel program was tested with data from a weather station located at Dubai International Airport which is almost as far away from the weather in Trondheim one could possibly get. Instead of downloading the vertical radiation values the *horizontal* were selected as a test to see how the program reacted. The visual results of this test presented on the next page. It can be concluded that the program works quite effective in this case as well, when looking at the results.

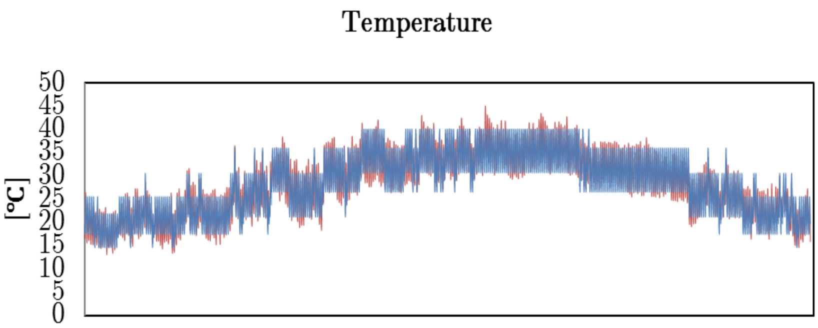


Fig. 33

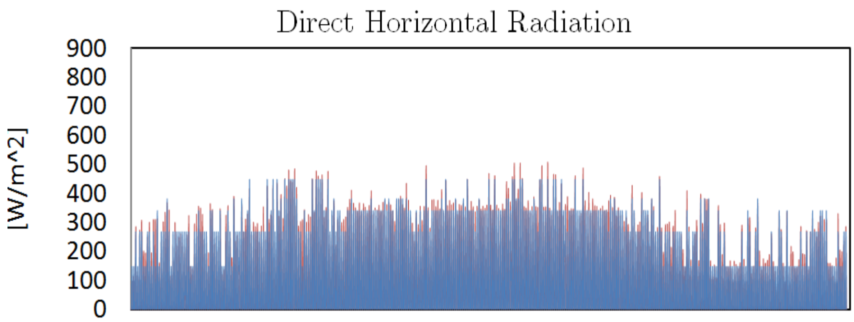


Fig. 34

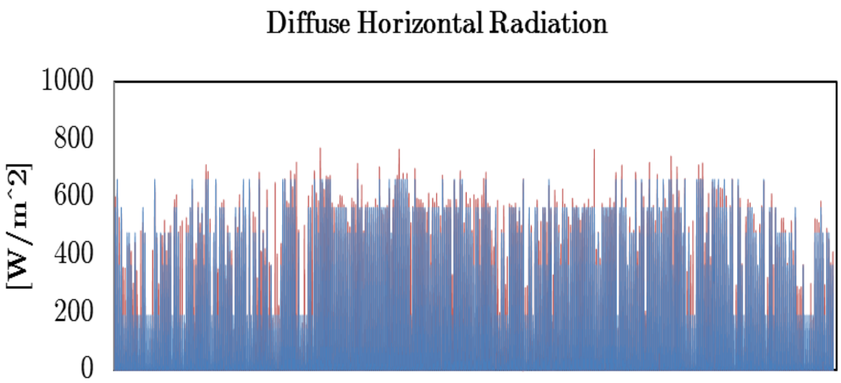


Fig. 35

Annual Relative Deviation	
Temperature	0.0388
Direct Horizontal	0.0280
Diffuse Horizontal	0.0328

Fig. 36

Description of metenorm

Short Review 1 *meteonorm*

1 Short Review

What is *meteonorm*?

meteonorm is a comprehensive climatological database for solar energy applications:

- ***meteonorm*** is a meteorological **database** containing comprehensive climatological data for solar engineering applications at every location on the globe. The results are stochastically generated typical years from interpolated long term monthly means. They represent a mean year of the selected climatological time period based on the user's settings. As such the results do not represent a real historic year but a hypothetical year which statistically represents a typical year at the selected location.
- ***meteonorm*** is a **computer program** for climatological calculations.
- ***meteonorm*** is a **data source for engineering design programs** in the passive, active and photovoltaic application of solar energy with comprehensive data interfaces.
- ***meteonorm*** is a **standardization tool** permitting developers and users of engineering design programs access to a comprehensive, uniform meteorological data basis.
- ***meteonorm*** is **a meteorological reference** for environmental research, agriculture, forestry and anyone else interested in meteorology and solar energy.

What is it based on?

meteonorm's orderly facade conceals not only numerous **databases** from all parts of the world but also a large number of computational models developed in international research programs.

***meteonorm* is primarily a method for the calculation of solar radiation on arbitrarily orientated surfaces at any desired location**. The method is based on databases and algorithms coupled according to a predetermined scheme. It commences with the user specifying a particular location for which meteorological data are required, and terminates with the delivery of data of the desired structure and in the required format.

Depending on user requirements, the calculation procedure employs between one and four computation models (Tab. 1.1). In addition to the monthly values, ***meteonorm*** provides maximum radiation values under clear sky conditions. For Switzerland, standardized data (design reference years) for building simulation purposes are available for a number of locations.

