Master's thesis

NTNU
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Investigation of R744 refrigeration system for supermarkets and the possibility of integrating a cold thermal energy storage

Master's thesis in Energy and Environment
Supervisor: Armin Hafner
June 2021

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Preface

This report summarizes work done for the master thesis spring 2021. The report is the continuation of the preliminary project written in autumn 2020. Both the preliminary project and master thesis are written at the Norwegian University of Science and Technology at the Department of Energy and Process Engineering.

The topic of the report is CO2 refrigeration systems for supermarkets and the possibility of implementing a thermal storage. The topic is chosen based on work for both the preliminary project and the master thesis. The focus of the preliminary project was to gain knowledge about the systems to be investigated for the master thesis and an understanding of the programs that are used for simulations. Simulations and calculations are performed during the master thesis.

I would like to thank my supervisor Professor Dr.ing Armin Hafner at NTNU for good guidance. I would also like to thank PhD student Lolanda for supplying many good articles and information about the subject, Muhammad Zahid Saeed for teaching the modelling software Dymola and simulation software DaVE, and Dr. Ángel Álvarez Pardiñas for good help. Finally, I would like to thank my family and friends. In a year with Covid where everything is different has been hard. Having a close group of friends to meet for lunch every day and on the weekend has been very important and appreciated.

Abstract

This report summarizes work done for the master thesis spring 2021. The report is the continuation of the preliminary project written in autumn 2020. The topic of the report is CO2 refrigeration systems for supermarkets and the possibility of implementing a thermal storage. The system to be investigated is an existing CO2-only refrigeration unit. A simplified version of the existing system will be the basis for the simulation model that will be developed. This system will be referred to as supermarket 1.

The topic is chosen based on work for both the preliminary project and the master thesis. The focus of the preliminary project was to gain knowledge about the systems to be investigated for the master thesis and an understanding of the programs that are used for simulations. Simulations and calculations are performed during the master thesis. During the preliminary project, a thorough literature review of CO2 refrigeration systems and thermal storage was conducted. This has been added to the current report and later both improved and expanded.

The report will look at the possibility of implementing a storage for the AC unit and the MT evaporators. Implementation and operation of the storage will be illustrated and explained. It will also be looked at how the implementation of the storage will affect the rest of the system, both size and cost will be considered.

Important data from the refrigeration system to be investigated are unfortunately not received until after delivery of the master thesis. There were some problems with the data logs. There was only possible to obtain data from the winter months and not summer months (that are most important to look at). Data from a second supermarket was therefore collected. Data is collected from supermarket 2 which has data for an entire year. Supermarket 2 has a little bit different system solution than supermarket 1. However, one can assume that the loads on the stores will be similar during the summer months and that supermarket 2 will be representative for the loads on supermarket 1.

Assuming that one install equal/bigger amount of storage capacity as the capacity of one of the compressors it can be possible to reduce the number/size of compressors for the system. It also may be possible to reduce the size of the existing system (components, piping, size of the gas cooler, etc). Simplified calculations show that the cost of the storage can be lower than the savings for the rest of the system. Thereby integrating a storage will be profitable based on the calculations. However, one has to take into account that these calculations are simplified, and further calculations will be needed.

Sammendrag

Denne rapporten oppsummerer arbeidet som er gjort for masteroppgaven våren 2021. Rapporten er en videreføring av forprosjektet skrevet høsten 2020. Temaet for rapporten er CO2-kjølesystemer for supermarkeder og muligheten for implementering av kulde lager. Systemet som skal undersøkes er et eksisterende CO2-kjølesystem. En forenklet versjon av det eksisterende systemet vil være grunnlaget for simuleringsmodellen som skal utvikles.

Tema er valgt ut fra arbeid for både forprosjektet og masteroppgaven. Fokuset for semesteroppgaven var å få kunnskap om systemene som skulle undersøkes i masteroppgaven og forståelse av programmene som skal brukes til simuleringer. Simuleringer og beregninger er utført under masteroppgaven. I løpet av forprosjektet ble det gjennomført et grundig litteratursøk om CO_2 – kjølesystemer og termisk lager. Dette er lagt til i den nåværende rapporten og senere både forbedret og utvidet.

Rapporten vil se på muligheten for å implementere termisk lager for ventilasjons aggregatet og MT fordampere. Implementering og drift av lageret vil bli illustrert og forklart. Det vil også bli sett på hvordan implementeringen av lageret vil påvirke resten av systemet, både med tanke på størrelse og kostnad.

Viktige data fra kjølesystemet som skal undersøkes mottas dessverre ikke før etter prosjektets slutt. Dette skyldes problemer med logging av data. Det var derfor bare mulig å skaffe data fra vintermånedene og ikke sommermånedene (som er de viktigste å se på). Data er hentet fra en annen butikk som har logge data for et helt år. Butikk nummer 2 har litt annerledes systemløsning. Man vil likevel kunne anta at lasten på butikken vil være noenlunde lik i sommermånedene og at butikk nummer 2 derfor er representativ for lasten på butikk nummer 1.

Forutsatt at man installerer lager kapasitet som er tilsvarende/større enn kapasiteten til en av kompressorene, kan det være mulig å redusere antall/størrelsen av kompressorer for systemet. . Det vil også kunne vøre mulig å redusere størrelsen på det eksisterende systemet (komponenter, rør, størrelse på gasskjøler etc). Forenklede beregning viser at lager kostnadene kan være mindre enn besparelsen for resten av systemet. Dermed vil integrering av lager være lønnsomt basert på beregningene. Man må imidlertid ta i betraktning at disse beregningene er forenklede, og ytterligere beregninger vil være nødvendig.

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Abbreviations

TES	Thermal energy storage
CTES	Cold thermal energy storage
LT	Low temperature
MT	Medium temperature
MP	Medium pressure
HP	High pressure
HTF	Heat transfer fluid
GHG	Greenhouse gas
LPR	Low pressure receiver
HPR	High pressure receiver
HPV	High pressure valve
FBV	Flash gas bypass valve
COP	Coefficient of Performance
GC	Gas cooler
V	Volume
CO_2	Carbon dioxide
HEX	Heat exchanger
AC	Air condition
AHU	Air handling unit

Nomenclature

T	Temperature [K]
P	Pressure [Bar]
ρ	Density [kg/m^3]
Q	Energy need [kW]
$\dot{m}$	massflo [kg/s]
C_p	Specific heat [
Δh	Enthalpy difference
V	Airflow [m^3/h]
$\vec{g}$	Gravitational acceleration [m/s^2]
$\overrightarrow{F_M}$	Driving force [N]
Δz	Height difference [m]
$\dot{V}$	Airflow [m^3/s]
Δz	Height difference [m]

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Chapter 1. Introduction

1.1 Background

There is a large transition in supermarket refrigeration with a strong focus on reducing the energy demand and the installation cost. Highly efficient system configurations with R744 are introduced in various locations throughout Europe; however further improvements are necessary and possible, for example with the use of ejector-based expansion work recovery, pivoting compressor arrangements, and the implementation of local cold storages, etc.

Supermarket refrigeration systems have a significant energy demand to preserve the quality of valuable food products. Thus, several approaches to reduce the energy demand, and to solve other challenges a supermarket refrigeration system faces have been considered in the past years. One of the main challenges that have been addressed is the temperature instability of the food products in the display cabinets/freezers in general and during the defrost period. Other issues that have been considered are high electricity costs during peak demand, due to the grid dependency of the refrigeration unit, and the power consumption irregularity of the system.

Multi-ejector expansion modules, intended as a substitute for standard high-pressure control valves (HPV), were designed by SINTEF/Danfoss and experimentally investigated at the SuperSmart-Rack test facility (Varmeteknisk laboratory, Trondheim). The implementation of a low-pressure lift multi ejector block for air conditioning (AC) production has also been tested in the past at NTNU's laboratory, showing its potential for reducing the power consumption of integrated CO2 refrigeration systems. The pivoting compressor principle complements these advances and allows to make the aforementioned solutions cost-effective. It is based on switching compressors between the different suction groups so that fewer units are needed to meet the requirements in the different seasons.

On the other hand, the implementation of local cold storage in combination with the R744 refrigeration system can have enough benefits which enable an implementation in the market. As cold thermal storage reduces the peak load and allows for shifting it to periods with low electricity cost or high electricity production with renewables (e.g. solar panels). These units can also lead to a radical downsizing of the compressor packs. The first approach to thermosyphon-driven, local cold storage has been experimentally investigated at NTNU's laboratory, but the concept needs to be refined and deeply investigated related to the impact on the total system architecture.

1.2 Task and scope

Task: Investigation of R744 refrigeration system for supermarkets and the possibility of integrating cold thermal energy storage.

The following tasks will be considered for the preliminary project:

1. Literature review on current commercial refrigeration systems and cold thermal energy storage (CTES) options
2. Processing of data supplied from existing supermarket refrigeration system
3. Operation and implementation strategies for the CTES
    a. Charging
    b. Discharging
4. Describe architecture of storage and implementation in the system (air conditioning, freezers..)
5. Investigate various strategies for peak shavings, constant compressor power operation and load/demand adaption via active CTES units.
6. Further development of Modelica model representing supermarket units
    a. Implementing dynamic data (load and temperature profiles)
    b. Implementing storage devices
7. Evaluate investment costs: what could be the cost rage of such systems, how will the implementation of CTES within the refrigeration system affect the total cost of ownership for the supermarket?
8. Data processing and analysis of the modelling and experimental results. Write a report including a discussion, summary and further work chapter.
9. Draft version of a Scientific publication related to the main results

1.2.1 Scope of the work

The topic is chosen based on work for both the preliminary project and the master thesis. The system to be investigated is an existing CO2-only refrigeration unit. A simplified version of the existing system will be the basis for the simulation model that will be developed. There will be conducted a literature review of CO2 refrigeration systems and thermal storage. The PCM to be used for the storage is water and therefore the main focus for the literature review has been CTES with water. It will be looked at the possibility of implementing a storage for AC (Air condition) and MT evaporators. The implementation and operation of the storage will be explained. It will also be looked at how the implementation of the storage will affect the rest of the system, both size and cost will be considered.

Some changes have been made to the project task during the semester. The task description was revised after the literature research and meetings with people involved in the project and by request from the supervisor. The main idea was to implement the storage above the MT evaporators as described in the preliminary project. Then one storage would cover one cabinet. The main focus was changed to the implementation of one big storage for the AC unit. During the semester one could see that there was a big potential for this solution. Later the MT evaporators were also added to the storage loop (but with a different solution than originally).

A simulation model has been developed in Dymola. The model is developed based on an existing system given by the supervisor and literature research. The storage was supposed to be built in the lab at Varmeteknisk NTNU in autumn 2020, but this was postponed. Therefore data are based on literature and data received from supervisor and PhD students.

Simple investment analysis will be performed on the new system. For the analysis, there will be considered cost of the CTES and also reduced costs due to reduction of the main system (reducing number or size of compressors, removing the glycol loop, etc). Economic data regarding costs of the existing system was supposed to be received from the company that built the supermarket. However, this was not received before the delivery of the report. Prices from the contractor that will be building the storage in the lab have been requested. The results from the investment analysis were sent earlier in the semester, but the response was unfortunately not received before delivery. The investment analysis is therefore simplified.

Chapter 2. Literature review

This chapter will give an overview of the most important topics related to the subject. It will give information about research and technology available at present. It starts with the development of CO2 systems and cold thermal storage and then the implementation of cold storage in the refrigeration systems. The literature was started during the preliminary project autumn 2020. The literature review from the preliminary project has been added to the current report and later both improved and expanded.

2.1 Energy consumption, emissions and refrigerants

Supermarket has a vital role in the modern society and the number of stores in the world is increasing. Figure 1 shows the increase in number of stores from the year 2000 to 2011. [1] Some of the main reason for the growth is: increased urbanization, female labour and emerging middle class. People spend less time cooking at home and people are buying more prepared foods. [2] Norway is on the top of the statistics when it comes to number of stores per inhabitants. In industrialized countries supermarkets consumes 3-4% of the annual electricity production. [1] The average consumption for a Norwegian supermarket is according to Enova 630kWh/m^2. For new passive houses this number is considerably lower with 130 kWh/m^2. [3]

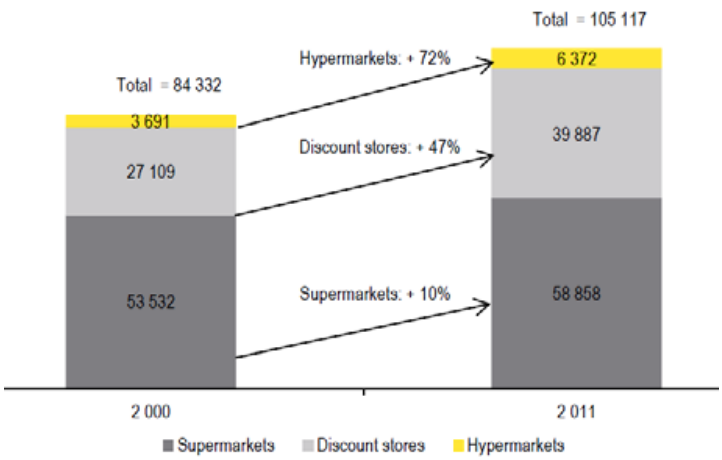


Figure 1: Growing number of stores [4]

In a heat pump the refrigerant transports energy by circulating and changing phase. When it comes to heat pumps and cooling machines the selection of refrigerant is the most important choice. [5] Which refrigerant is the best fit will depend on several factors like: thermal properties, cost, and availability. One also has to consider the refrigerant based on environment- and safety properties. Refrigerant used in heat pumps are divided into two groups: Synthetic and natural. [6]

Figure 2 shows the GHG refrigerant consumption in EU. One can see that commercial refrigeration stands for a big percentage of the total consumption. During the last decades several measures has been implemented to reduce the impact on the environment. Under the Montreal protocol an international treaty was made to protect the ozone layer. The treaty stated that all ozone depleting substances should be phased out by 2030(2020 for developed countries). Also the Kigali amendment aim to phase down the use of HFC, leading to increased use of natural refrigerants. [7] In addition to the Montreal protocol EU has implemented its own F-Gas Regulation. The purpose is to reduce the use of HFCs by 79% by 2030. By 2022 there will be prohibited to use HFCs with a GWP higher than 150 in facilities larger than 40kW/11TR. There are exceptions for cascade facilities where the refrigerant in the primary circuit can have a GWP of 1500. [7]

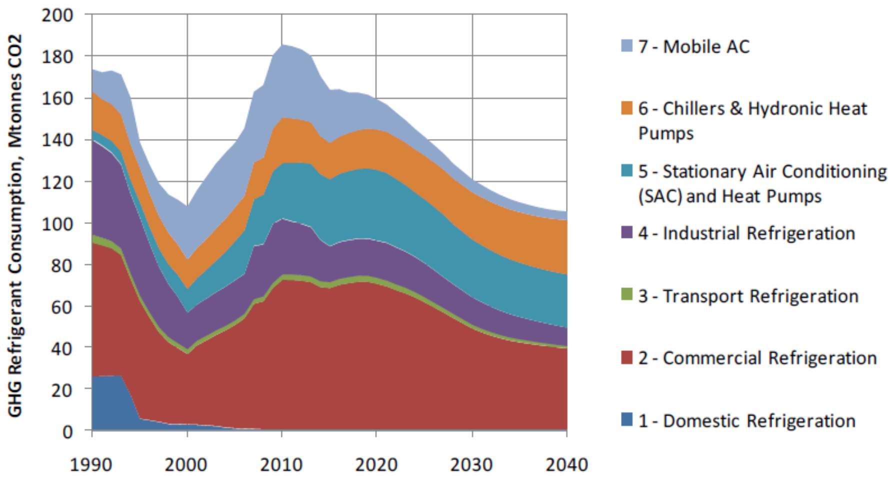


Figure 2: GHG refrigerant consumption EU [8]

When the existing systems have to be replaced there are two possible solutions: One could continue developing existing systems using new HFCs with lower GWP or one could develop new systems using natural working fluids like ammonia, propane and CO_2. Requirements in the future will probably become even stricter. Choosing to develop systems with refrigerants that has low/zero GWP is therefore a safe option that can give benefits in the future.

2.2 CO_2 heat pumps

CO_2 heat pumps has the last several years become the dominant solutions for refrigeration systems in stores in Scandinavia. It was rediscovered by Gustav Lorentzen a professor at NTNU. Studies conducted by Lorentzen and the simultaneous phase out of ozone depleting refrigerants lead to a renewed interest in CO_2 for use in refrigeration systems. [9]

2.2.1 CO_2 as refrigerant

Carbon dioxide is a natural working fluid and can be found in the earth's biosphere. It has no undesirable environmental effects and when used as a refrigerant it has a GWP factor of zero. [10] The refrigerant has good thermodynamic properties. It has a high specific heat capacity and heat transfer abilities. [11] It has low critical temperature compared to other refrigerants like HFC, HFO, R717 and hydrocarbons. The practical upper limit for condensation is 28 °C. The refrigerant has high critical pressure and low-pressure ratio. Compared to HFC, CO_2 typically has 5 to 10 time's higher pressure. The refrigerant has high critical pressure and low-pressure ratio. [10] Table 1 presents the fundamental fluid properties of R744.

Table 1: Properties of R744 [10]

Properties of CO_2	
Chemical formula	CO_2
Refrigerant number	**R744**
Molecular weight	**44.01**
Critical temperature	**31.1℃**
Critical pressure	**73.8 bar**
Triple point	**- 56.6℃, 5.2 bar**
Concentration in air	**Ca. 410-420 ppm**
Working pressure	**30 to 120 bar**

The use of CO_2 for freezing and cooling has in the last years become more popular. The reason for this is that the freezing process requires a lot of energy. From Figure 3 one can see that CO_2 has high volumetric performance at low temperatures. [5] The low-pressure ratio gives a high compressor efficiency. High gas density makes it possible to have smaller compressors and smaller dimensions on heat exchangers and pipes. [11] Because of the high working pressure there is need for adapting the components. [12] It is also important with pressure testing of the facility.