Reference: Meteonorm Handbook pt. 1:
http://meteonorm.com/fileadmin/user˙upload/mn˙7/mn7˙software.pdf

Simulation Data Processing

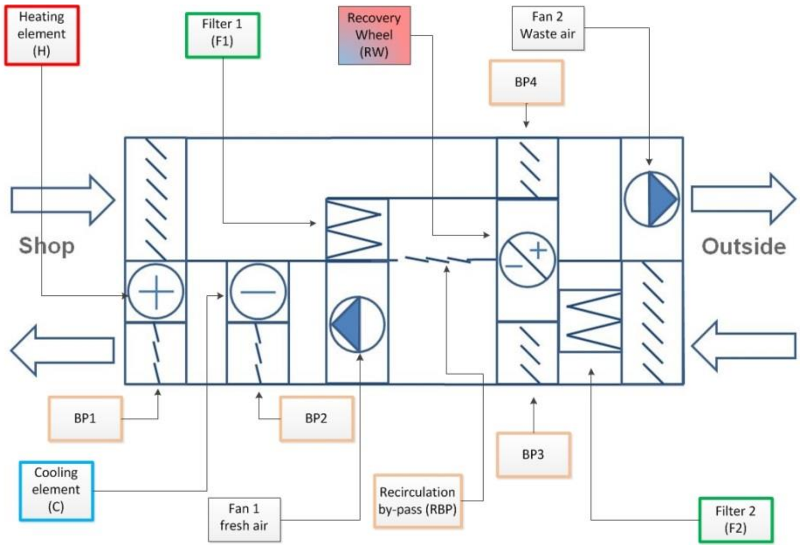
The simulation results sent from Braunschweig were raw untreated data which was not presentable. This has to do with the way the simulation model works. The iterative processes in the model become faster if there is an initiation period in which the thermodynamic data is allowed to become stable within the model simulation. For that reason there is several thousand values without any purpose that comes with the results. In addition to this there is also no linear time step for each data value. The simulation stretches over 24 hours, but one can have 100 values between the two first seconds of the first hour and then maybe 4 values in the next 5 minutes of the day. Some simulation had a total of 800 usable values within the 24 hours and some had 3000 which complicated things further since they were all to be compared.

A total of 27 simulations each with 5 properties; total heat, total COP, Refrigeration COP, GCI heat and GCII heat, had to be processed, each with the issues describet above.

The solution was to use VBA (Visual Basic for Applications) coding in Excel, which initially was unfamiliar territory. Two codes were written. One code was to identify and delete all unimportant values (initiation process values). The other code had to save all values at a time step of 1800 s (30 minutes) for each property and delete all in between, then arrange the saved values in a sequence of 48 (30 min*48=24 hours), regardless of the total number of raw data values from the simulations. That way, the values from one simulation at for instance 2 pm could be compared with the values from another simulation at the same time of day.

After this was done the values were organized by the type of simulation (all winter simulations together and so on) and a full year could be generated by synchronizing it with the temperature category distribution from the weather scenario program previously described.

HVAC modes



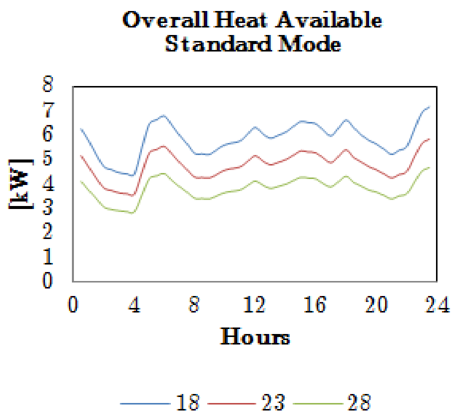
- Path 1: Air from the shop is blown straight out through BP4. The same amount of ambient air is entering the shop going through F2 and then BP3, BP2 and BP1.
- Path 2: Air from the shop goes through F1, then RBV, enters C and is blown into the store again through BP1. Full recirculation.
- Path 3: Air from the shop goes through F1, then RBV, trough BP2 and then H before entering the store. Full recirculation.
- Path 4: Air from the shop goes through F1, then RBV and then continuous through both C and H. Full recirculation.
- Path 5: Part of the shop air goes through F1, then RBP before C and lastly BP1. The other part goes straight out through BP4. Fresh air from the outside goes through F2, then BP3 before entering C and finally BP1. Partial recirculation.
- Path 6: Part of the shop air goes through F1, then RBP before BP2 and lastly H. The other part goes straight out through BP4. Fresh air from the outside goes through F2, then BP3 before entering BP2 and finally H. Partial recirculation.
- Path 7: Part of the shop air goes through F1, then RBP before C and lastly H. The other part goes straight out through BP4. Fresh air from the outside goes through F2, then BP3 before entering C and finally H. Partial recirculation.
- Path 8: Air from the shop goes directly out through BP4. Ambient air goes through F2, then BP3, enters C and finally into the shop through BP1.

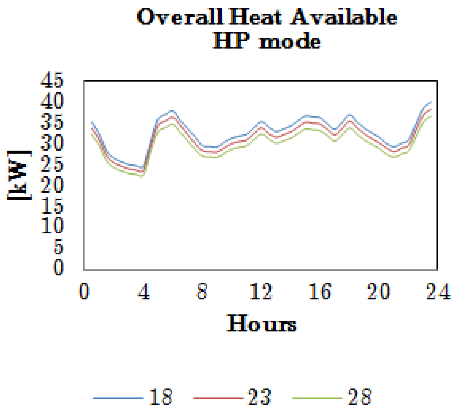
- Path 9: Air from the shop goes directly out through BP4. Ambient air goes through F2, then BP3, enters BP2 and finally H.
- Path 10: Air from the shop goes directly out through BP4. Ambient air goes through F2, then BP3, Through C and finally H.
- Path 11: Part of the air from the shop goes through F1, then RBP and then back to the shop through BP2 and BP1. The other part goes through F1 then RW and out. The fresh air goes through F2, then RW before entering the shop through BP2 and BP1.
- Path 12: Part of the air from the shop goes through F1, then RBP and then back to the shop through BP2 and finally H. The other part goes through F1 then RW and out. The fresh air goes through F2, then RW before entering the shop through BP2 and H.
- Path 13: Part of the air from the shop goes through F1, then RBP and then back to the shop through C and H. The other part goes through F1 then RW and out. The fresh air goes through F2, then RW before entering the shop through C and H.
- Path 14: The shop air goes through F1 and RW, then out. Fresh air enters through F2, RW and into the store through BP2 and BP1.
- Path 15: Air from the shop goes out through F1 and RW. The fresh air goes through F2, RW then BP2 and finally H.
- Path 16: Air from the shop goes out through F1 and RW. The fresh air goes through F2, RW then C and finally H.

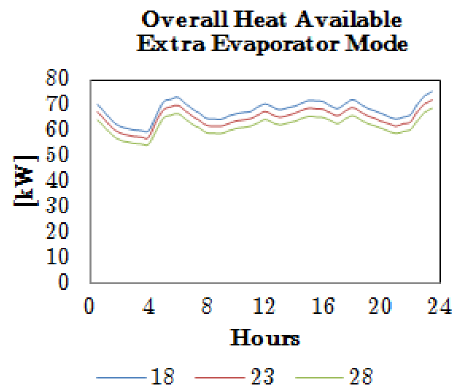
C. Simulation Results

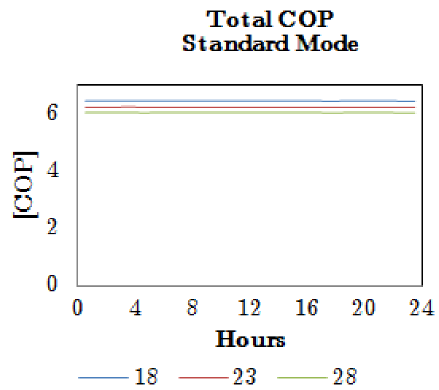
The complete set of total heat and total COP results from the simulations is found in this section.