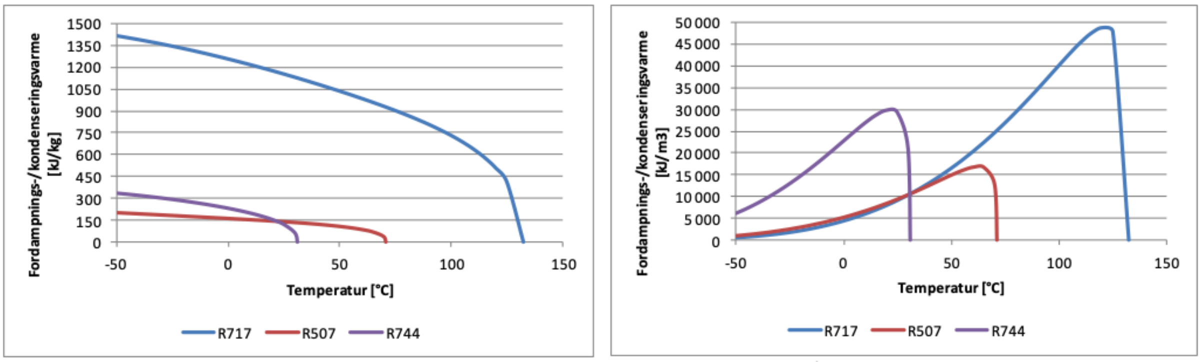


Figure 3: Volumetric performance of CO_2 [5]

If one has an evaporation temperature of -40 °C and a condensation temperature of 0°C the pipe dimensions can be reduced by 40% compared to facilities with ammonia. This is due to less volume flow. Compared to the two other refrigerants in Figure 3 the pressure conditions are small and the compressor efficiency will be very good. [5]

Safety considerations

"Kulde-og varmepumpenormen" give classifications for pressurized equipment based on the danger level. The higher the category the stricter requirements are for documentation of the equipment. Pressurized equipment, containers, accessories, and compilations are categorized from I to IV. Pipe systems are categorized from I-III. The degree of danger with respect to the refrigerant are divided into two categories: Group 1 which is the dangerous refrigerants and group 2 which is other mediums. CO_2 is in group 2. [13]

The refrigerant R744 is non-flammable and nontoxic but can lead to suffocation. According to health and safety ASHRAE gives classifications for the different refrigerants and CO_2 belongs to group A1(non-flammable and slightly toxic). When CO_2 is inhaled the PH-value in the blood will be affected and poisoning of CO_2 will therefor occur at significantly lower concentrations than for suffocation. [13] Other characteristics that are important to remember when using CO_2 is the fact that it is heavier than air. When using CO_2 as refrigerant it is necessary with detectors that measures CO_2 concentration. Oxygen detector can also be used but should be avoided. The detector should be placed by the floor because the refrigerant is heavier than air. [13]

2.2.2 Heat pump cycle CO_2

In Figure 4 one can see a simple CO_2 heat pump cycle with the main components: Evaporator, gas cooler, compressor, expansion valve, heat exchanger and receiver. Because most heat demands have a temperature requirement that is higher than 30 °C, the CO_2 heat pump operates in what is called a trans critical process. [11] A transcritical cycle (also called supercritical) is a cycle with a maximum pressure higher than the critical pressure. [14]. Heat is absorbed at subcritical pressure and heat is rejected at overcritical pressure. [10]

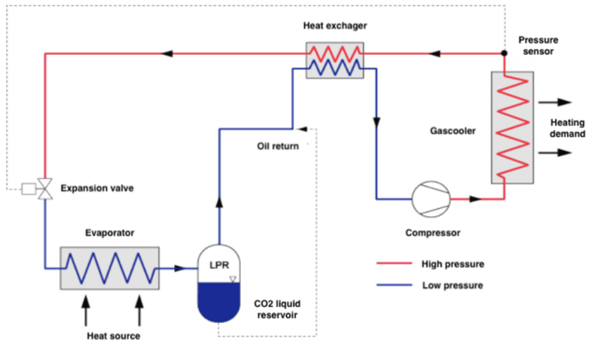


Figure 4: Principle of CO_2 Heat Pump [11]

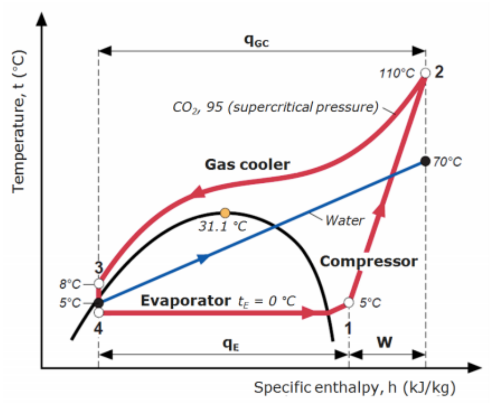


Figure 5: Temperature-/enthalpy diagram CO_2 heat pump [6]

In Figure 5 one can see a temperature/-enthalpy diagram for a simple heat pump cycle. The refrigerant evaporates in the evaporator and enters the LPR (low pressure receiver). The LPR is implemented to allow the use of liquid-filled evaporators. In the compressor the refrigerant is compressed to transcritical pressure above the critical point. CO_2 have high density at low temperatures. This gives a high compressor capacity compared to other refrigerants. When designing a heat pump the superheat is normally set to 8K but in reality, the superheat will be a lot higher about 30K. The reason for this is that pipes are often not insulated.

In a CO_2 heat pump the heat exchanger is called gas cooler not a condenser. The CO_2 will not condensate like in a conventional heat pump process, the refrigerant is being cooled down over a big temperature area and we say that the temperature is “gliding”. In the transcritical process the refrigerant will not have a phase change in the way we normally have. The refrigerant will start as a gas and end up as liquid, but without precipitation of liquid along the way. [5] After the gas cooler the pressure is lowered and enters the HPR (high pressure receiver). In CO_2 systems there are higher expansion losses. This is possible to recover using different technologies like ejectors or expansion engines. This will be further elaborated in chapter 2.3.3.

2.3 Development of CO_2 systems for supermarket refrigeration

The use of transcritical CO_2 refrigeration systems is growing around the world. In Scandinavia almost 100% of new refrigeration systems developed for supermarkets use CO_2 as refrigerant. When Shecco started collecting their data in 2008 there were only registered 140 systems and all of them were in Europe. From the report published in September 2020 the new number is over 35 500 systems worldwide. Figure 6 shows the CO_2 refrigeration systems around the world as of 2020. For Europe 90% of the 29 000 refrigeration applications are in supermarkets. [7] The use of transcritical CO_2 systems for supermarket refrigeration are mostly used in colder climate countries, but are starting to grow also in warmer countries. [9]

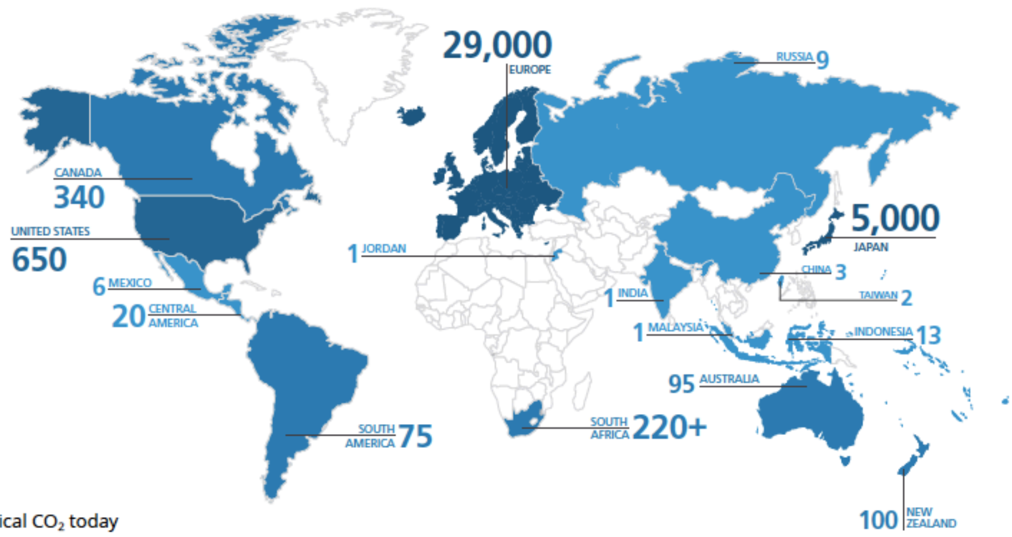


Figure 6: Use of Transcritical CO_2 systems worldwide published by Shecco 2020 [7]

Figure 7 shows the development in refrigeration systems using CO_2. It started in 1993 with the first subcritical system developed for colder climates. The last two decades there has been development of transcritical systems and implementation of parallel compressors and ejectors. These systems work better in warmer countries because one can reuse the expansion losses. The next few chapters in the report will present the development of the systems from 1. generation transcritical system to the 3. generation parallel compressor ejector system.

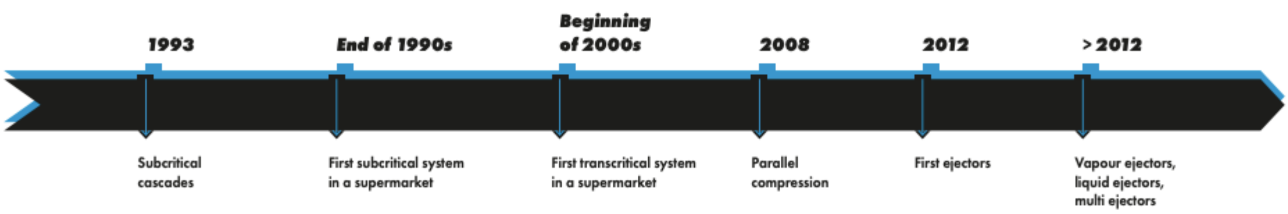
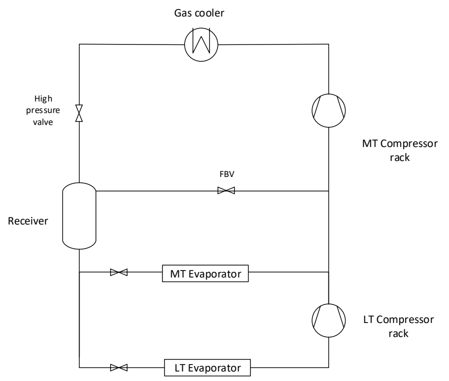


Figure 7: Development CO_2 systems [7]

2.3.1 1. generation Transcritical booster system

Figure 8 and Figure 9 shows a simple booster system for supermarket application. The first stage is the LT (low-temperature) stage. Refrigerant passes through the LT evaporators and are then compressed by the LT compressor. Then refrigerant from the LT compressors mixes with refrigerant from the MT (medium-temperature) evaporators. The refrigerant flow then enters the MT compressors and are compressed to the HT (high temperature) stage. The refrigerant passes through the gas cooler and enters the receiver. The refrigerant in the receiver will be both liquid and gas. From the receiver the liquid refrigerant is fed to the evaporators. There is a flash gas bypass valve (FBV) connected in a pipe from the receiver to the MT compressors. Flash gas from the receiver is fed to the MT compressors. The amount of flash gas will depend on the temperature in the gas cooler.



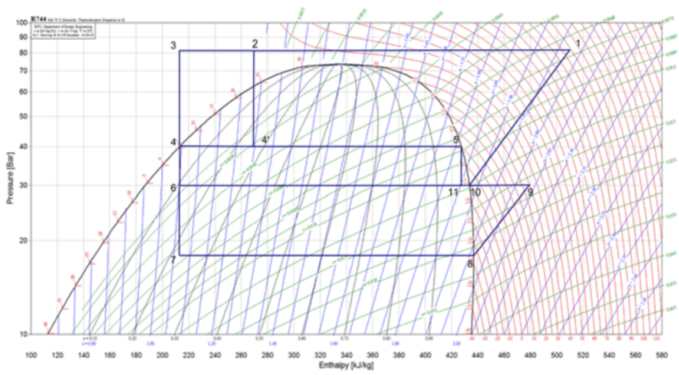


Figure 9: 1. generation Transcritical booster system

Figure 8: Log P-h diagram 1. generation booster system [18]

Subcritical and transcritical mode

Heat pumps with R744 can be run in transcritical and subcritical mode. The subcritical process is achieved when the outside temperature is below critical temperature. In warmer countries the temperature outside is high and the system will be run in transcritical mode for larger periods than for colder countries. There will be produced a lot of flash gas which leads to low efficiency for the system.

According to Gullo et.al [15] CO_2 systems perform worse than HFC systems in mild/warm climates. His simulations shows that the CO_2 system has higher peak demand in the hot season and lower peak demand in the cold season compared to a cascade system with r134a. Calculations done by Gullo et al. [16] shows that estimated mass flow of flash gas for transcritical running mode is 45% of the total mass flow.

2.3.2 2. generation Parallel compression

In a CO2 system with two stage throttling the refrigerant will after the first throttling be both liquid and vapor. The amount of flash gas coming from the gas cooler will increase with rising outdoor temperature. The amount of flash gas compressed by the MT compressor will be very high when the outdoor temperature is high, resulting in low efficiency for the system. By inserting a parallel compressor one can take advantage of the vapor from the flashing and thereby enhance the COP of the system. [9] Figure 10 and Figure 11shows a parallel compressor system.

The refrigerant in the receiver feeds the evaporators with liquid as in the first-generation system. The difference is the distribution of the flash gas. The parallel compressor sucks off the flash gas from the receiver and compresses it from the intermediate pressure level to the high-pressure level. The refrigerant then mixes with the refrigerant coming from the MT compressors.

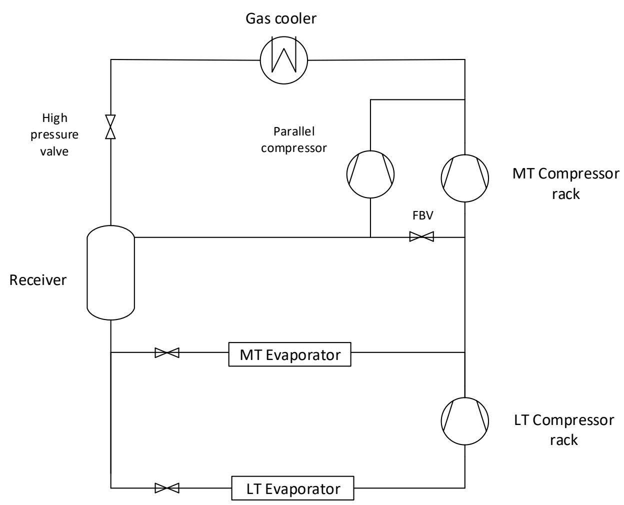


Figure 10: 2. generation parallel compressor system [17]

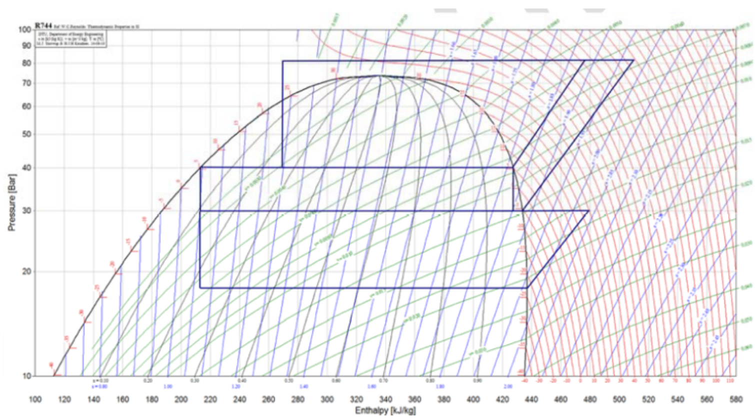


Figure 11: Log P-h diagram 2.gen parallel compressor [18]

2.3.3 3. generation parallel compressor and ejector system

The 3. generation of CO_2 refrigeration systems is more suited for warmer climates than the 1 and 2. Generation. This solution takes advantage of the expansion loss by inserting both parallel compressor and ejector or expander. The use of ejectors and expanders in heat pumps is a way of enhancing the efficiency of the heat pump. When using CO_2 as refrigerant the expansion loss are higher than with other refrigerants and it is therefore normal to use an ejector.

Figure 12 shows an example of a 3. generation CO_2 system. Ejectors allows for operation of the system with flooded evaporators. In DX evaporators part of the evaporator has to be used for superheating of the refrigerant. With flooded evaporators there is no need for superheat. Flooded evaporators will lead to increased heat transfer and better efficiency of the system. It also allows for higher evaporator temperature. [19]

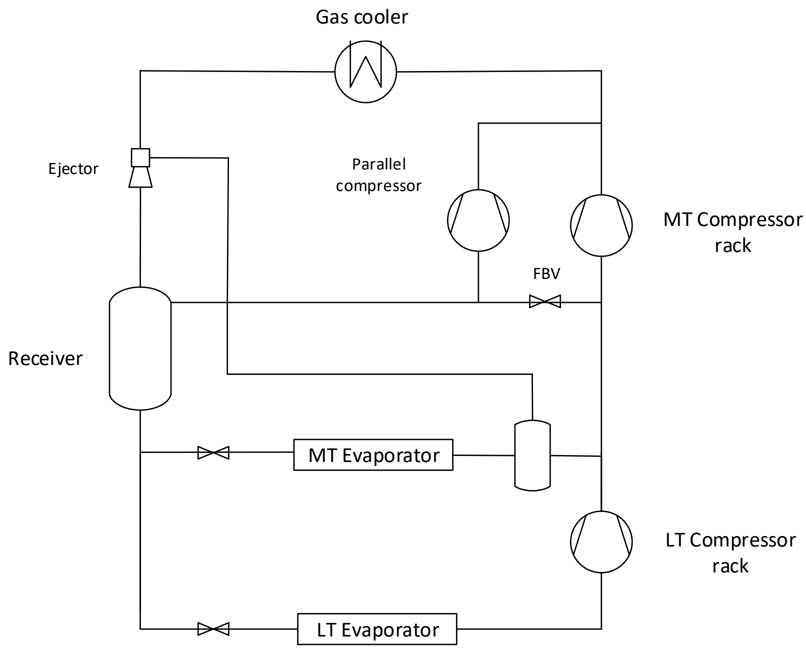


Figure 12: 3. generation parallel compressor and ejector system

Ejector

In order to retrieve some of the expansion work the throttle valve can be replaced by an ejector. The ejector takes advantage of the expansion energy in order to increase the compressors suction pressure. [10] By exploiting the pressure difference in the facility the ejector can suck off low pressure gas from the evaporator and increase the pressure of the gas. By inserting the ejector, the suction pressure is increased and the required effect is decreased. [11]

Figure 13 shows an ejector. The ejector will accelerate high pressure stream through a nozzle. This way the pressure energy is converted into velocity. A low-pressure region is created around the nozzle tip (where the velocity is highest). As Bernoulli's principle states: the pressure will decrease as the velocity increases. Energy from the motive stream is used to do work on the suction stream. In the low-pressure suction stream the pressure will increase and in the motive stream the pressure will decrease. [20]

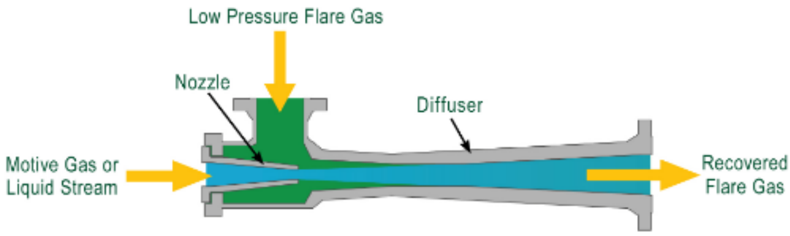


Figure 13: Ejector [20]

The ejector is connected after the receiver as shown in . The pressure difference between receiver and evaporator/compressor is about 7 to 10 bar and this can be utilized by the ejector. [21] There as several alternatives of how the ejector can be integrated in the system:

Alternative 1: Small amount of gas and high increase of pressure.
If there is a system with several compressors the ejector can receive gas from one compressor and send it into the other compressor. The expansion loss can then be exploited to increase the pressure and thereby give compressor number two gas with higher pressure.