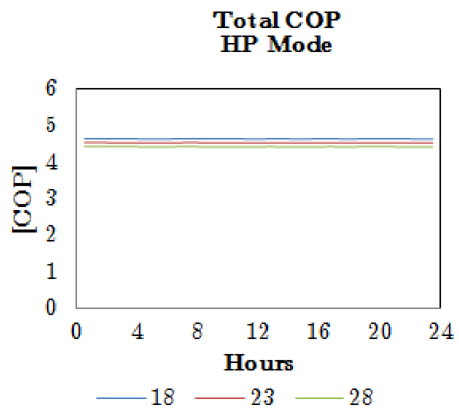
Summer

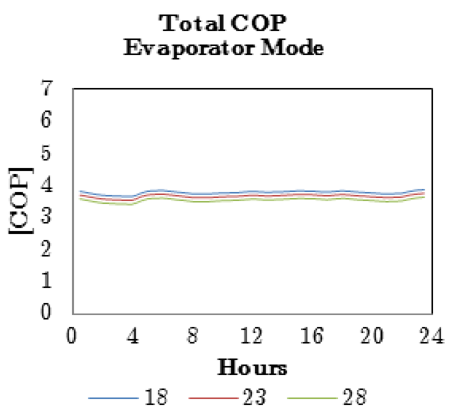


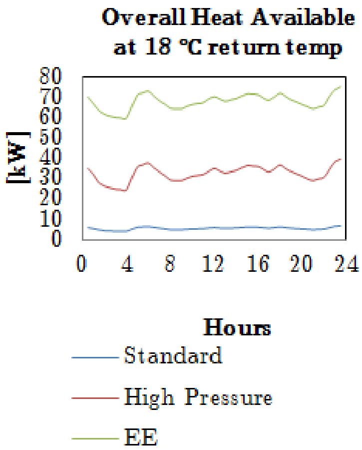


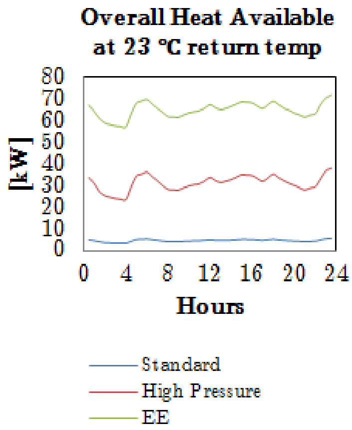


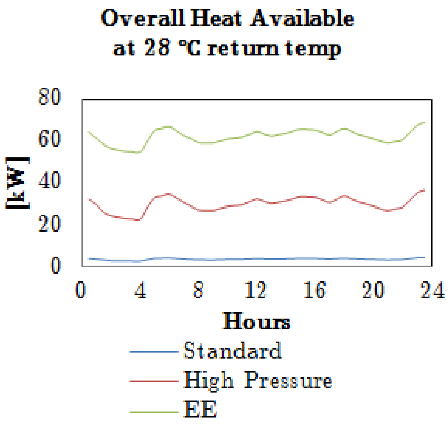


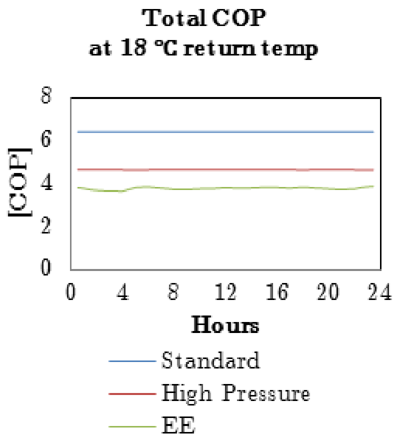


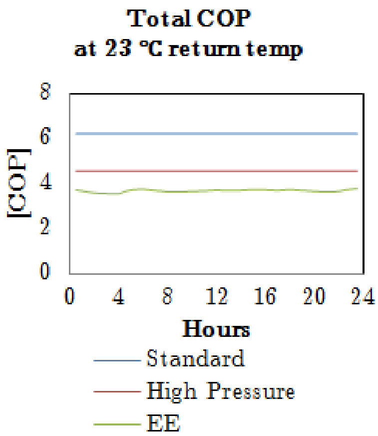


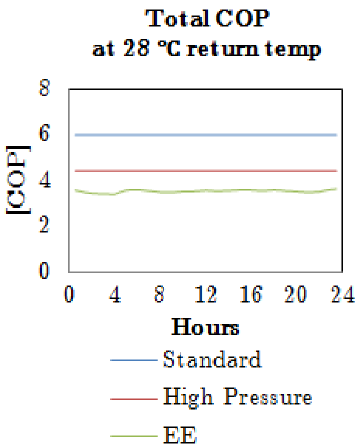

Overall Heat Available
at 18 ℃ return temp
[kW]
80
70
60
50
40
30
20
10
0
0 4 8 12 16 20 24
Hours
Standard
High Pressure
EE


Overall Heat Available
at 23 ℃ return temp
[kW]
80
70
60
50
40
30
20
10
0
0 4 8 12 16 20 24
Hours
Standard
High Pressure
EE


Overall Heat Available
at 28 ℃ return temp
[kW]
80
60
40
20
0
0 4 8 12 16 20 24
Hours
Standard
High Pressure
EE


Total COP
at 18 ℃ return temp
[COP]
8
6
4
2
0
0 4 8 12 16 20 24
Hours
Standard
High Pressure
EE

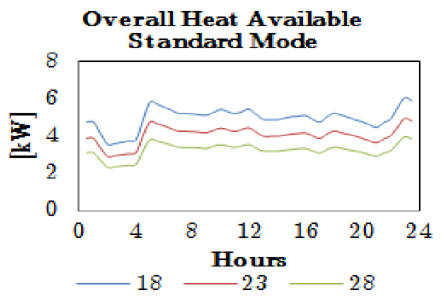

Total COP
at 23 ℃ return temp
[COP]
8
6
4
2
0
0 4 8 12 16 20 24
Hours
Standard
High Pressure
EE

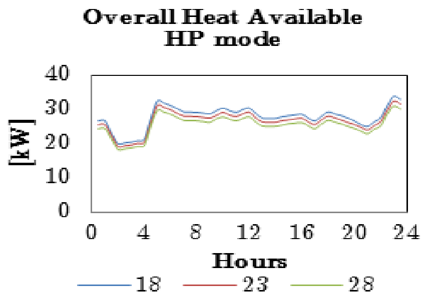

Total COP
at 28 ℃ return temp
[COP]
8
6
4
2
0
0 4 8 12 16 20 24
Hours
Standard
High Pressure
EE

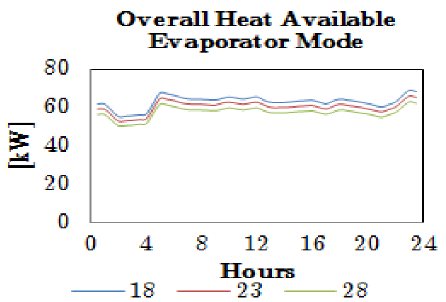
Average Summer values

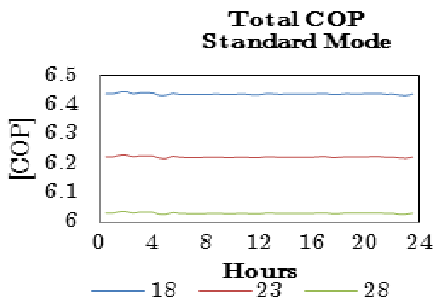
	18			23			28		
	STD	HP	EE	STD	HP	EE	STD	HP	EE
Power	5.04	12.75	25.07	5.04	12.75	25.07	5.04	12.75	25.07
GCI	3.48	29.37	60.61	3.47	29.37	60.62	3.47	29.37	60.61
GCII	2.39	3.39	7.38	1.31	2.05	4.51	0.35	0.63	1.47
Ref COP	5.26	2.08	1.06	5.26	2.08	1.06	5.26	2.08	1.06
Tot COP	6.43	4.66	3.79	6.22	4.56	3.68	6.03	4.44	3.55