Alternative 2: Large amount of gas and small increase in pressure.
Another alternative is for the ejector to take all of the gas and increase the pressure a little bit.

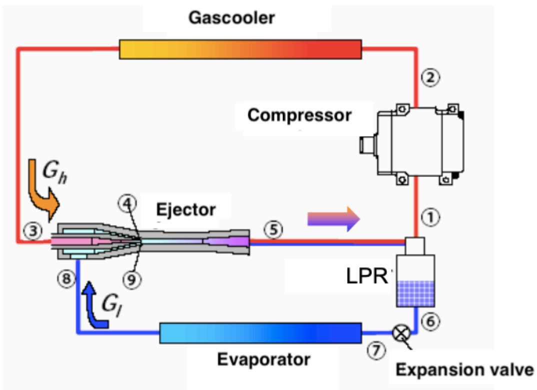


Figure 14: Ejector2 [80]

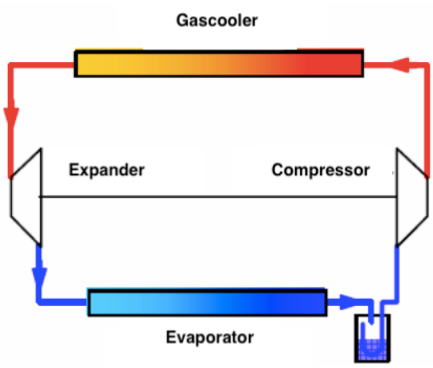


Figure 15: Expander [80]

Expander

An expander is used to recover the expansion work and thereby increase the COP. In difference to the ejector the expander has not been commercialized and are still under development. It exploits the expansion energy in order to produce mechanical work. It can be used to relieve the compressor engine. [10] Figure 15 illustrates how an expander is constructed. The expander can be mounted directly to the main compressor.

2.3.4 Integrated system solution

The state-of-the-art systems have more functions than only supplying refrigeration for the food. Several systems are combined into one integrated system. The systems can supply both refrigeration for the food, air-condition and also make use of the excess heat. Heat can be recovered to be used for space heating, heating of tap water, snow melting or a combination of several solutions.

Figure 16 shows an integrated system solution for Nordic countries by Armin Hafner. According to Hafner integrated solution are characterized by “The ability to provide most of the heating and cooling demands within a certain area or part of a building” [22]. The system has both air-conditioning (AC) and heat exchangers for heat recovery and snow melting. In case of heat demand, heat recovery has priority and the gas cooler can be bypassed. Integrating heat recovery into the system will increase the total efficiency of the system. It will also lead to reduced purchased energy for heating.

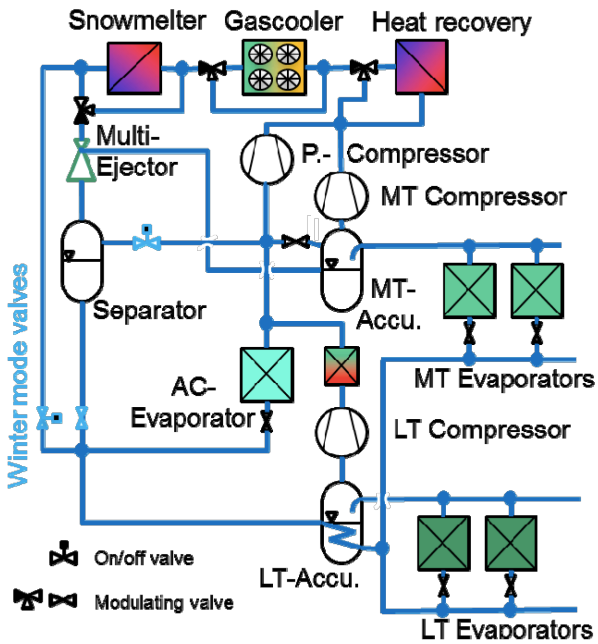


Figure 16: Integrated system solution for Nordic countries [22]

Winter mode: If the temperature drops below 5°C after heat exchangers rejecting heat, this will give problems for the rest of the system. If the temperature in the pipe going into the separator drops below the saturation temperature the pressure will collapse inside the separator. The solution to this is that if the outside temperature becomes too low the separator is bypassed. Refrigerant is supplied directly to the evaporators. [22]

2.4 Natural circulation system

One can divide between natural and forced circulation systems. The forced circulation loop requires a pump or compressor to drive the fluid flow. The natural circulation loop is a system where the fluid flow take place due to the density gradient caused by an imposed temperature difference. In the natural circulation loop the heat sink has to be located at a higher elevation than the heat source. The fluid condenses in the top heat exchanger and moves down. The fluid then evaporates in the bottom heat exchanger and rises up. Thereby the loop does not need a pump or compressor. [23]

Figure 17 shows a simple natural circulation system. The left leg with upward flow will have a density denoted ρ_h and the right leg with downward flow will have a density denoted ρ_c. The letters H and C stands for hot and cold. The hydrostatic pressure for location a and b for the system is then given by equation 1 and equation 2. Where H is the height in meters, ρ is the density in kg/m^3 and g is the gravitational acceleration in m/s^2.

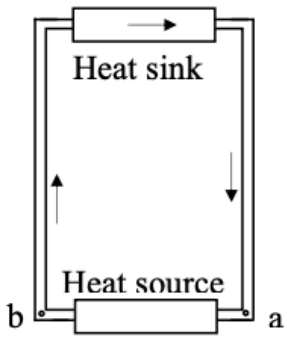


Figure 17: Natural circulation system [24]

$$P_a = \rho_c * g * H \tag{1}$$

$$P_b = \rho_h * g * H \tag{2}$$

A natural circulation system will have economic advantages due to elimination of pump/compressor. This will reduce investment cost and costs for maintenance and operation. With a natural circulation system there will also not be risk of pump failure or downtime. [24]

According to Yadav et. al the natural circulation loop will have lower heat transfer than the forced circulation loop. [23] The driving force for natural circulation will be lower than for forced circulation. The power rating will thereby be lower. One on the ways of increasing the driving force is to increase the height. According to Vijayan et.al the height for a natural circulation system is normally no more than 10 meters. One should also avoid having many u bends and elbows in the system [24]

Yadav et.al has performed experiments on a natural circulation loop with CO_2 as the refrigerant. The experiment loop is shown in Figure 18. The length of the heat exchangers is 1.6m and the height of the loop is ca 1.8m. Results from the experiment shows that the temperature difference decreases as the pressure in the system increases. This occurs due to increase in specific heat at increasing pressure. The heat transfer reached its maximum at a pressure of 90 bar.

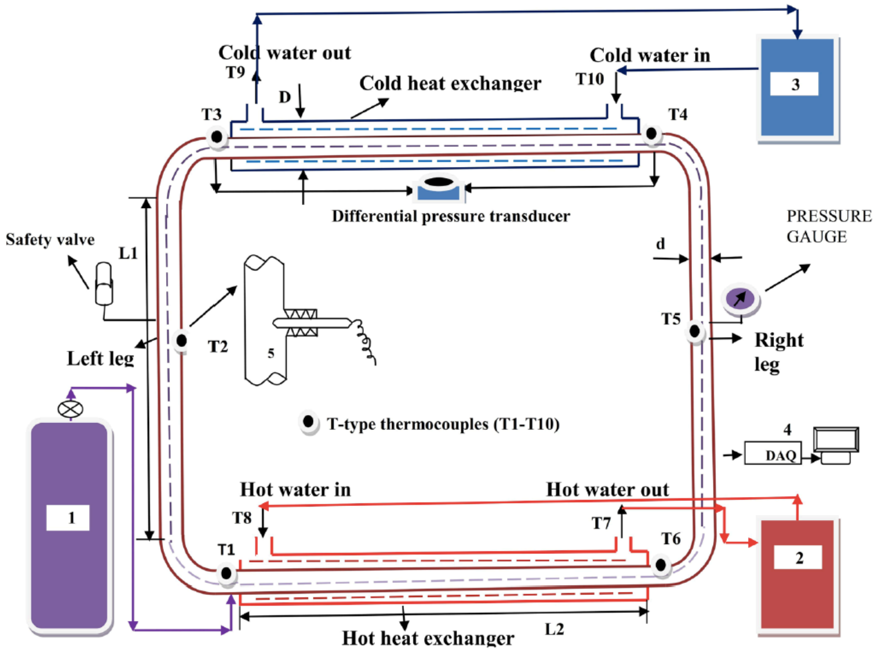


Figure 18: Natural circulation loop by Yadav et.al [23] (1)CO_2 reservoir, (2) Thermostatic bath for HHX,(3) Thermostatic bath for CHX, (4)Data acquisition system, (5) Enlarge portion of inside thermocouple arrangement

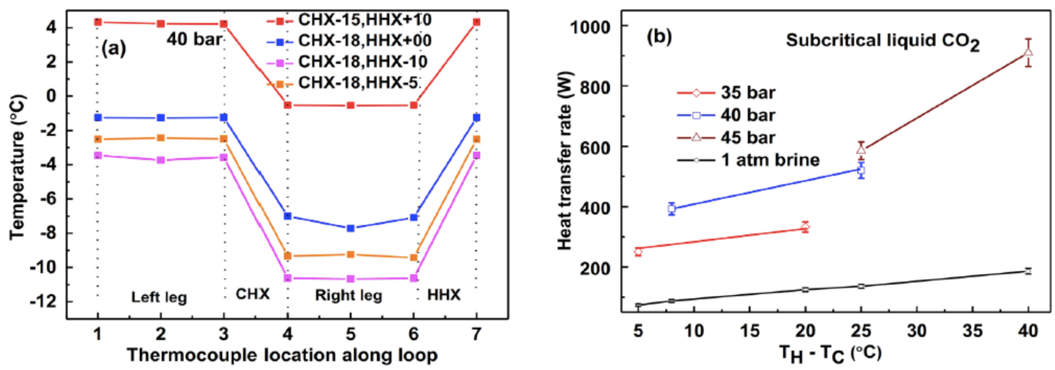


Figure 19:Results experiments Yadav et.al [23]

2.5 Cooling rooms and cooling load

Right temperatures are important to preserve the quality of the food. Refrigeration of food slows down the chemical and biological processes in food. Thereby preserving both taste and aesthetics. In refrigeration engineering one distinguishes between two types of rooms: rooms for chilled products and rooms for frozen products. According to Granrud in rooms for chilled product one is aiming for temperatures in the range 0 to +8 °C while for frozen products the temperature is below -18 to -25 °C. In this report the focus will be on rooms for chilled products. [25] According to Granrud the dimensioning temperature is 0 to +2 °C for chilled products like meat, fish and milk products and +6 to +8 °C for vegetables. [26] Sintef give guidance to the climate in cooling rooms. They divide between cold (0-5°C) and "warm" (5-15°C) cooling rooms [27]

Refrigeration load

The refrigeration load is the heat that is generated inside the refrigerator and the transmission into the refrigerator. The cooling capacity is defined by Eikevik as "the system ability to remove the heat and lift this heat from the temperature inside the refrigerated space up to the ambient temperature and release it at this level". The load and capacity have to be balanced. If the load is bigger than the capacity, then the temperature will increase. If the capacity is bigger than the load, then the temperature will decrease. If the temperature is still going down after desired temperature is reached, the capacity has to be adjusted by compressor control. [28] This will be explained further in chapter 2.5.

The refrigeration load will vary a lot over the day and year. There are several different parameters that influence the load. Most important parameters influencing the load [26]:

- Heat transmission
- Exchange of air
- Cooling of products
- Internal heat generation

There will be two periods of the day when the load is at its highest. The first peak is at the beginning of the day when goods are loaded into the store and the cabinets. This period is between 8am and 12 am. The second peak is when most people are buying groceries, which is between 3 pm and 6 pm. The system will run on base load most of the day. According to Selvnes the base load for CO_2 booster systems is approximately 20%. Figure 20 shows the assumed cooling load for an average day for a supermarket with a maximum load of 75kW. Cooling load for the system is 60kW and freezing capacity is 15kW. [29]

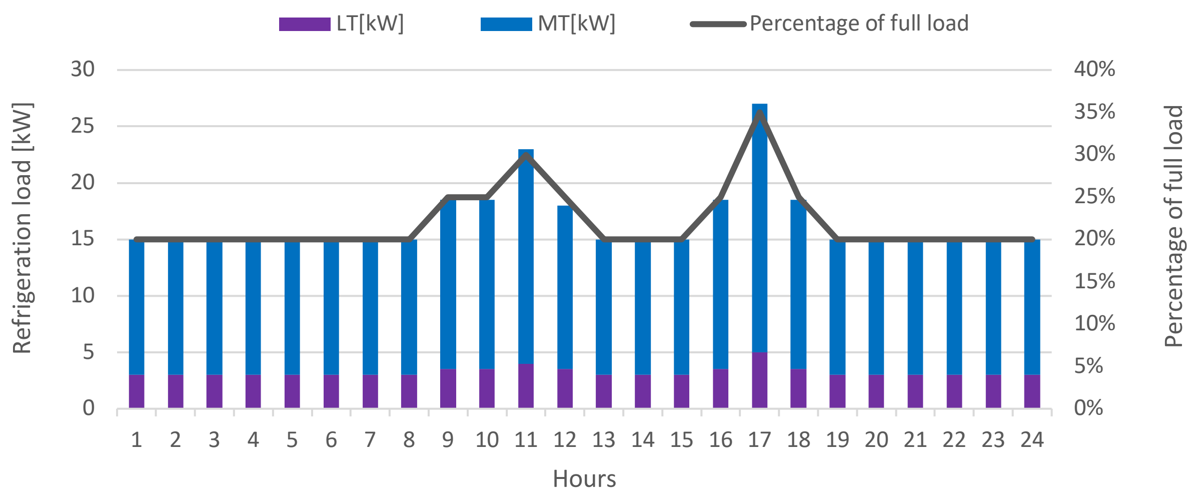


Figure 20: Refrigeration load of a supermarket with maximum load of 75kW [29]

Person load

Figure 21 shows the person load for a supermarket in USA [30]. The table has to be adjusted for different countries, but one can assume that the load will be similar with two peaks. In Norway most shops close at 11pm and open at 7am. Thereby the person load will be zero at night. Nevertheless, the table gives a similar load at day when comparing to Figure 20 which shows the refrigeration load. There will be two peaks during the day one around lunch time and one in the afternoon when people buy groceries after job.

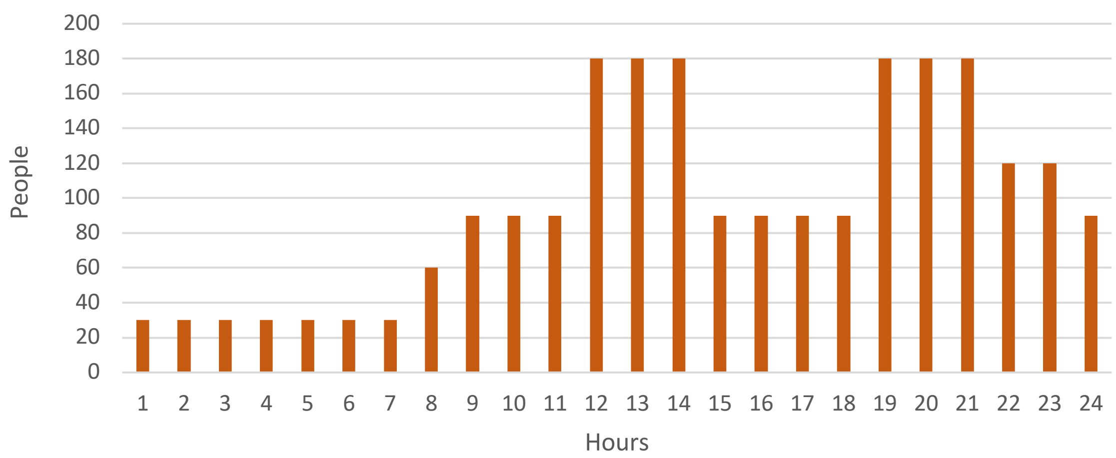


Figure 21: Schedule of people occupancy [30]

Total loads on the system

Figure 22 shows the total cooling and heating loads for a refrigeration system in Sweden. A is hourly load for a day in January, B is monthly load in January, C is hourly load for a day in July and D is monthly load in July. The different colours are indicated on the figure and represents loads for LT evaporator, MT evaporator, space heating, tap water heating, air conditioning and gas cooler.

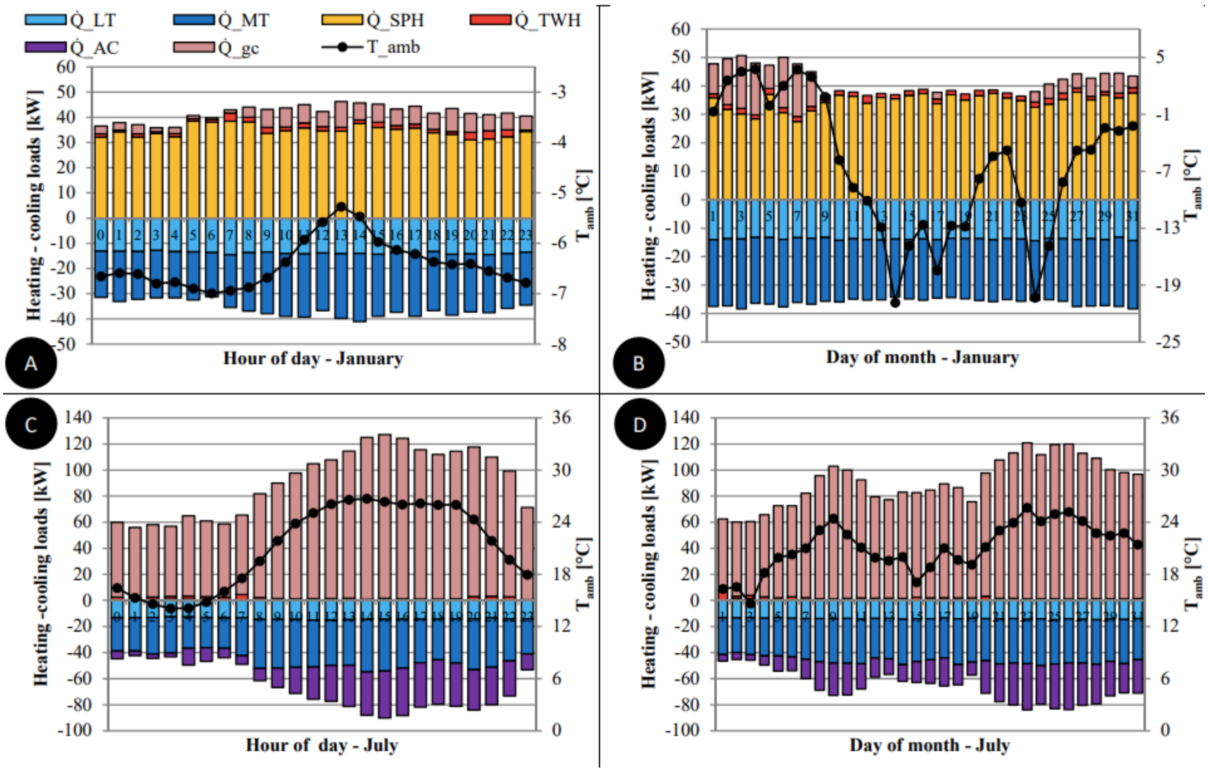


Figure 22: Cooling and heating loads for refrigeration system in Sweden

2.6 Capacity adjustments/part load operation

During the day the cooling load for the refrigerator will change. In order to adjust the compressors for the given cooling load there are according to Eikevik five main principles of capacity adjustments [31]:

- By-pass from high pressure side of the compressor to the low-pressure side
- Pressure regulation in the suction line (constant evaporator pressure), throttling in front of compressor
- On/off regulating of parallel compressors
- Compressor suction valve unloading (steps of number of cylinders)
- Speed control of the electric motor (normally minimum 50% of max rpm)

By-pass and pressure regulation are not recommended [31]. A supermarket refrigeration system normally has more than one compressor. The load can thereby be adjusted by switching on and off one or more compressors. The compressors can be combined in several different ways in order to achieve different loads. In addition to on/off regulation one of the compressors in the compressor rack normally has speed control. Compressors in Norway have 50Hz as standard. A frequency-controlled compressor can normally adjust the speed between 25Hz and 75Hz. This is dependent on the type of compressor. The compressor load can be reduced down to 25%.

BITZER has made a new CRII system for capacity control for reciprocating compressors for transcritical CO_2-applications. In combination with the compressor module CM-RC-01 this system allows for capacity control over wide range. The system combines capacity regulation by compressor unloading and speed control. The newly developed system allows to reduce the capacity down to 10%. [32] The system has been tested for two years in over 100 applications before being released to the market. The system allows for a larger lead compressor as the part load percentage is lower. One can either have more capacity of fewer compressors. [33]

2.7 Thermal energy storage

Thermal energy storage is temporary storage of high- or low temperature thermal energy for later use. Thermal energy storage is normally installed for two reasons [34]:

- Lower initial cost
- Lower operating costs

Thermal energy storage is used to adjust the time discrepancy of supply and demand of power. [35]. During peak demand the need for energy is high. By using a thermal storage one can store energy during off-peak hours and use the energy during peak hours. This leads to an even use of energy and a lower energy use during peak hours. When designing a storage, it is important to find a balance between maximizing the operation savings and minimizing the initial cost of the system that are needed to achieve the savings. [36]

2.7.1 Storing period

When talking about Thermal energy storages we divide between short term and long-term energy storage.

Seasonal storage: Long term storage is storing of seasonal heat or cold. This can be storing of summer heat for winter use or winter ice for space cooling in the summer. Seasonal storage requires immense storage capacity and are likely to only be economical in multi dwelling or industrial park design. [36]

Short term storage: An example of short-term storage is storing solar energy for overnight heating. Another example in the summer can be ice storage for space cooling. Freezing a PCM material at night and melting it in the day.

2.7.2 Storing of thermal energy

There are three types of ways to store thermal energy: sensible, latent and chemical. In this report the focus will be on storing of latent and sensible heat.

Sensible

In sensible heat storages the energy will be stored by increasing or decreasing the temperature of the storage medium. Equation 3 gives the sensible energy. The energy stored is dependent on the heat capacity c_p and the temperature change dT. [37]

$$Q_{sensible} = \int_{T1}^{T2} c_p * dT \tag{3}$$

Latent

In latent heat storage one will have a phase change in the storage medium. Using a PCM will increase the storage density because the phase change allows for storage of more energy with the same volume as sensible energy storage. Equation 4 gives the latent energy. The energy stored is both the sensible heat and the latent heat in addition to the heat of fusion at the phase change temperature T_{PC}. [37]

$$Q_{latent} = \int_{T1}^{T_{PC}} C_s * dT + \Delta H_{ls} + \int_{T_{PC}}^{T_2} C_l * dT \tag{4}$$

Figure 23 shows stored energy for thermal storage. The graph to the left shows a storage with only sensible heat while the graph to the right shows a storage with both sensible and latent heat. Using a PCM will increase the storage density because the phase change gives latent heat.

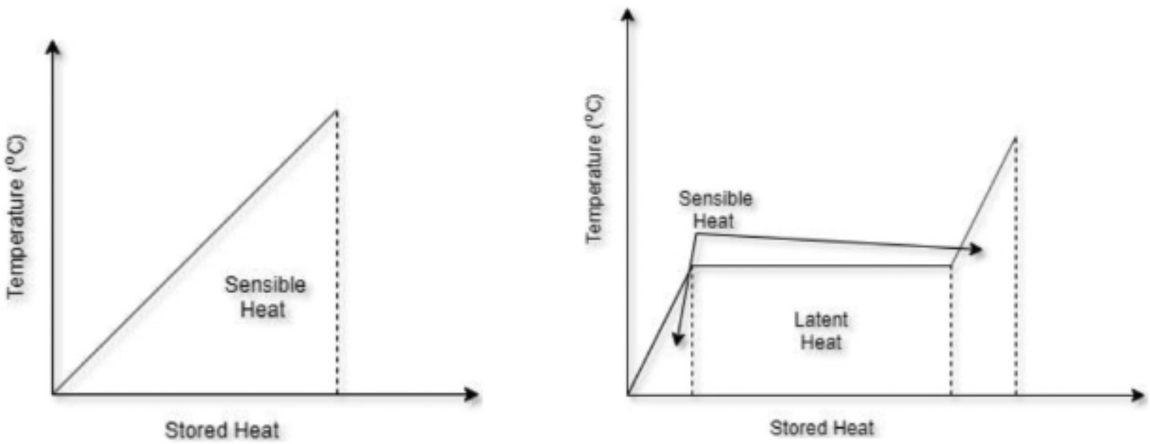


Figure 23: Sensible and latent heat storage [38]

2.7.3 PCM as storage materials

There are several different materials to use when storing heat and cold. Oró et.al has collected information from several research's and made a list of the main desired characteristics of PCM. The list is shown below [37]:

Thermophysical properties:

- Melting temperature in the desired operating temperature range
- High latent heat of fusion per unit volume
- High specific heat to provide additional significant sensible heat storage
- High thermal conductivity of both solid and liquid phases
- Small volume change on phase transformation
- Small vapor pressure at operating temperature
- Congruent melting of PCM for a constant storage capacity of the material with each freezing/melting cycle
- Reproducible phase change

Nucleation and crystal growth:

- High nucleation rate to avoid subcooling of the liquid phase during solidification, and to assure that melting and solidification process occurs at the same temperature
- High rate of crystal growth, so that the system can meet the demand for heat recovery from the storage system

Chemical properties:

- Complete reversible freeze/melt cycle
- No degradation after a large number of freeze/melt cycles
- No corrosiveness to the construction/encapsulation materials
- Non-toxic, non-flammable and non-explosive

Economics:

- Abundant
- Available
- Cost effective
- Easy recycling and treatment
- Good environmental performance based on Live Cycle Assessment

PCM materials are divided into different groups. Table 2 shows the three main groups of PCMs and their advantages/disadvantages. The three main groups are: Organics, Inorganics and Eutectics. These are further divided into subgroups. [39] Table 3 shows typical materials used for latent heat storage and their melting temperature and enthalpy according to Cabeza et.al [40].

Table 2: Classification of PCM materials [39]

Classification	*Advantages*	*Disadvantages*
Organics	1. Low or none undercooling 2. Chemical and thermal stability 3. High heat of fusion and no corrosives 4. Availability in large temperature range 5. Good compatibility with other materials	1. Low thermal conductivity 2. Lower phase change enthalpy 3. Relatively large volume change 4. Inflammability
Inorganics	1. High thermal conductivity 2. Greater phase change enthalpy 3. Low volume change and cost	1. Undercooling and corrosion 2. Phase separation 3. Lack of thermal stability
Eutectics	1. High volumetric thermal storage density 2. Sharp melting temperature	1. Low thermal conductivity 2. Corrosion in high temperature

Table 3: Common PCMs [40]

Material	*Melting temperature (C)*	*Melting enthalpy (MJ/m^3)*
Water-salt solution	-100-0	200-300
Water	0	330
Clathrates	-50-0	200-300
Paraffins	-20-100	150-250
Salt hydrates	-20-80	200-600
Sugar alcohols	20-450	200-450
Nitrates	120-300	200-700
Hydroxides	150-400	500-700
Chlorides	350-750	550-800
Carbonates	400-800	600-1000
Fluorides	700-900	>1 000

2.8 Cold thermal energy storage with water as PCM

For this report thermal storage using ice will be the main focus. This is a latent storage because of the phase change in the material when water freezes. Ice as thermal storage har been used for many hundred years. Before modern day technology was even invented ice was harvested during the winter and stored for use in the summer.

There are three main components needed for a thermal storage. The first is a PCM material that are suited for the desired temperature. To be able to store the PCM a storage tank is needed. In order to charge and discharge the storage one need a heat exchanger to transfer heat. The chapters below will explain the components and processes of CTES.

Water as PCM

Water is available everywhere and is a cheap material. It has good thermal properties with high heat capacity. Comparing storages of chilled water, ice and eutectic salt, ice needs considerably less volume for storage. [37] Table 4 shows the thermal properties of water and Figure 24 shows the different phases: frozen, liquid and gas.

Table 4: Thermal properties of water melting at 0°C

Values	
Melting temperature [°C]	0
Specific heat [kJ/kgK]	4.19
Density at [kg/m³]	Ca. 1000
Latent heat of fusion [kJ/kg ice]	334

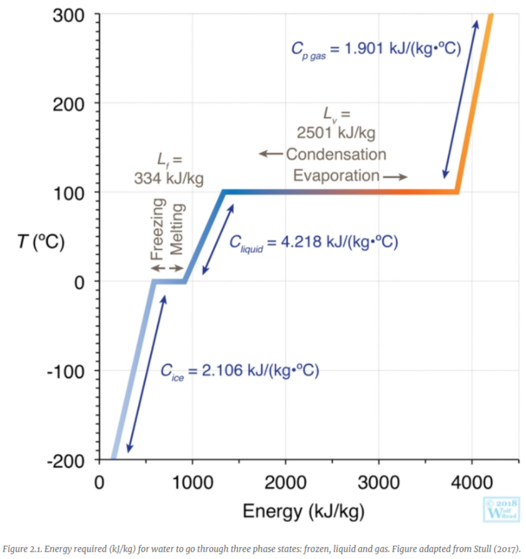


Figure 24: Energy required for water at different phases: frozen, liquid and gas [41]

2.8.1 Storage tank and design

In order to store the PCM one need a storage tank. The most important criteria that have limited the use of PCM in different systems are the type of container needed for the PCM and the number of cycles they can withstand without any degradation in their properties. [37] There are several different ways of designing storage tanks but they should all [42]:

- Meet the requirements of strength, flexibility, corrosion resistance and thermal stability
- Act as barrier to protect the PCM from harmful interaction with the environment
- Provide sufficient surface for heat transfer
- Provide structural stability and easy handling

Important factors when designing a storage unit with PCM [42]:

- Temperature limits within which the unit is to operate
- The melting-freezing temperature of the PCM
- The latent heat of the PCM
- The thermal load
- Configuration of the storage

When it comes to material used for PCM tanks, one should avoid the use of cooper and carbon steel due to high rate of corrosion and the presence of precipitates and PH change. It is also not recommended to use Aluminium due to the formation of bubble shapes on the surface. This could lead to holes in the container. Stainless steel alloys as tank material are highly recommended for longer periods. [43]

Charging and discharging

During charging cold refrigerant is circulated through the heat exchangers and the PCM freezes. During discharging warm refrigerant circulates through the heat exchanger and the PCM melts. The storage type is divided into static and dynamic systems. The static systems produce ice bonds on the cooling surface and forms a layer of ice on the cold surface. In dynamic systems the ice is continuously removed from the cold surface. [35] Figure 25 shows the charging process of a PCM storage. Working fluid is circulated in the pipe and transfers cold to the PCM. The PCM starts to crystalize close to the pipe.

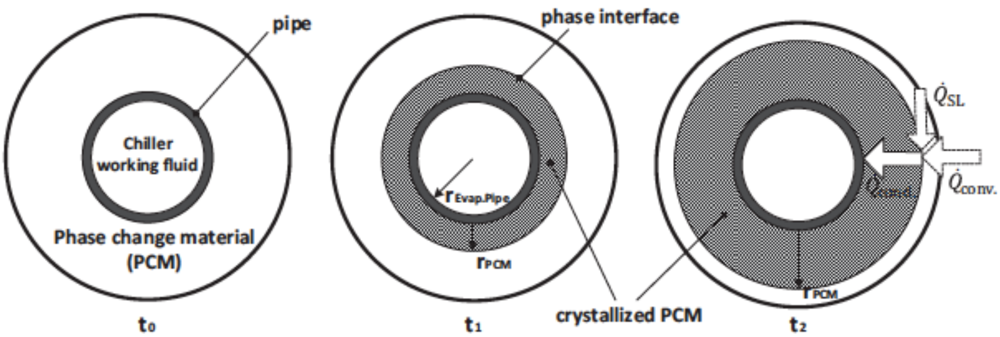


Figure 25: Charging of PCM storage [44]

One of the challenges for large-scale systems using CTES with low temperatures is to extract the energy fast enough during peak hours [45]. Another challenge is that the critical physical dimensions for phase change thermal storage units varies as the storage material freezes and melts [46] Ice has lower thermal conductivity than liquid water. The heat transfer will decrease as the ice layer on the heat exchanger surface grows thicker. Transferring energy by convection is a more effective than by conduction. During discharging a layer of water forms around the heat exchanger surface allowing heat transfer by convection. Therefor the discharging process takes less time than the charging process [42]

There are many different storage tank designs. Some of the most common combinations of heat exchangers and tanks will be explained in the following chapters.

2.8.2 Tube in tank

Shell and tube

In a shell and tube heat exchanger the heat transfer fluid flows inside a tube and the PCM is contained inside a shell enclosing the tube. Research performed on shell and tube storage shows the during charging the most important parameters are: inlet temperature, thermal conductivity and diameter of the shell. During discharging the dominant parameter influencing the energy distribution is the inlet temperature. [47]

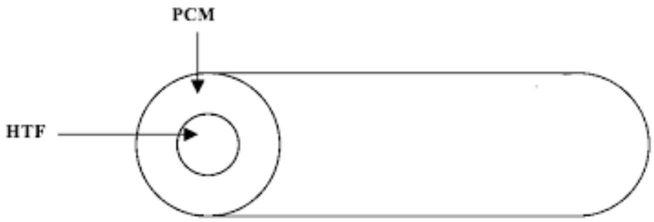


Figure 26: Shell and tube heat exchanger.

Figure 27 shows the charging cycle of a shell and tube thermal storage and Figure 28 shows the discharging cycle. The simulations are performed by Ezan et.al [47]. In Figure 27 one can see the ice forming on the outer walls of the tube. The charging process takes longer time than the discharging cycle. As explained in chapter 2.8.1 the formation of ice on the tube reduces the heat transfer between the tube to the PCM.

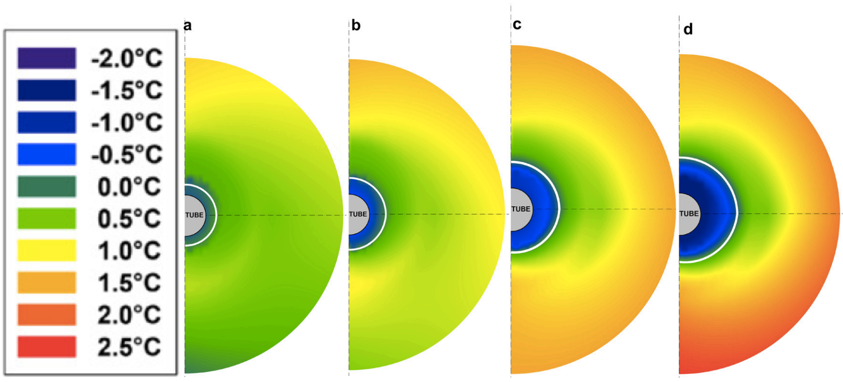


Figure 27:Charging(Tin=-10℃, V=2L/min) a(t=60min), b(t=120min), c(t=240min) and d(t=480min) [47]

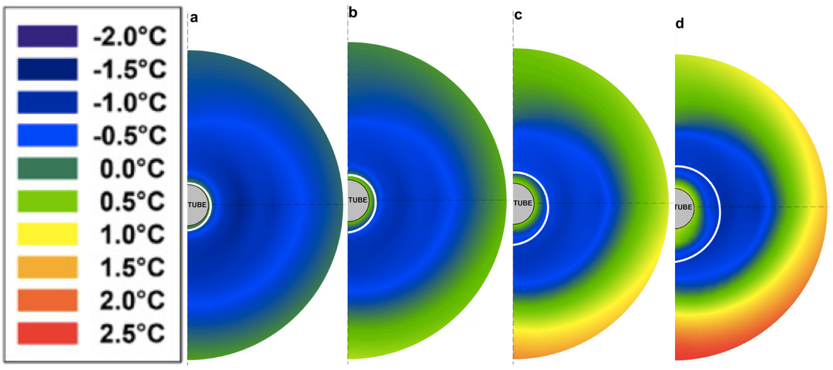


Figure 28: Discharging (Tin=+5℃, V=2L/min) a(t=60min), b(t=120min), c(t=240min) and d(t=480min) [47]

Tubes in tank

One of the main challenges of PCM systems is according to Bush et.al the low heat transfer between the PCM and HTF. In order to secure sufficient heat transfer at large heat transfer surface is required. For a tube in tank system this is accomplished by having several tubes packed closely with a long length. [48]

Tay et.al performed experiments on tube in tank systems. They used several different experimental setups with different PCMs and number of tubes. Tubes of polyvinyl chloride was added to a tank with circulating HTF. Figure 29 shows the system for one tube in tank and four tubes in tank. Figure 30 and Figure 30 shows the results from the experiment performed with water as PCM and four tubes in tank system. [49]

For the discharging process a temperature difference of 18 degrees was used and for the charging process a temperature difference of 30 degrees was used. The first period of the diagram represents the sensible process. [49] For this experiment the discharging process is longer than the charging process. Compared to simulations performed by Ezan et.al [47] the temperature of the HTF during charging is a lot lower.