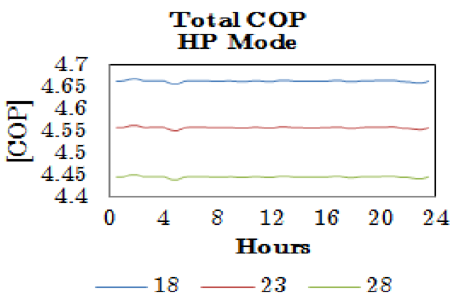
Spring/Fall

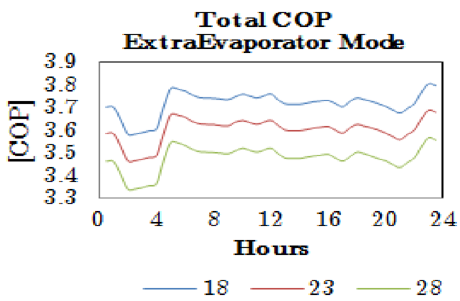


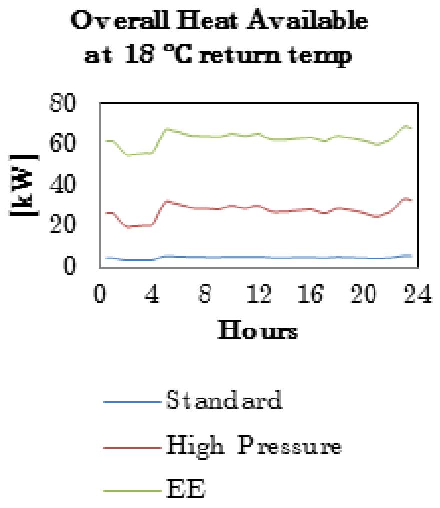


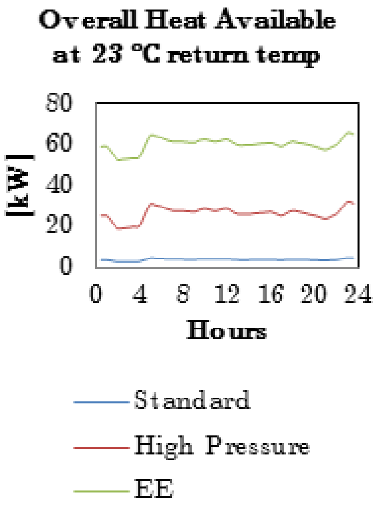


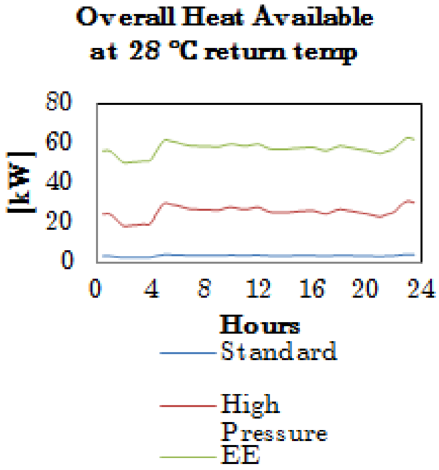


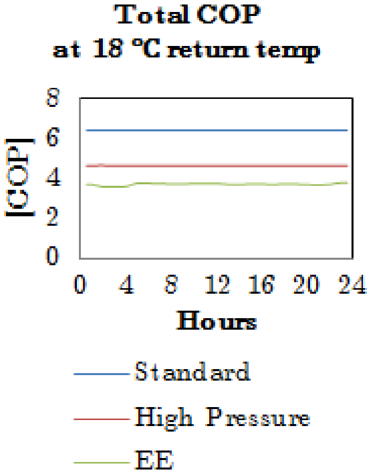


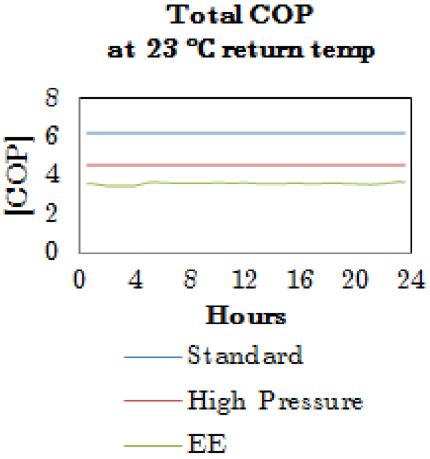


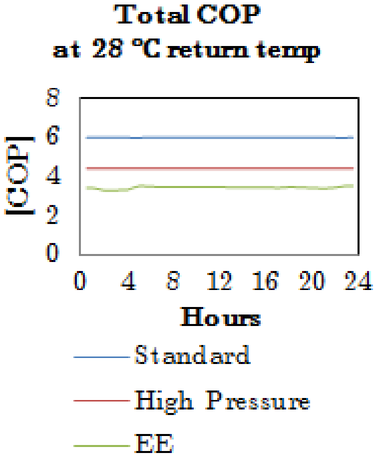

Overall Heat Available
at 18 °C return temp
[kW]
0
20
40
60
80
0 4 8 12 16 20 24
Hours
Standard
High Pressure
EE


Overall Heat Available
at 23 °C return temp
[kW]
0
20
40
60
80
0 4 8 12 16 20 24
Hours
Standard
High Pressure
EE