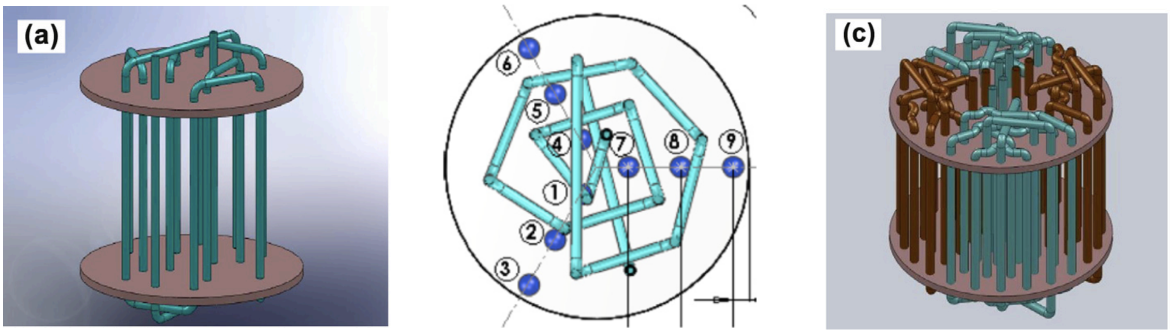


Figure 29: Tube in tank systems by Tay et.al [49](a)One tube tank, (b) measuring points and (c) Four tube tank

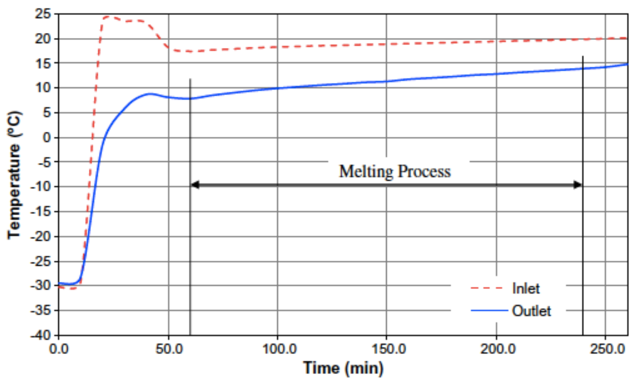


Figure 31: Discharging of four tubes in tank (m=0.021kg/s, ε=0.42)

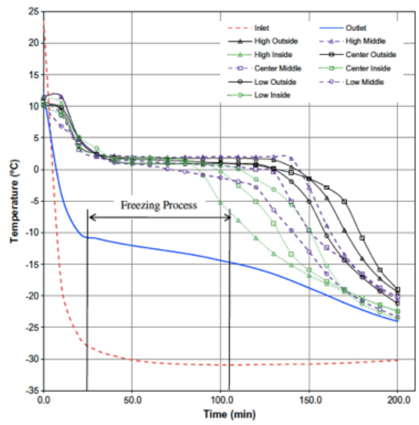


Figure 30: Charging of four tubes in tank (m=0.019 kg/s, ε=0.56)

2.8.3 Coil in tank

The coil in thank system is one of the most common thermal storage methods. There are several different parameters influencing the storage. One has to determine geometric parameters of the coil like: length, diameter and pitch. The position of the tank is also important and will influence the charging and discharging process. The tank can be placed either vertical or horizontal.

Horizontal tank

Ajarostaghi et.al conducted experiments of the discharging process of a coil in tank system. The storage tank was in this experiment placed horizontally. Figure 32 shows the thermal storage design and Figure 33 shows results from the experiments. The tank has a cylindrical shape with a HTF coil inside.

Effect of natural convection

Figure 33 shows the results from the simulation by Ajarostaghi et.al. The simulation shows that natural convection will have a big influence on the discharging process. The temperature in the upper part of the tank was higher than in the lower part. The experiment showed that the beginning of the discharging process is dominated by convection. The PCM starts to melt close to the coil creating an area of liquid around the coils. The liquid area will start to grow and after a few minutes the discharging process is dominated by convection. [50]

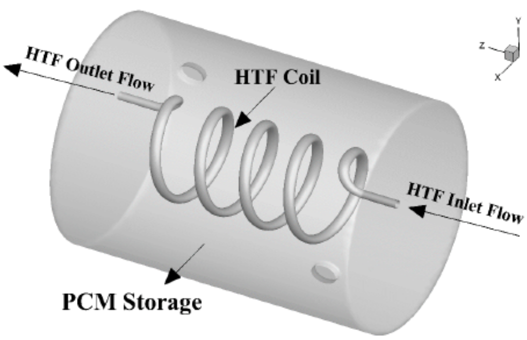


Figure 33: coil in tank design by Ajarostaghi et.al

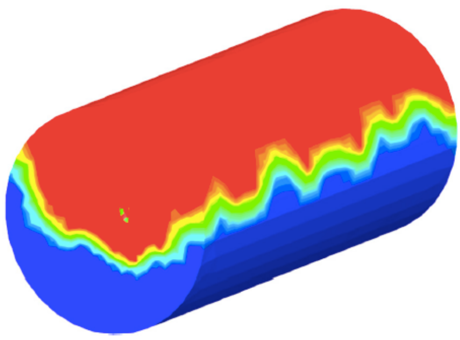


Figure 32:coil in horizontal tank [50]

Effect of inlet temperature and mass flow

Ajarostaghi et.al conducted experiments on the effect of inlet temperature Figure 34 shows the results from the experiment. The results show that increasing the temperature from 5 to 7 °C reduces the melting time by 25%. The research also showed that increasing the mass flow had little effect on the discharging time. Increasing the mass flow from 2 to 8 L/min only gave a decrease in melting time of 2% [50]

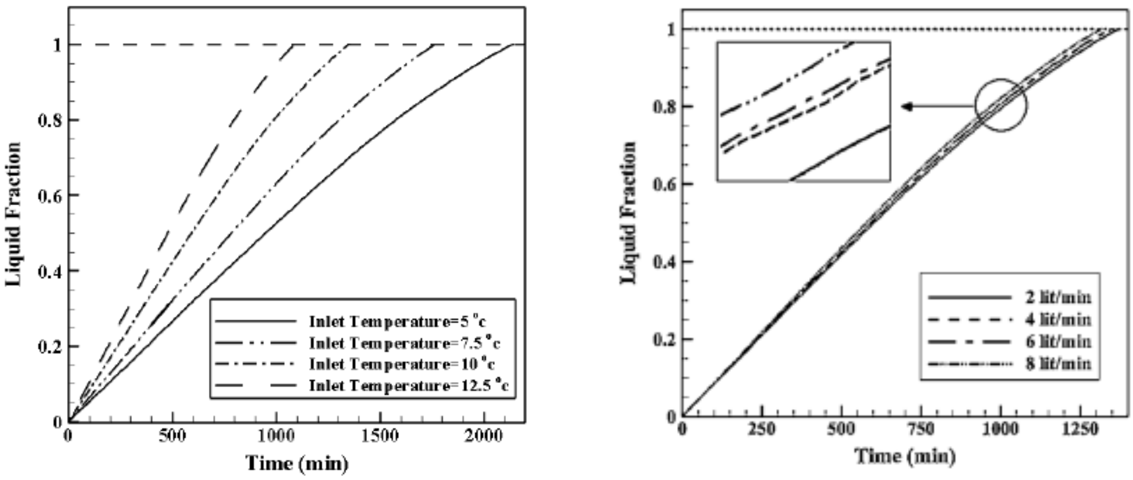


Figure 34: Melting time for different HTF inlet temperatures and mass flows [50]

2.8.4 Pillow plate heat exchangers

There are limited published work on heat transfer in pillow plates. When using this type of heat exchanger there is no need for a secondary fluid. The refrigerant can both evaporate and condense directly in the storage exchanging heat with the PCM material. Pillow plate heat exchangers can withstand the relatively high operating pressure of CO_2 systems. [45] There are several ways of enhancing heat transfer in PCM storages, many of which is quite expensive. The wavy structures of the pillow plates are one way of enhancing the heat transfer. Compared to conventional heat exchangers, pillow plates have low manufacturing costs and compact design. [51]

Figure 35 shows the geometry of the PPs (pillow plates). The geometry of PPs looks like a pillow. The way it is created is that two plates are welded together and then put under pressure. As shown in Figure 36, several plates are stacked together to form the heat exchanger. On one side there is an inlet supplying refrigerant and on the other side there is an outlet collecting the refrigerant after evaporation/condensation. [51].

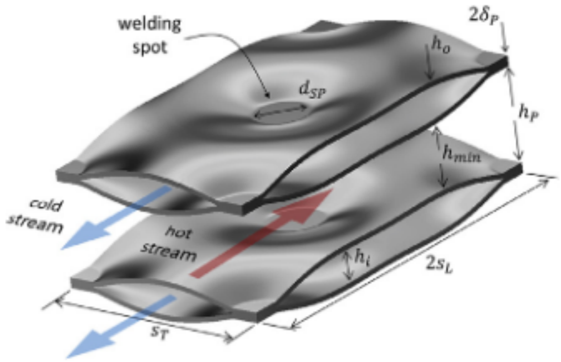


Figure 36 Geometric of a pillow-plate heat exchanger [74]

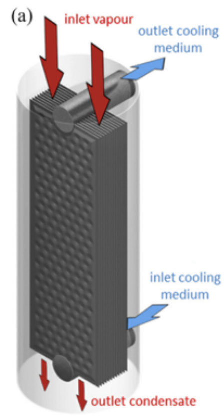


Figure 35: stacking of pillow plates [51]

Advantages of pillow plates [52]**:**

- Cost-effective (due to low welding costs)
- Reduced material costs (Thinner materials can be used)
- High overall heat transfer coefficients (Pillow Plates welding pattern guarantees high turbulence)
- Reduced deposition (due to higher turbulence)
- Less pump capacity is required (Due to the low volume in the plate, little cooling / heating fluid needs to be circulated)
- By working with laser welding machines that are CNC-programmed, all desired shapes, recesses, fastening points and connections are possible

2.8.5 Capsules of PCM in storage tank

Using a storage tank with encapsulated PCM is a type of thermal storage. There are three main types of storage tank designs for this solution based on the geometry of the capsule: Rectangular, cylindrical, and spherical. Figure 37 shows storage with rectangular and cylindrical capsules. [39] Figure 38 shows storage with spherical capsules.

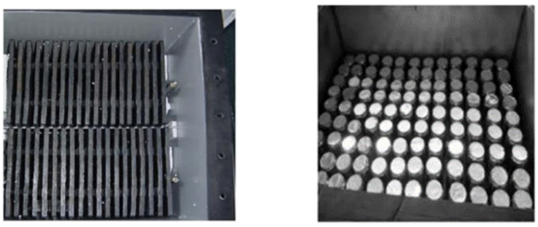


Figure 37: From the left: Rectangular and cylindrical capsules [39]

Spherical capsules

Spherical storage tanks are one of the tanks that are most studied. [37] Castell et al. has studied different shapes for storage tank design. Based on the research he found that spherical tanks due to the packaging factor can have a reduction of effective storage density of 50%. [53] Figure 38 shows a storage with spherical capsules used for an experimental investigation by Lee et.al [54]. In Figure 38 (a) illustrates the melting model with natural convection, (b) illustrates the thermal conductivity model and (c) the total thermal storage.

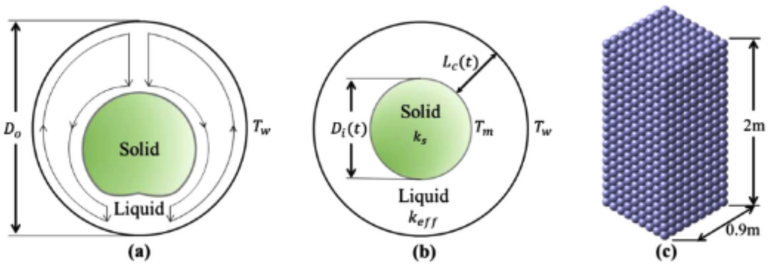


Figure 38: Storage system with PCM capsules [54]

Figure 39 shows the result of liquid fraction during discharging of the storage by Lee et.al. The effect of natural convection is an important factor in discharging of a thermal storage. This process will highly affect the time required for simulation of thermal storage models. Lee et.al simplified the model by modelling the storage with higher thermal conductivity instead of convection. [54]

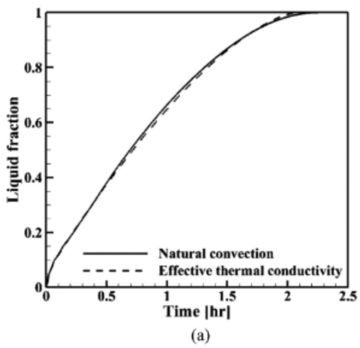


Figure 39: Discharging storage with PCM capsules [54]

2.8.6 Enhancing thermal properties of CTES

There are two approaches to enhancing the thermal properties of a CTES system [45]:

- Act on the PCM
- Act on the storage

Thermal conductivity promoters: It is possible to enhance the heat transfer in the storage by using fins, metal matrices, high conductivity particles, metal fibres etc. An example of material used is copper, aluminium and stainless steel. [37]

Porosity: Lower porosity can lead to higher charging time, internal heat transfer coefficient and heat capacity. The time required for freezing thereby increase. For systems with high porosity the charging time and the capacity of the storage decreases. [37]

Use of bubbles to enhance solidification: Studies has also been conducted on the use of bubbles to enhance the heat transfer rate and accelerate the growth of ice layer. Mohammed found that the solidification velocity increased by 20-45% by using bubbles [55]. The bubbles also created turbulence and thereby increased the stored energy. [37]

Close contact melting: At the NTNU lab there is a CTES using pillow plates. The pillow plates are stacked horizontally. By stacking the plates horizontally, it is possible to take advantage of the buoyancy effect. During discharging the CO_2 is condensing in the plates. The ice will start to melt close to the plates as this is where the heat is rejected. Since there is difference in density between the solid and liquid PCM the solid PCM will rise. The ice will then make contact with the upper pillow plate and a situation that is called close-contact melting will occur. This will provide high heat transfer between solid parts of the PCM and the condensing CO_2 in the pillow plates. [45]

Emulsions and microencapsulation: Ice forming on the surface of the heat exchanger will reduce heat transfer during charging. There are two techniques used to prevent PCM from solidifying on the surface: emulsification and microencapsulation. [56]

2.9 Combination of chiller and storage

When choosing the size of the chiller we first need to decide how big storage we want to have. We need to consider if we want a full storage that can cover the entire load or a partial storage that cover parts of the load. Storage and chillers can be combined in several different ways in order to achieve different goals. ASHRAE journal has published an article written by Brian Silvetti explaining different system solutions for ice-based thermal storage. He lists four different solutions for combination of chiller and ice thermal storage The different solutions are shown in Figure 40. [46]

The full storage solution covers the entire cooling load, while the partial storage covers only the peak load. The partial storage thereby reduces the peak load required from the grid in the hours where the energy demand is at its highest. The next option is a system with two chillers. For this solution both chillers operate at night in order to charge the storage. At daytime when the cooling need is at its highest, only one chiller operates. For the last solution the storage covers only a shorter period of on-peak hours, while the chiller covers the rest. The on peak period is normally set from 12 to 18. [46]

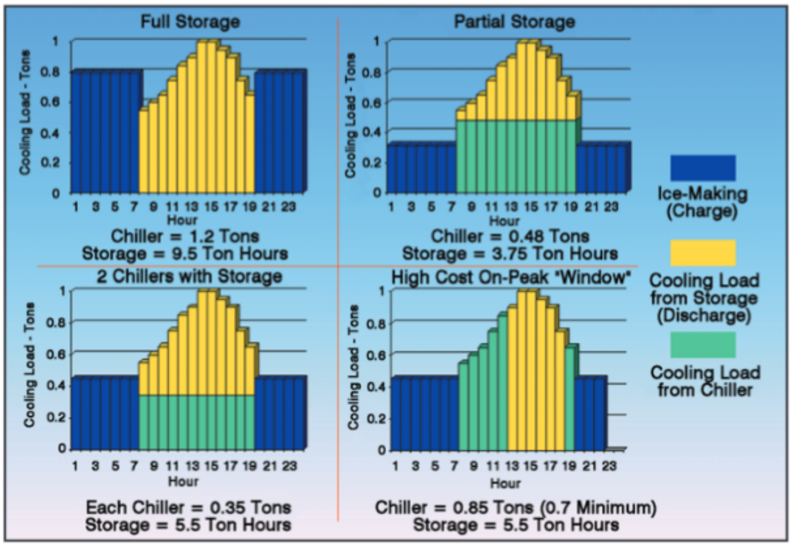


Figure 40: Combination of chiller and storage ASHRAE [46]

Parallel or series

The system can be arranged in both series and parallel. If the system is in parallel, there will be need to change the flow path when changing between charge and discharge mode. If the system is in series this will not be necessary [46] The most common configuration is having the storage in parallel. This allows the system to be operated in many different modes. One can only charge the storage, only cool using the storage or only cool using the chiller. It also allows for a combination of using chiller and charging/discharging the storage. [34]

2.9.1 Peak shaving and load levelling

Combining chiller and storage allows for peak shaving of the cooling load. By inserting a thermal storage into the refrigeration system one can correct the mismatch that occur between supply and demand of energy. [34] The integration of thermal storage in the refrigeration system allows to reduce the load during peak hours and transfer them to off peak hours. Figure 41 illustrates the peak shaving method. The load is transferred to off-peak hours making the load on the system lower and more constant.

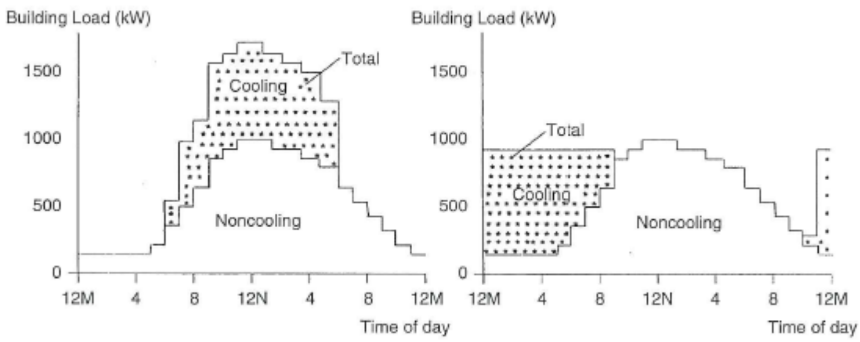


Figure 41:Peak shaving [36]

Figure 42 shows different combinations of CTES and refrigeration system. One can (a) either have a full storage where the refrigeration system operates at a higher load during off peak hours and are turned completely off during peak hours. One can (b) have partial storage solution where the refrigeration system runs at a constant load during the entire day. This is called load levelling. For the last alternative (c) the refrigeration system operates at a higher load during off peak hours and a lower load during peak hours. [57]

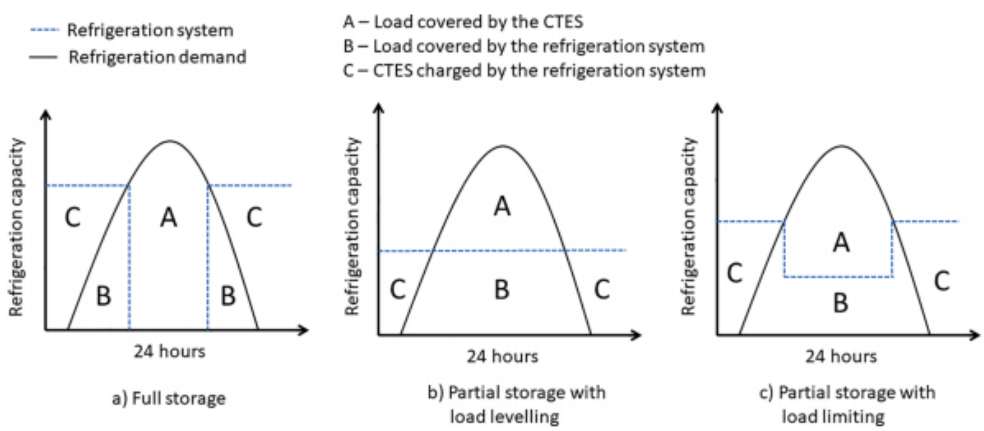


Figure 42: Full and partial cold thermal energy storage solutions in combination with refrigeration system [57]

Peak shaving can also be based on operating costs. This is most relevant for countries where the energy prices are high. If the country has big variations in the energy prices during the day, then it will be profitable to use energy when the price is low and not use energy when the price is high. Figure 43 shows graph for price changes during a day. As one can see the energy prices are highest between 15 and 20. A solution with charging at night/early morning and discharging during the day can be an optimal solution.

Figure 43: Price depending on hour [58]

2.10 Advantages of integrating cold thermal energy storage

According to Selvnes et.al there are mainly three reasons to look at implementation of CTES in industrial refrigeration systems: peak load shifting, better utilization of refrigeration equipment and lower emissions of greenhouse gases. [59] As mentioned in chapter 2.7, thermal storage is normally installed to get lower initial costs or lower operating costs. The optimal thermal storage will have balance between maximizing the savings for operating costs and minimizing the installation costs needed to reduce the operation costs. [34]

Advantages of cold thermal storage [60] [37]**:**

- Reduce initial cost
- Reduced pipe and pump sizes for chiller water distribution
- Reduced operating and maintenance costs
- Increased flexibility of operation
- Extended capacity of existing system

Reduced size and cost of system: One of the reasons why thermal storage has become an interesting solution for cooling systems is the low investment cost and low operating costs. [50] In every production process it will be economically inefficient to install equipment that accommodates for the maximum demand (peak demand). The productivity will decrease during off-peak hours when the equipment cannot operate at full capacity. [61] When using CTES one can use smaller and less expensive compressors. One can also use smaller pumps, pipes and air handlers. [34]

Reduced operating and maintenance cost: Shifting the chiller load from on-peak periods to off-peak periods will reduce electricity costs. Companies in the industrial sector pays for the highest peak load. By reducing the chiller load during on-peak hours will thereby reduce the electricity bill. Integrating a thermal storage makes it possible to run the compressor more constant and thereby reduce frequent starts and stops. This is better for the compressor and reduces the need for maintenance. It can lead to increased lifetime of the compressor.

Improved system COP: Shifting the load from the day to night gives better working conditions for the system (colder at night). This will give a higher system COP. A storage can improve the system COP by reducing throttling losses during unfavourable conditions.

Increased flexibility and longer backup: Supermarkets have high values of groceries stored in the shops. As an example, the vegetable refrigerators have an income of approximately 100 000 NOK per week. Installing a thermal storage will give backup in case of system failure. A thermal storage can give several hours of refrigeration without power supply. The integration of a storage allows to move the energy consumption from the grid to off-peak periods and increases the display cabinet flexibility for energy management applications. [62]

Improved food quality and reduced off time of compressors: Temperature fluctuations will affect the quality of the food. According to Oro drop in product and air temperature are caused by: Electrical power failure, door openings and heat generated during defrosting [63]. Integrating a storage can reduce temperature of the goods during defrosting and give the food a more stable temperature. Stable temperature will give longer shelf life.

2.11 Implementation of CTES in CO_2 refrigeration systems

This chapter will go through some existing systems and facilities that have integration of storage. It will start with different placement of the storage, both in the refrigeration system and inside the MT cabinets. Then there will be shown some systems built by Viessmann, NTNU and at HVL (Western Norway University of applied science).

2.11.1 Integration of CTES in refrigeration system

Fidorra et.al has investigated different solutions for integration of CTES. Three of the solutions are shown in Figure 44. Layout #1 charges the storage after the receiver and discharges the storage after the gas cooler. The storage is used to subcool the refrigerant after the gas cooler. Layout #2 shows a solution where the storage is integrated in the MT level load. This can as an example be a storage integrated in the back of the MT cabinet. In Layout #3 the storage is discharged after the receiver. Thereby the refrigerant from the receiver is cooled before entering the evaporators. This reduces the mass flow needed for cooling. [64]

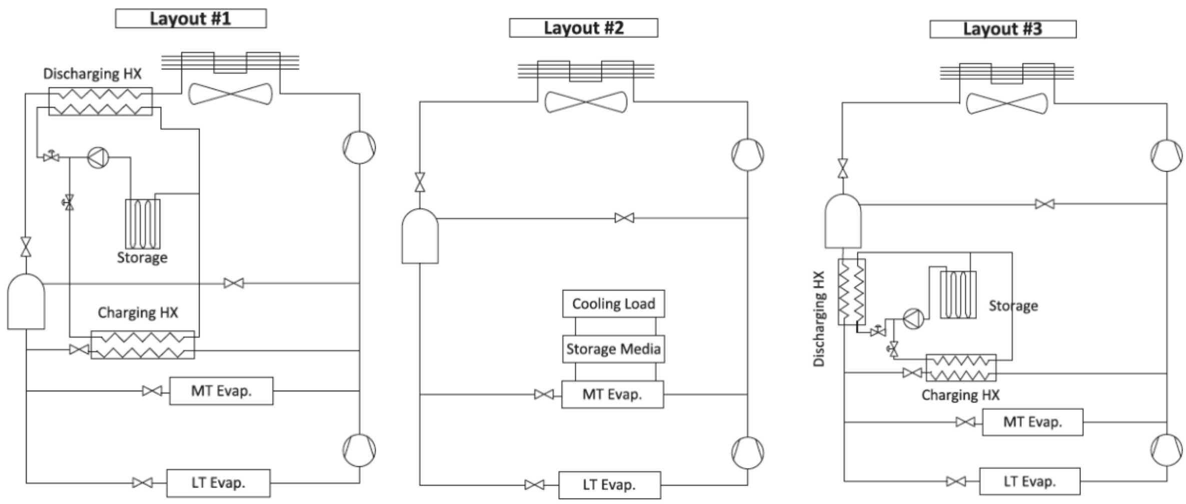


Figure 44: Different solutions for integration of storage by Fidorra et.al [64]

Research by Fidorra et.al showed that Layout #1 had high potential for reduction of demand. [64] For warmer countries it will be beneficial to implement the discharging of the storage after the gas cooler. Discharging the storage after the gas cooler, will reduce the vapor quality of the refrigerant and give the system a better efficiency. Research performed by Llopis et.al shows that subcooling after the gas cooler can improve the COP between 6.9% to 30.3% with an evaporation temperature of -10 °C [65]. In colder countries implementation of the storage at MT level can be a good solution. In colder countries one does not have the same need for subcooling after the gas cooler.

Heerup and Green has investigated different solutions for integration of CTES. Figure 45 shows one of the solutions. The storage is here placed upstream of the receiver. The storage will reduce the load on the MT compressors by condensing the flash gas after the high-pressure valve. Calculations shows that the energy savings is 4% for cold climate and 14.4% for warm climate (Using climate from Denmark and San Francisco). During the warmest day of the year the peak power is reduced by 50%. [57]

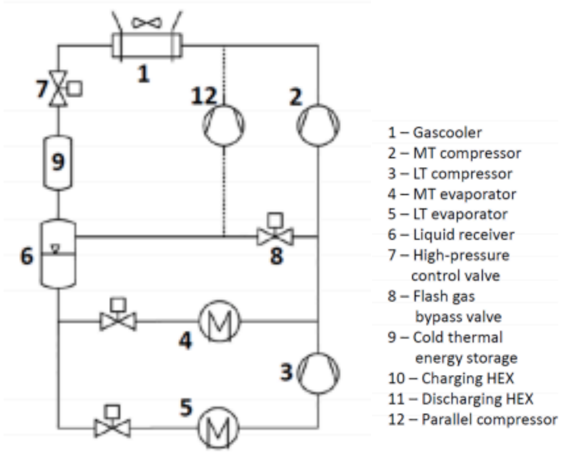


Figure 45: Different solutions for integration of storage by Heerup and Green [57]

Wang et.al has investigated several different solutions for implementation of storage. Figure 46 show three of the different solutions investigated. Solution A and B gave an increase in COP by 6% and 8%. Solution B gave the best improvement in COP while solution A and C gave better stabilization for the system. [57]

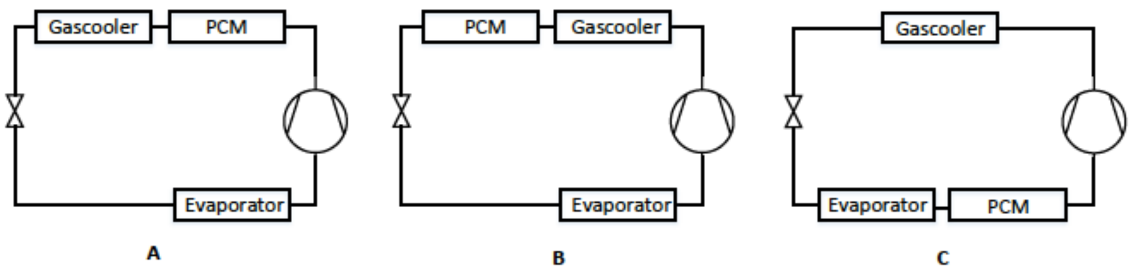


Figure 46: Different solutions for integration of storage by Wang el.al [57]

2.11.2 Integration of CTES in refrigeration cabinet

There are several possible solutions for placement of the storage. Storage used for big facilities often have one big centralized storage. For supermarkets the space is more limited. A possible solution is to locate the storage inside the refrigeration cabinets. In periods where the compressor is off, the temperature inside the cabinet will rise. If there is used PCM inside the cabinet, the PCM will absorb most of the heat by changing from solid to liquid state. The PCM will allow the food products to maintain their temperature during periods where the compressor is off. There has been conducted several researches on the solution with storage integrated in the cabinet. Two different solutions are shown in this chapter.

One storage placed in the back of the cabinet

Figure 47 shows a solution where the storage is placed in the back of the cabinet. This is a system created by Abdallah et.al. The system consists of a finned tube heat exchanger with 7 kg of water. Table 5 shows charging and discharging time for the storage. One can see that the charging time is more than two times longer than the discharging time.

Table 6 shows the temperature of products during compressor stop. One can see that the temperature rise in the product is lower for the system with PCM. The temperature rise of the products is approximately 1 degree Celsius after 2 hours. [62]

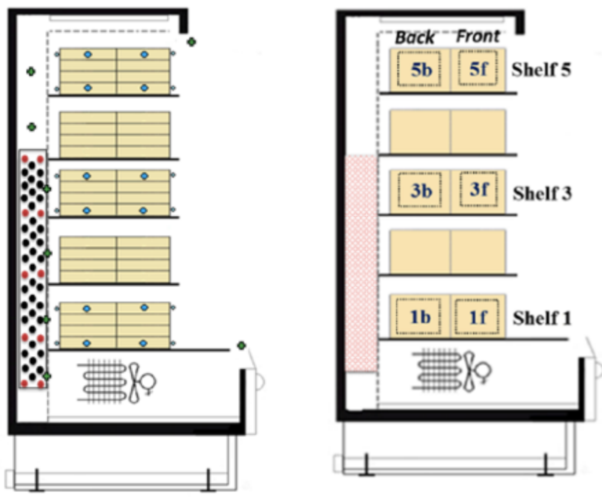


Figure 47: CTES located in the back of a refrigeration cabinet [62]

Table 5: Charging/discharging time [62]

Charging ratio [%]	Charging time [min]	Discharging ratio [%]	Discharging time [min]
0	0	100	0
25	24	75	25
50	125	50	60
75	183	25	68
100	225	0	96

Table 6: Product temperature rise during compressor stop with ambient temperature 16℃ [62]

Position	*After 1h without PCM*	*After 1h with PCM*	*After 2h without PCM*	*After 2h with PCM*
1f	0.7	0.2	1.8	1.1
1b	1.0	0.0	2.5	0.0
3f	1.0	0.5	2.5	1.2
3b	0.9	0.2	2.1	0.8
5f	0.6	0.3	1.5	0.8
5b	0.8	0.4	1.9	0.7

Several small PCM storages placed around the cabinet

Figure 48 shows a system solution by Gin et.al. [66] Several PCM panels are placed in the back of the cabinet and in the sides of the upper part of the cabinet. Research performed by Prim [43] shows that the temperature of food during electrical failure is always lower when PCM is used. Using PCM in the cabinet resulted in better food quality during power cuts.

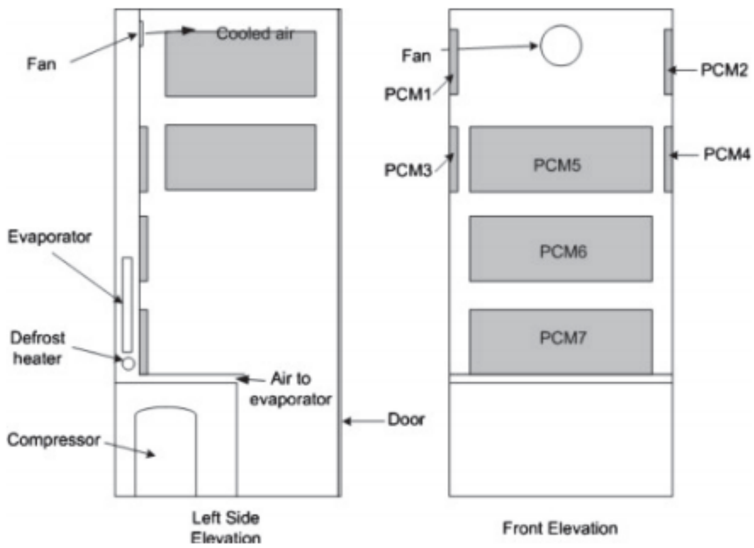


Figure 48:Several small CTES located different places in the refrigeration cabinet [66]

2.11.3 Local storage with centralized CO_2 units

The space inside supermarkets is limited. Figure 49 shows integration of storage above the refrigeration cabinet. The solution is presented in a report by Manescu et.al [67]. This gives optimal use of space in the shop. One can exploit the available space above the cabinets. Compared to placement of storage inside the cabinet one does not loose space in the cabinet or floor space.

The numbers in Figure 49 represents the different components. (1) is the storage and (2) is the evaporator. The letters show: (a) charging mode and normal operation of the cabinet, (b) normal operation of cabinet, (c) charging mode of PCM storage and (d) discharging of storage. [22] During discharging the system will be self-circulating and disconnected from the rest of the refrigeration system.

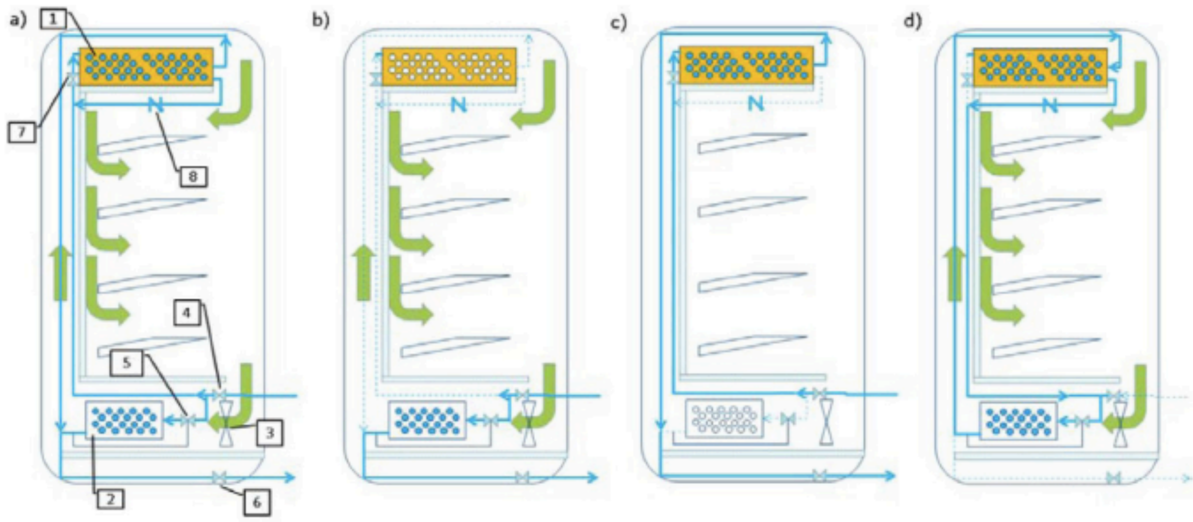


Figure 49:local storage with centralized CO_2 unit. (a) charging and normal operation of cabinet, (b) normal operation, (c) charging of storage and (d) discharging of storage [67]

Placing the thermal storage close to the products is important to secure cooling during power cuts and stabilized temperature inside the cabinets. It is possible to obtain more stable product temperatures during defrosting of the evaporators with the integration of a storage. During defrosting the storage can absorb the heat. [22]

2.11.4 Vissmann system

Vissmann has developed a supermarket refrigeration system with integrated CTES. The system is called ESyCool green integral plus. The refrigeration system is a cascade system with brine as heat transfer medium. For the MT cabinets R290 is used and for the LT cabinets CO_2 is used. Figure 50 shows the ESyCool green integral. This is the standard system without CTES. [68]

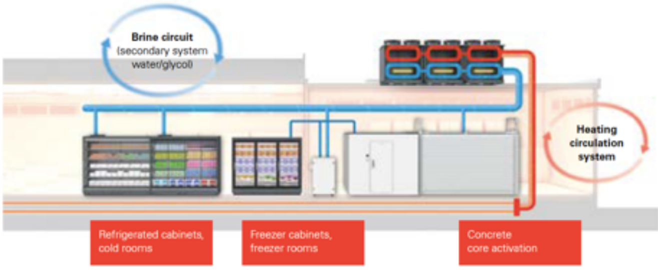


Figure 50: Vissmann ESyCool green integral [68]

Figure 51 shows the ESyCool green integral plus with integrated thermal storage. The system allows to reduce the energy use by combining heat recovery, ambient air, ice energy storage and electric heating. When the building has a heating demand, excess heat from the cabinets can be used for space heating through floor heating. In Figure 51 one can see the floor heating loop in the floor. [68]

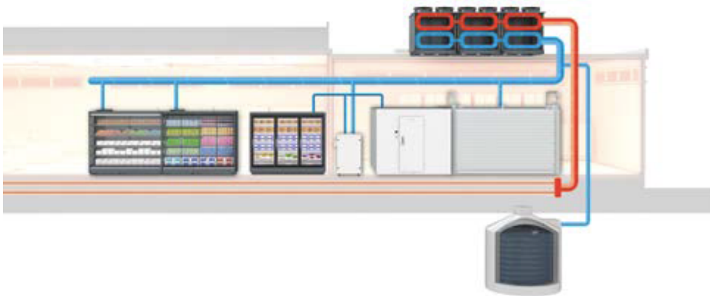


Figure 51:Vissmann ESyCool green integral plus [68]

Due to the compact hermetic heat pump design the system has 90-95% less refrigerant charge. The system also has less life cycle costs with up to 25% less energy consumption. The requirements for service and maintenance are simplified. [69]

2.11.5 CTES at NTNU

NTNU has built its own test facility for PCM storage at the laboratory at Varmeteknisk. The storage system built at the lab is integrated in an industrial NH_3/CO_2 cascade refrigeration system. Figure 52 shows the P&ID for the heat pump system with storage. The storage is shown in Figure 53. There are installed two windows in the storage where one can observe the process of charging and discharging of the PCM. The storage consists of pillow plates emerged in water. The pillow plates are stacked horizontal. There is a pipe welded to the plates at each side for the refrigerant to flow through the plates. The storage can fit 20 pillow plates and about 1 275 kg water. The number of plates and the distance between the plates can be adjusted for different experiments.