Overall Heat Available
at 28 °C return temp
[kW]
0
20
40
60
80
0 4 8 12 16 20 24
Hours
Standard
High Pressure
EE


Total COP
at 18 °C return temp
[COP]
0
2
4
6
8
0 4 8 12 16 20 24
Hours
Standard
High Pressure
EE

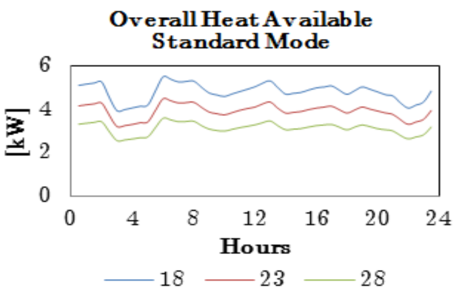

Total COP
at 23 °C return temp
[COP]
0
2
4
6
8
0 4 8 12 16 20 24
Hours
Standard
High Pressure
EE

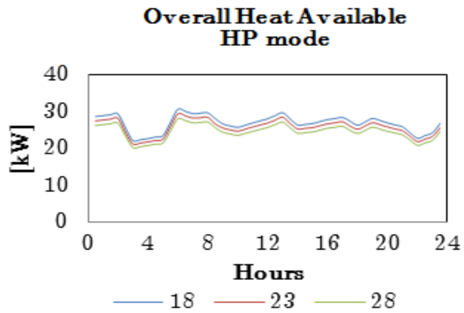

Total COP
at 28 °C return temp
[COP]
0
2
4
6
8
0 4 8 12 16 20 24
Hours
Standard
High Pressure
EE

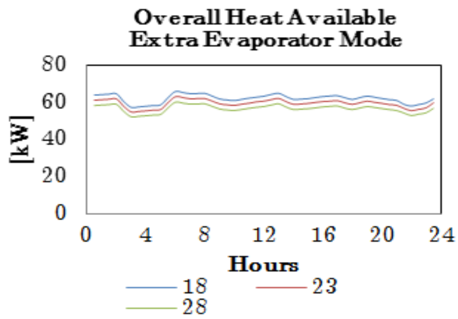
Average Spring/Fall values

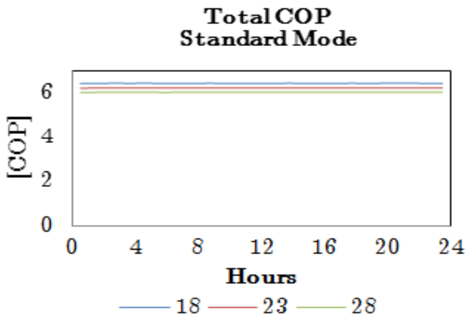
	18			23			28		
	STD	HP	EE	STD	HP	EE	STD	HP	EE
Power [KW]	4.25	10.77	23.05	4.25	10.77	23.06	4.25	10.77	23.06
GCI [kW]	2.94	24.90	56.17	2.94	24.90	56.17	2.94	24.93	56.16
GCII [kW]	2.02	2.87	6.83	1.11	1.73	4.16	0.30	0.53	1.33
Ref COP [-]	5.27	2.08	0.97	5.27	2.08	0.97	5.27	2.08	0.97
Tot COP [-]	6.44	4.66	3.72	6.22	4.56	3.60	6.03	4.45	3.48