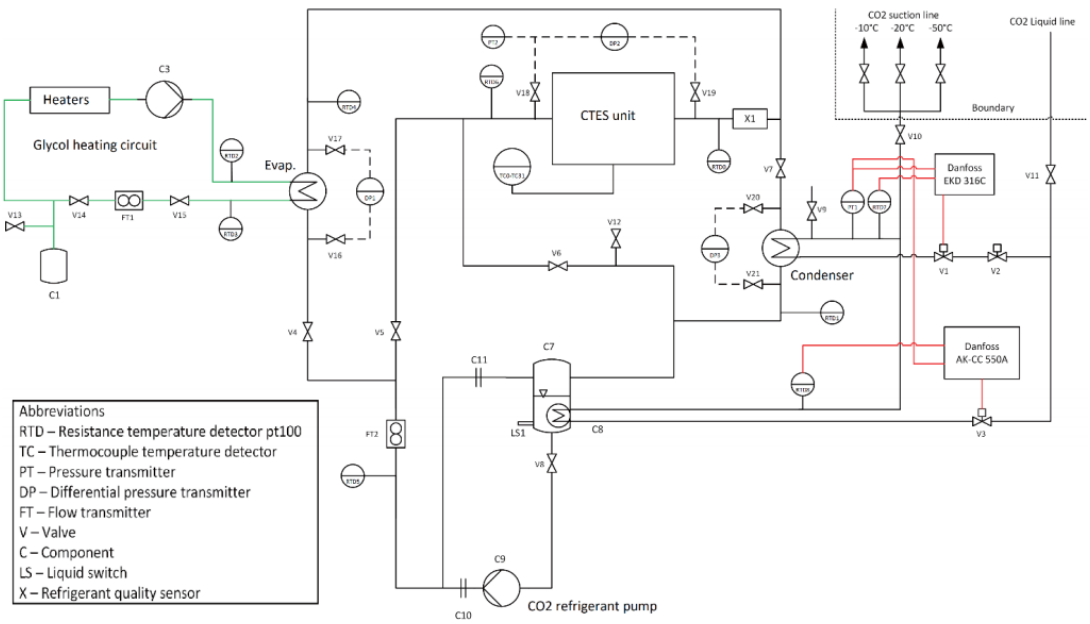


Figure 52: P&ID NTNU lab [70]

Figure 53: CTES unit NTNU LAB [70]

Experiments on the storage has been performed by Håkon Selvnes. Figure 54 and Figure 55 shows some of the results from the experiments. Figure 54 show discharge duty for the storage at different inlet temperatures and different plate distances. The discharge duty will be high in the beginning and then decrease. As one can see the distance between the plates will have a significant influence on the discharge duty. Figure 55 shows the average discharge duty for different inlet temperatures and mass flows. Higher inlet temperature giver higher average discharge duty. [71] [70]

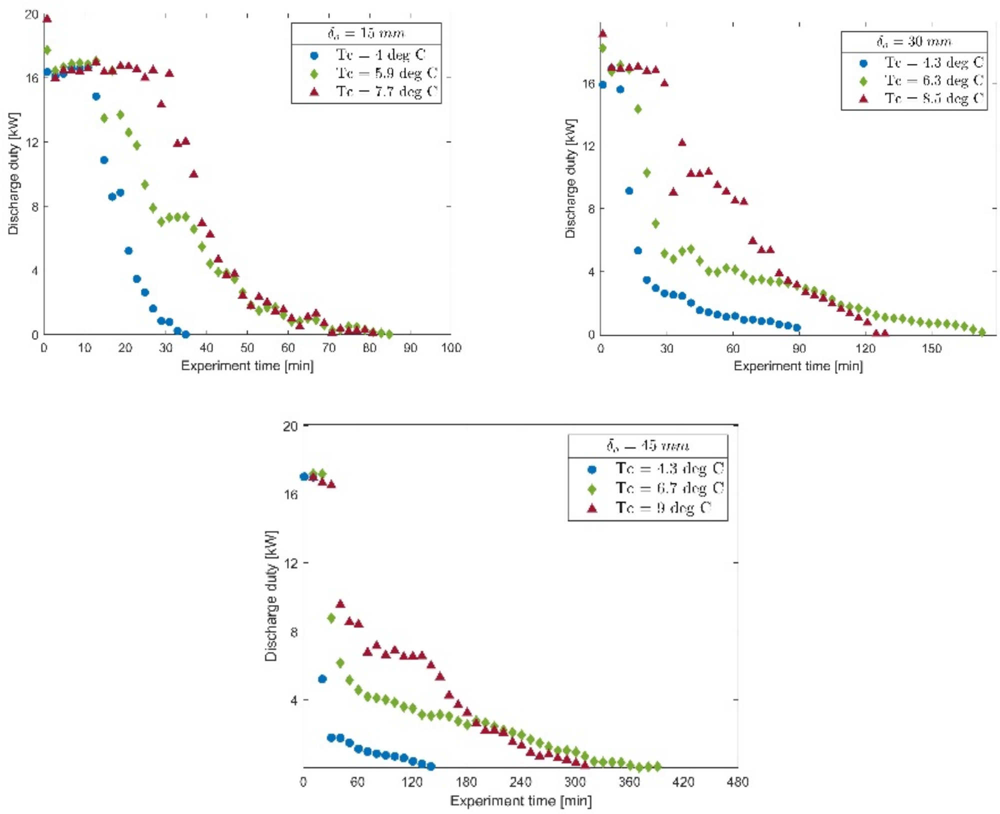


Figure 54:Discharge duty of PCM storage by Håkon Selvnes [71]

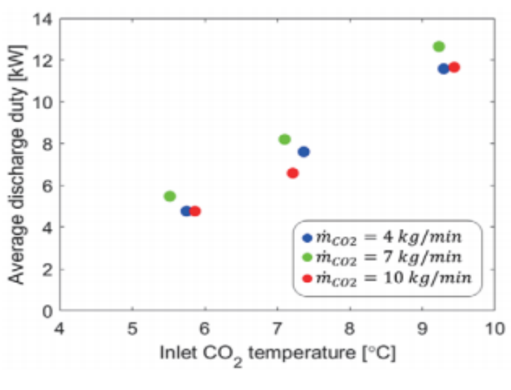


Figure 55: Average discharge duty. Experiments by Håkon Selvnes [70]

2.11.6 Examples of implementation of CTES for AC load

Figure 57 shows a typical flat-ice container. Figure 56 shows the flat-ice PCM system at HVL (Western Norway University of applied science). The system is installed to reduce peak demand for cooling in the summer. The PCM freezes in the night when the outside temperature is lower. At daytime when the outside temperature is high, the PCM melts and gives cooling to the AC. The system uses a eutectic salt that freezes at 10 degrees Celsius. There are installed 4 tanks of 64 m^3 that holds 11 000 kWh together. The tanks are shown in Figure 58. Figure 59 shows how the CTES reduces the energy use for cooling during the day.

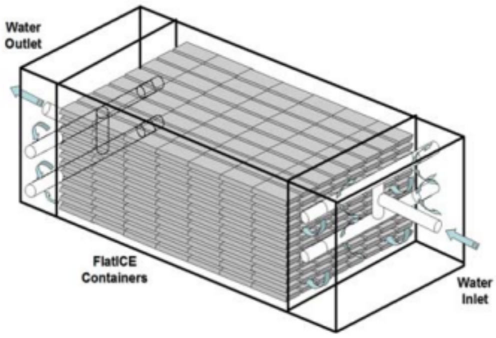


Figure 57: Flat-ICE container [72]

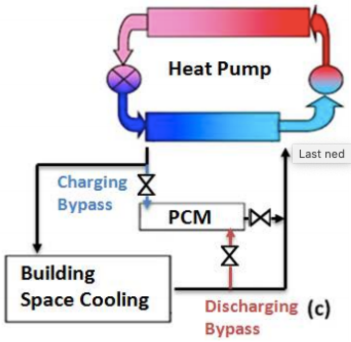


Figure 56: system HVL [84]

Figure 58: Flat-ICE container HVL Bergem [72]

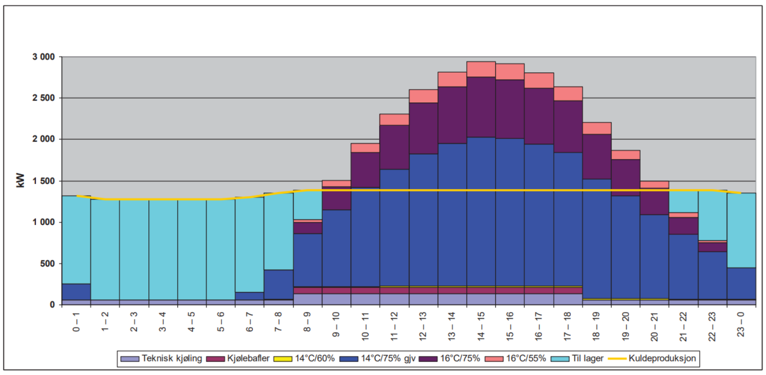


Figure 59: Cooling requirement for dimensioning day. Melting temperature 10°C [72]

Chapter 3. System design

This chapter shows the system used for the master thesis. This is the “state of the art” facility today. The system used for the master thesis (supermarket 1) is a new store and does not have logging data from the summer months. Data from an equal store was requested but was not received before the delivery of the master thesis. A second supermarket system (supermarket 2) was therefore needed to see how a typical store operates in the summer when the outdoor temperature is high. The two supermarket systems are shown in the chapters below.

3.1 System design supermarket 1

Based on the supermarket P&ID the main components are collected and made into a new simplified version. The new simplified P&ID is presented in Figure 60 and is the basis for further calculations and simulations. The system is a centralized transcritical CO_2 booster system for a supermarket located in Norway. The system is referred to as supermarket 1.

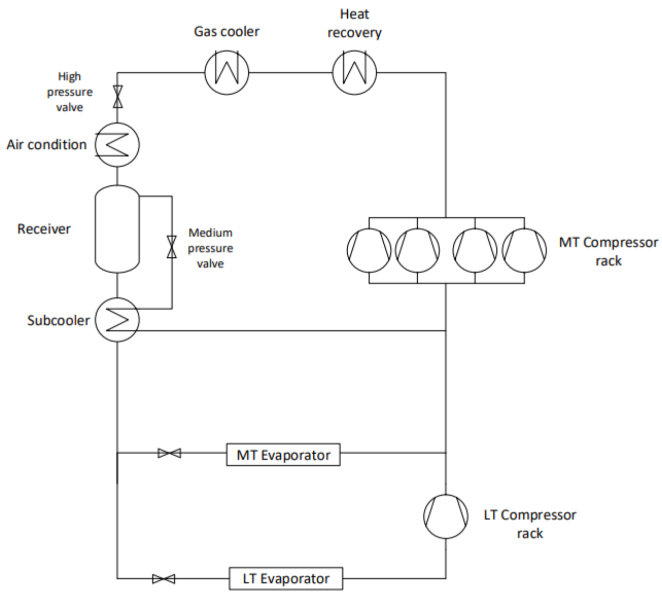


Figure 60: System design supermarket 1

The LT and MT cabinets are placed around the store and piping is collected in a technical room. The system has one LT compressor and four MT compressors. One of the compressors at each step is frequency controlled. The LT evaporator and MT evaporators have evaporation temperatures of -7 °C and -30 °C. This corresponds to LT pressure of 15 bar and MT pressure of 35 bar. The gas cooler outlet temperature is set to 29 °C and the pressure in the receiver is 40 bar. Table 7 shows data collected from the P&ID. These are the input values that will be used for further simulations and calculations.

The system has heat recovery before the gas cooler where heat is sent to the AHU. Excess heat can be used for the heating coil. The heat recovery will help reduce the temperature of the refrigerant in the CO_2–system and thereby increase the efficiency of the system. A heat exchanger for AC is implemented after the high-pressure valve (HPV).

Table 7: Data used for calculations and simulations

Characteristic	*Value*
Refrigerant	R744
Cooling capacity [kW]	75
Freezing capacity [kW]	4
LT evaporation temperature [℃]	-30
MT evaporation temperature [℃]	-7
Gas cooler outlet [℃]	29
Intermediate pressure [bar]	40

Figure 61 shows a simplified version of the total system. The blue compressor indicates the LT compressor and the four black compressors indicate the MT compressors. The CO_2 heat pump supplies the cabinets and freezer with refrigeration. It also supplies the AHU (air handling unit) with cooling and heating for the ventilation air. The heating and cooling for the AHU are supplied with separate pipes with Monoethylene glycol (MEG 30%). The solution requires a lot of piping.

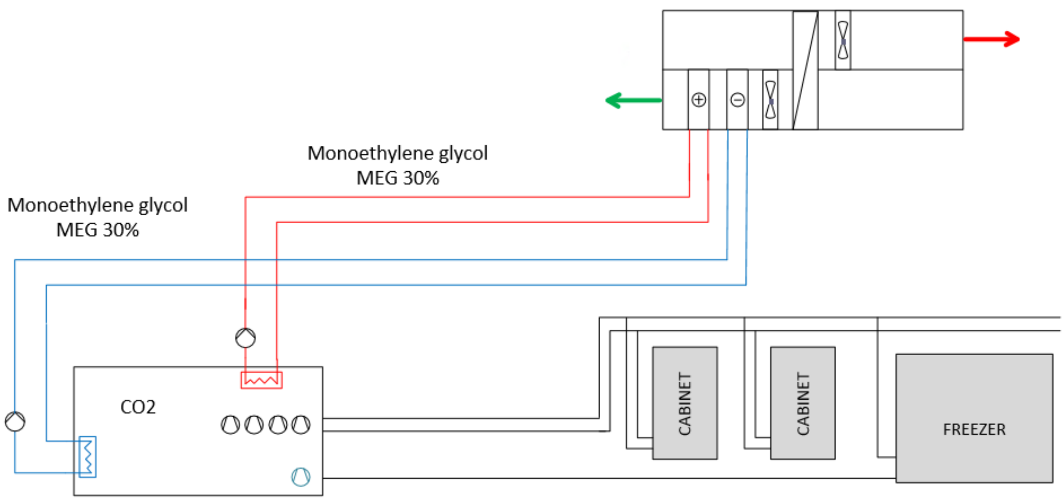


Figure 61: Simplified version of existing system

Compressors

The compressors used for the system are semi-hermetic reciprocating compressors. The compressors were listed in the refrigeration systems P&ID and information about the different compressors was collected from the website of Bitzer. Table 8 shows the cooling capacity and the power consumption of the individual compressors at 50 Hz and with the input given in Table 7. MT compressor number 1 and LT compressor number 1 have frequency control.

Table 8:Compressors supermarket 1

	Type	*Refrigeration capacity [kW]*	*Power consumption [kW]*
MT compressor 1	4PTE-6KC 40S (VSD)	10.43	4.3
MT compressor 2	4PTE-6KC 40S	10.43	4.3
MT compressor 3	4KTE-10KC 40S	25.3	9.6
MT compressor 4	4KTE-10KC 40S	25.3	9.6
LT compressor 1	2MME-0.7KB 40S (VSD)	3.7	1.7
Total		75	29

3.2 System design supermarket 2

The system for supermarket 2 is shown in Figure 62. The system has two LT compressors, three MT compressors and one parallel compressor. The cooling coil in the AHU is supplied with refrigeration from the CO_2 system. The piping for AC is connected between the gas cooler and the receiver. AC refrigeration is supplied with CO_2 and does not have a glycol heat exchanger as for supermarket 1. The parallel compressor is controlled by a valve (FBV). If the valve has an opening bigger than 26% then the valve closes and the parallel compressor turns on. Table 9 shows data collected from the P&ID for supermarket 2. These data will be used for further calculations.

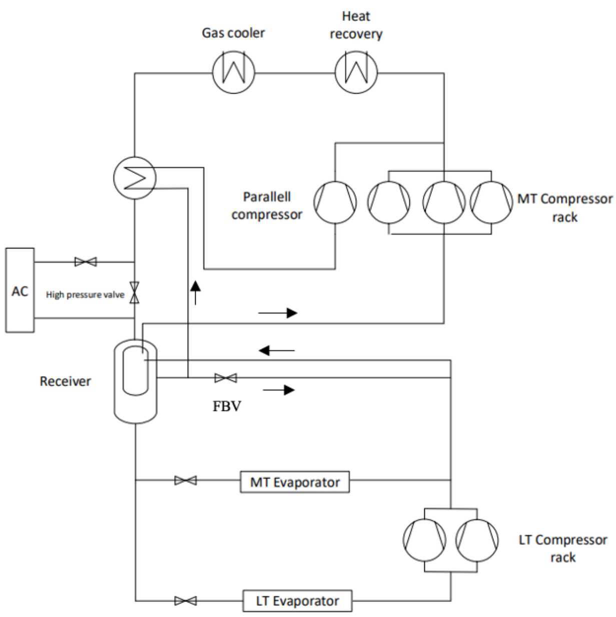


Figure 62: System design supermarket 2

Table 9: Data used for calculations and simulations supermarket 2

Characteristic	*Value*
Refrigerant	R744
Refrigeration capacity MT compressor [kW]	45
Refrigeration capacity parallel compressor [kW]	31
Refrigeration capacity LT compressor [kW]	10
AC [kW]	35
LT evaporation temperature [℃]	-30
MT evaporation temperature [℃]	-6
Gas cooler outlet [℃]	33
Intermediate pressure [bar]	38

Compressors

The compressors used for the system are semi-hermetic reciprocating compressors. The compressors were listed in the refrigeration systems P&ID and information about the different compressors was collected from the website of Bitzer. Table 10 shows the cooling capacity and the power consumption of the individual compressors at 50 Hz and with the input given in Table 9. MT compressor number 1 and LT compressor number 1 have frequency control.