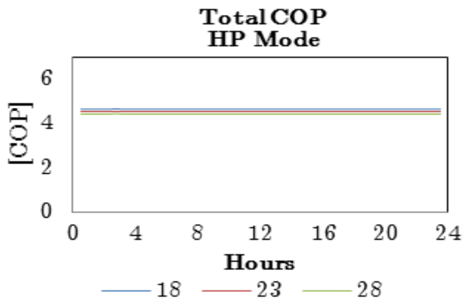
Winter

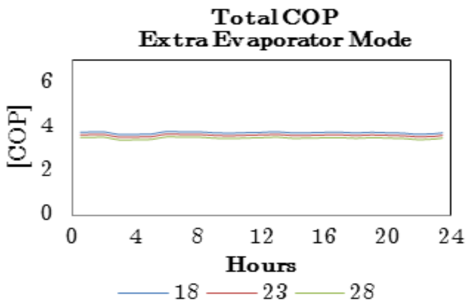


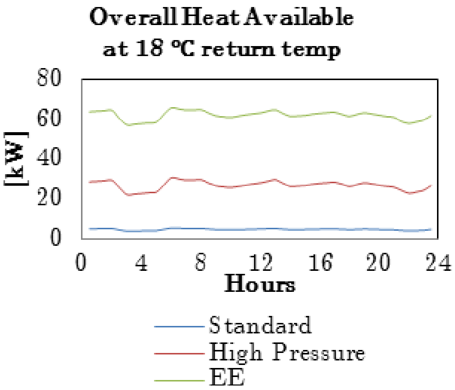


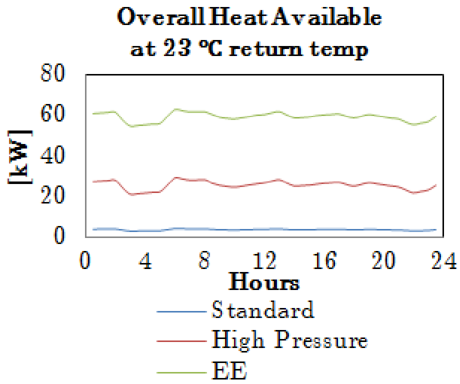


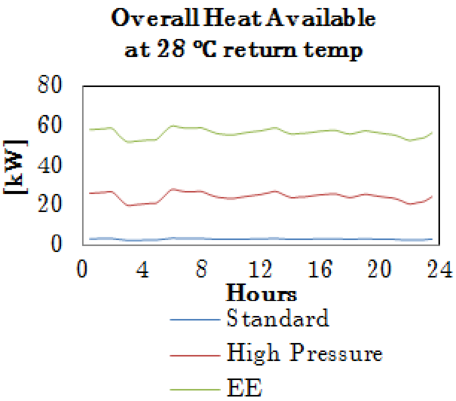


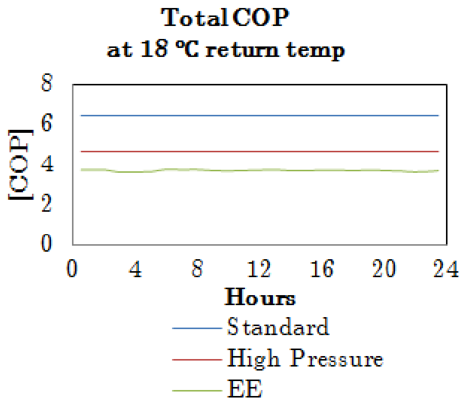


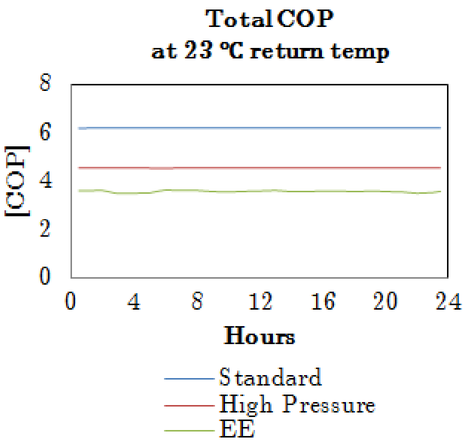


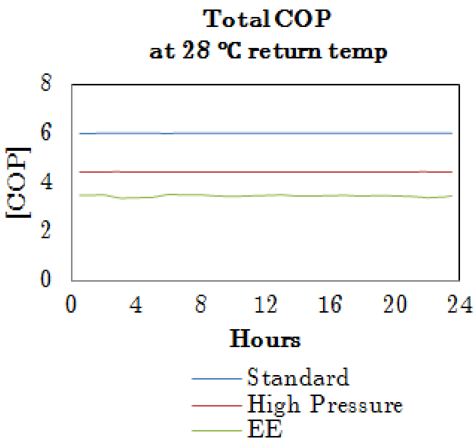

Overall Heat Available
at 18 ºC return temp
[kW]
Hours
Standard
High Pressure
EE


Overall Heat Available
at 23 ºC return temp
[kW]
Hours
Standard
High Pressure
EE


Overall Heat Available
at 28 ºC return temp
[kW]
Hours
Standard
High Pressure
EE


Total COP
at 18 ºC return temp
[COP]
Hours
Standard
High Pressure
EE

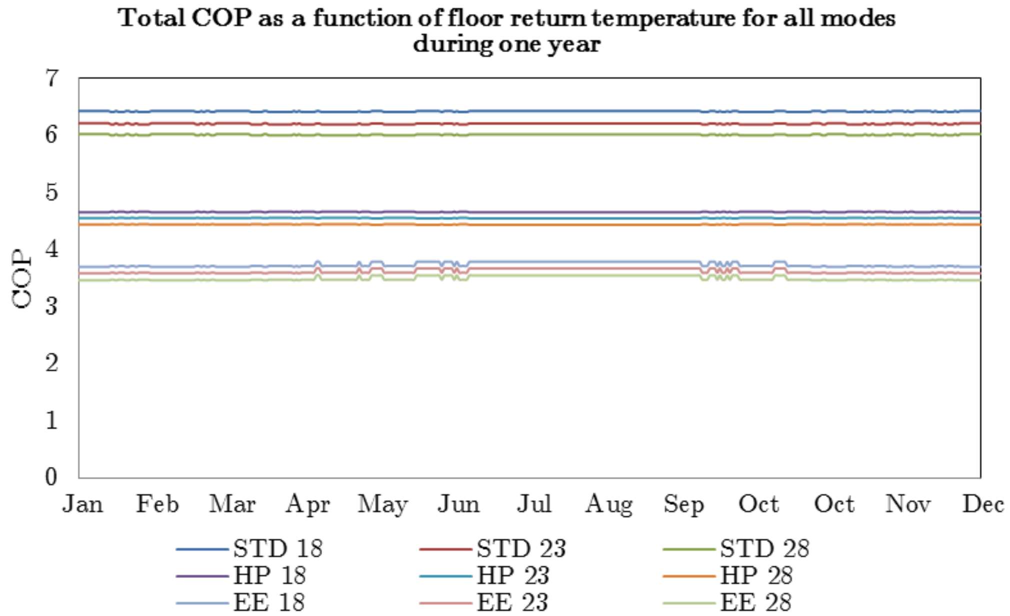

Total COP
at 23 ºC return temp
[COP]
Hours
Standard
High Pressure
EE

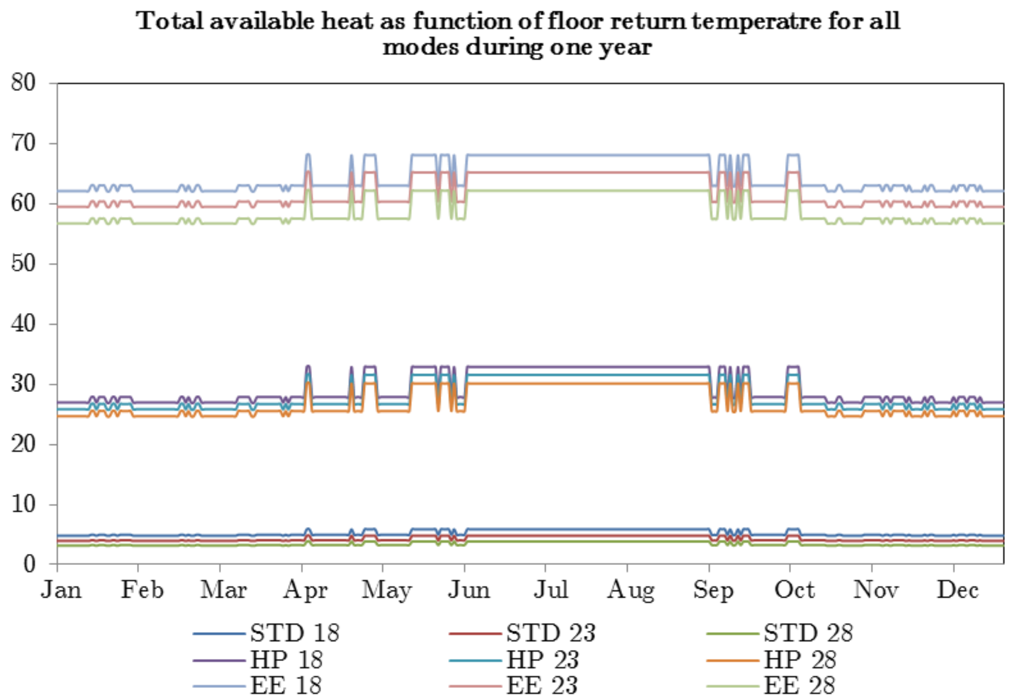

Total COP
at 28 ºC return temp
[COP]
Hours
Standard
High Pressure
EE

Average Winter values

	18			23			28		
	STD	HP	EE	STD	HP	EE	STD	HP	EE
Power [kW]	4.14	10.49	22.76	4.14	10.49	22.76	4.14	10.49	22.76
GCI [kW]	2.86	24.20	55.46	2.86	24.18	55.47	2.86	24.20	55.47
GCII [kW]	1.96	2.79	6.74	1.08	1.68	4.11	0.29	0.51	1.31
Ref COP [-]	5.27	2.08	0.96	5.27	2.08	0.96	5.27	2.08	0.96
Tot COP [-]	6.43	4.66	3.71	6.22	4.56	3.59	6.03	4.45	3.47

D. Annual estimations results





Annual estimation results for all strategies at 18 °C return temperature

Strategy 3			
		Case 1	Case 2
Energy [kWh]	Work	53 869	78 889
	Electrical Heat Needed	0	0
	Heat Available	95 113	175 393
	Heat Wasted	25	51
	Total Power Consumption	53 869	78 889
Efficiency [-]	SPF	5.61	4.85
Mode Frequency	Standard	79.2 %	56.9 %
	HP	18.6 %	29.9 %
	EE	2.2 %	13.2 %

Strategy 4			
		Case 1	Case 2
Energy [kWh]	Work	53 619	77 446
	Electrical Heat Needed	0	0
	Heat Available	95 097	175 344
	Heat Wasted	8	2
	Total Power Consumption	53 619	77 446
Efficiency [-]	SPF	5.64	4.94
Mode Frequency	Standard	77.1 %	45.6 %
	HP	21.9 %	48.2 %
	EE	0.9 %	6.2 %

Strategy 5			
		Case 1	Case 2
Energy [kWh]	Work	55 286	81 123
	Electrical Heat Needed	0	0
	Heat Available	95 116	175 353
	Heat Wasted	27	10
	Total Power Consumption	55 286	81 123
Efficiency [-]	SPF	5.47	4.71
Mode Frequency	Standard	90.4 %	74.7 %
	HP	0.0 %	0.0 %
	EE	9.6 %	25.3 %

Strategy 6			
		Case 1	Case 2
Energy [kWh]	Work	53 492	75 215
	Electrical Heat Needed	16	4 267
	Heat Available	95 048	171 045
	Heat Wasted	-40	-4 297
	Total Power Consumption	53 508	79 482
Efficiency [-]	SPF	5.65	4.81
Mode Frequency	Standard	75.9 %	37.8 %
	HP	24.1 %	62.1 %
	EE	0.0 %	0.0 %

Strategy 7			
		Case 1	Case 2
Energy [kWh]	Work	53 572	76 735
	Electrical Heat Needed	0	0
	Heat Available	95 217	175 433
	Heat Wasted	129	90
	Total Power Consumption	53 572	76 735
Efficiency [-]	SPF	5.64	4.98
Mode Frequency	Standard	75.9 %	38.0 %
	HP	24.1 %	60.6 %
	EE	0.0 %	1.5 %