Table 10:Compressors supermarket 2

	Type	***Refrigeration capacity [kW]***	***Power consumption [kW]***
MT compressor 1	4PTE-7K (VSD)	9.5	4.6
MT compressor 2	4KTE-10K	23.2	10.5
MT compressor 3	4MTE-10K	15.4	7.3
LT compressor 1	2KME-1K (VSD)	4.8	1.4
LT compressor 2	2KME-1K	4.8	1.4
Parallel compressor	4JTE-15K (VSD)	32.1	10.1
Total		90	35

3.3 Compressor combinations

Compressor configurations based on the existing system

Based on the information given about the system (supermarket 1) in chapter 3.1 tables for MT compressor configurations were made in excel. All possible combinations for compressors are listed in Table 11. The results are also presented graphically in Figure 63.

Compressor configurations based on new capacity control system from Bitzer

The newly developed BITZER system mentioned in chapter 2.6 allows for reduction of the capacity down to 10%. Due to lower part-load operation one can either have a higher maximum capacity (bigger MT1) or reduce the number of compressors. All possible combinations for compressors are listed in Table 13 The results are also presented graphically in Figure 64.

3.3.1 Compressor configuration for supermarket 1

Table 11 shows the different possible combinations of the MT compressors. At the minimum value the frequency-controlled compressor MT1 runs at minimum speed. At the maximum value the frequency-controlled compressor MT1 runs at maximum speed. Figure 63 shows the compressor steps presented in a bar chart. The dark blue colour indicates the minimum capacity where the frequency-controlled compressor runs at minimum frequency. The dark blue and orange added together represent the total MT1 when run at maximum frequency. Table 12 shows the intervals for the different compressor steps. Table 12 shows that the maximum MT load the system can have before having to use all four compressors is 64 kW (without compressor MT2).

Table 11: Compressor steps

Steps	*Combination*	*Capacity_min [kW]*	*Capacity_max [kW]*	*Difference next step*
Step 1	MT1	4	14	
Step 2	MT1+MT2	15	24	1
Step 3	MT3	25	25	1
Step 4	MT1+MT3	30	39	4
Step 5	MT1+MT2+MT3	40	49	1
Step 6	MT3+MT4	51	51	1
Step 7	MT1+MT3+MT4	55	64	4
Step 8	MT1+MT2+MT3+MT4	65	75	1

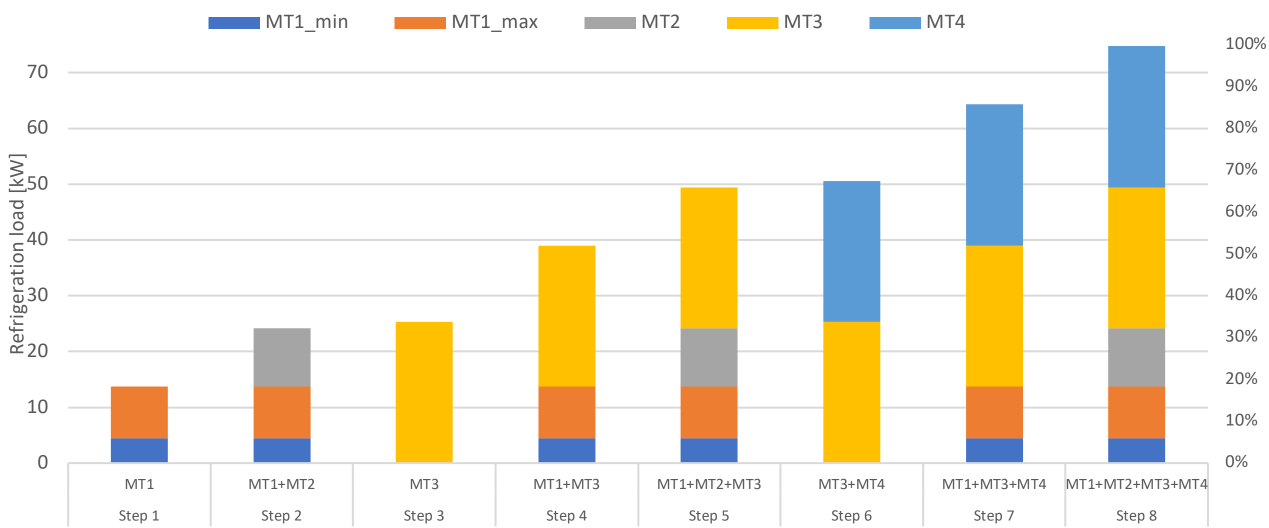


Figure 63: Compressor steps supermarket 1

Table 12: Compressor intervals for supermarket 1

Steps	*Interval [kW]*	*Min percentage*	*Max percentage*
1	4-14	6 %	18 %
2	15-24	20 %	32 %
3	25	34 %	34 %
4	30-39	40 %	52 %
5	40-49	54 %	66 %
6	51	68 %	68 %
7	55-64	74 %	86 %
8	65-75	88 %	100 %

3.3.2 New compressor system from Bitzer

The compressor configuration with the new Bitzer system is shown in Table 13 and Figure 64. The new combinations of compressors shows that there is less difference between the steps. The minimum value of the next step is closer to the maximum of the last step. Some of the steps also overlap. Table 14 shows that the maximum MT load the system can have before having to use all four compressors is still 64 kW (without compressor MT2). Assuming that the absolute minimum load of the system should be the same, MT1 can have a larger maximum load with the new compressor system. The compressor can be up to two times larger and still have the same minimum load. Table 14 shows the intervals for the different compressor steps.

Table 13: Compressor steps with new Bitzer system

Steps	Combination	Capacity_min [kW]	Capacity_max [kW]	Difference next step
Step 1	MT1	1.4	14	
Step 2	MT1+MT2	12	24	-2
Step 3	MT3	25	25	1
Step 4	MT1+MT3	27	39	1
Step 5	MT1+MT2+MT3	37	49	-2
Step 6	MT3+MT4	51	51	1
Step 7	MT1+MT3+MT4	52	64	1
Step 8	MT1+MT2+MT3+MT4	62	75	-2

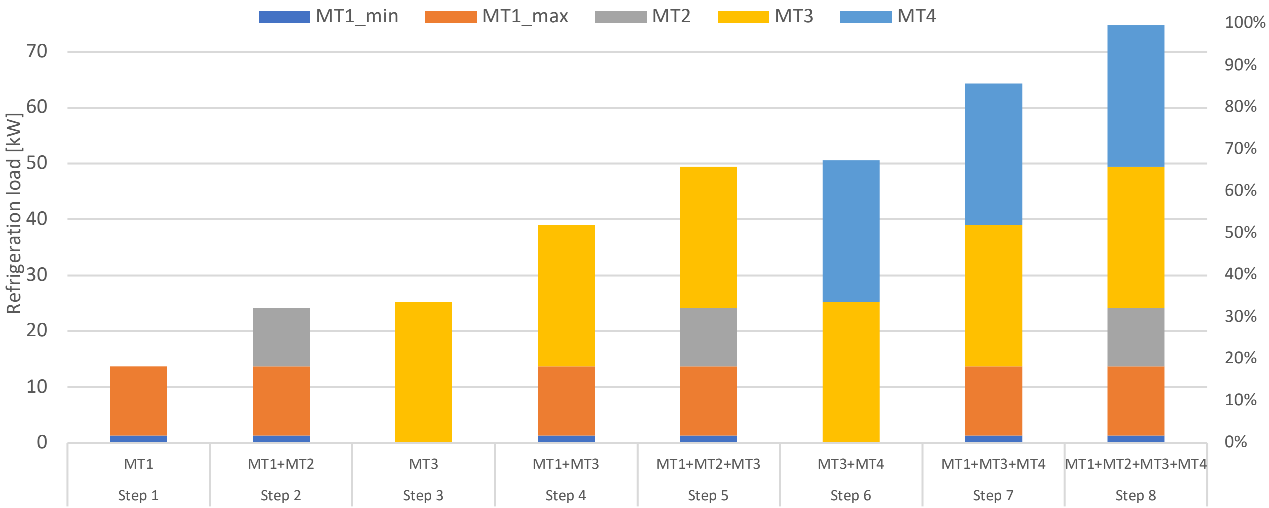


Figure 64: Compressor steps new Bitzer system

Table 14: Compressor intervals for new Bitzer system

Steps	Interval [kW]	Min percentage	Max percentage
1	1-14	2 %	18 %
2	12-24	16 %	32 %
3	25	34 %	34 %
4	27-39	36 %	52 %
5	37-49	50 %	66 %
6	51	68 %	68 %
7	52-64	70 %	86 %
8	62-75	84 %	100 %

Both solutions show that the maximum MT load the system can have before having to use all four compressors is 64 kW (without compressor MT2). This corresponds to a total load of 68kW with the LT compressor. For the new Bitzer compressor system the minimum load is reduced to 10%. Using this system for the frequency-controlled compressor allows for smaller gap between the steps. The system also allows for use of a bigger compressor as MT1. Combining this system with thermal storage can make it possible to reduce the number of compressors.

Chapter 4. Data processing and load curves

Data from two existing supermarket systems are collected to see how a typical store operates through the year and to be able to make load curves that will be used for further simulations of the system given in chapter 3.1.The stores are located outside Oslo and both have refrigeration systems that use R744. The data from supermarket 1 and supermarket 2 is simulated in the Bitzer software shown in chapter 4.1. Logging data from the facilities are also processed in excel and presented in graphs in chapter 4.2 and chapter 4.3. As mentioned, supermarket 1 is a new store and does not have logging data from the summer months. Therefore only winter data is presented for supermarket 1. A second supermarket system (supermarket 2) is needed to see how a typical store operates in the summer when the outdoor temperature is high. Data from the two supermarkets are shown in the chapters below.

4.1 Bitzer software simulation

Bitzer has its own Bitzer software where it is possible to perform simplified simulations for refrigeration systems. The software allows for selection of compressor types, cooling load, temperatures, etc. Based on the inputs the program runs simulations and makes a simulation report that can be downloaded.

Data given in Chapter 3 was added to the program. Each compressor was added with the correct compressor model and evaporation temperatures for cabinets and freezers were added. All input values for the simulations can be found in appendix A. For supermarket 1 there is performed three simulations. In the first simulation the compressors with frequency control are run at minimum frequency. In the second simulation all compressors are run at 50 Hz. In the third simulation the compressors with frequency control are run at maximum frequency. Appendix A shows the whole simulation. For supermarket 2 the simulation is performed for the "worst case scenario" which is a summer day in the middle of the day. The chosen day was 19. July 2020.

4.1.1 Results Bitzer simulation supermarket 1

There are limited options for refrigeration system solutions in the software. It was not possible to model the sub cooler as given in the P&ID. The Bitzer model is shown in Figure 65. The Bitzer software gave the following P-h diagram for the system shown in Figure 66. The blue lines show the LT circuit. The refrigerant is compressed in the LT compressor from 1 to 2 and is then subcooled before mixing with the refrigerant coming from the MT evaporators and receiver. Refrigerant is then compressed to HP level before entering the gas cooler. After the gas cooler the refrigerant goes through the HPV before entering the receiver in point 7. In the receiver the liquid part of the refrigerant is fed to the evaporators and the gas is sent through the FBV and to the MT compressors.

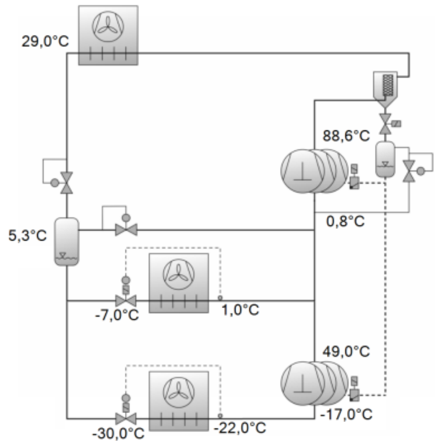


Figure 65: Bitzer model

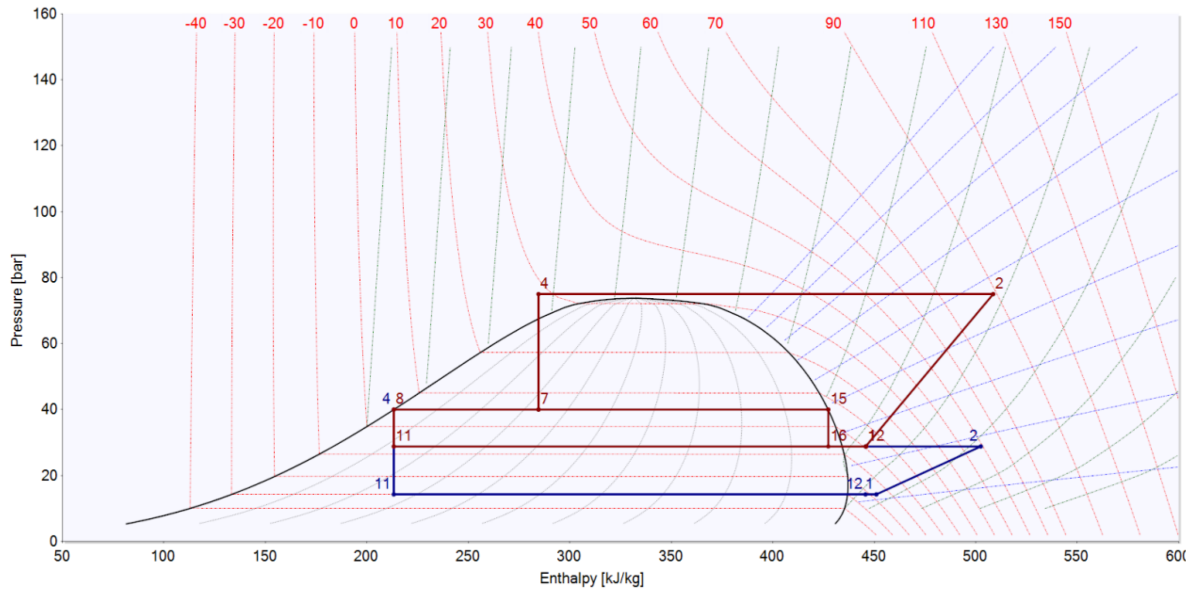


Figure 66: P-h diagram from Bitzer simulation

From the simulations the COP is approximately 2.3 for all cases. In the first simulation the compressors with frequency control (MT1) are run with minimum frequency and thereby MT1 shows the minimum load of the system. In the third simulation the compressor with frequency control is run with maximum frequency. Thereby showing the maximum load for the system. Table 15 shows the results from the three simulations. When run at 25 Hz the MT1 has an evaporation capacity of 4.39kW and when run at 75Hz the MT1 has an evaporation capacity of 13.68kW

Table 15: Results simulations Bitzer

	Total MT	*MT1*	*MT2*	*MT3*	*MT4*
Sim1					
Frequency		25 Hz			
Evaporator capacity [kW]	61.8	4.39	9.80	23.8	23.8
Gas cooler capacity [kW]	91.6	6.51	14.54	35.3	35.3
Power input [kW]	25.8	2.27	4.29	9.63	9.63
Mass flow [kg/h]	1473	104.7	234	567	567
Sim2					
Frequency		50Hz			
Evaporator capacity [kW]	65.8	9.60	9.61	23.3	23.3
Gas cooler capacity [kW]	99.4	14.49	14.50	35.2	35.2
Power input [kW]	27.9	4.31	4.29	9.63	9.63
Mass flow [kg/h]	1597	233	233	566	566
Sim3					
Frequency		75Hz			
Evaporator capacity [kW]	68.5	13.68	9.37	22.7	22.7
Gas cooler capacity [kW]	106	21.2	14.50	35.2	35.2
Power input [kW]	29.7	6.14	4.29	9.63	0.63
Mass flow [kg/h]	1696	339	232	563	563

4.1.2 Results Bitzer simulation supermarket 2

Bitzer simulations for supermarket 2 are performed for a summer day. Data from the facility is collected for 19. July 2020. All input values for the simulation can be found in appendix B There are limited options for refrigeration system solutions in the software. It was not possible to model the tank as given in the P&ID. The Bitzer model is shown in Figure 67. The Bitzer software gave the following P-h diagram for the simplified system shown in Figure 68.

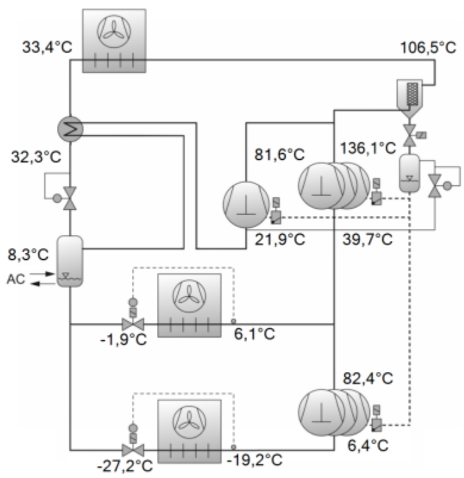


Figure 67: Bitzer model supermarket 2 – worst case scenario

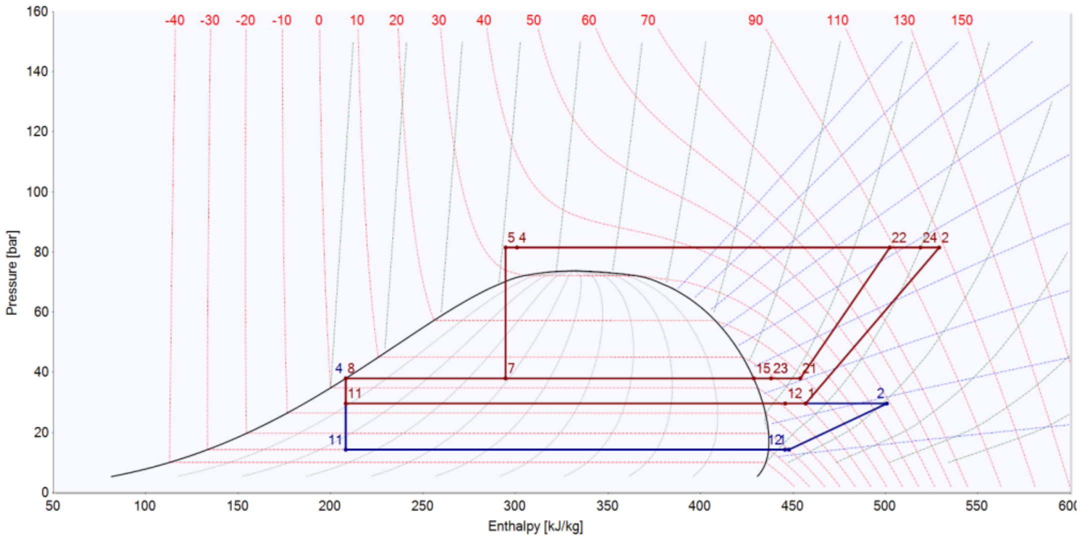


Figure 68: P-h diagram from Bitzer simulation supermarket 2

Table 16 shows results from the simulation. As one can see, there is one compressor that is not running on the given day (MT3). As one can see the parallel compressor (PC) is running at 28.4 kW which is 88% of maximum. The simulation is performed for a summer day and the amount of flash gas will therefore be high. When the PC is running the load on the MT compressors will be reduced. The entire simulation can be found in appendix B. The COP for the system at the given conditions is 2.4.

Table 16: Results Bitzer simulations summer day supermarket 2

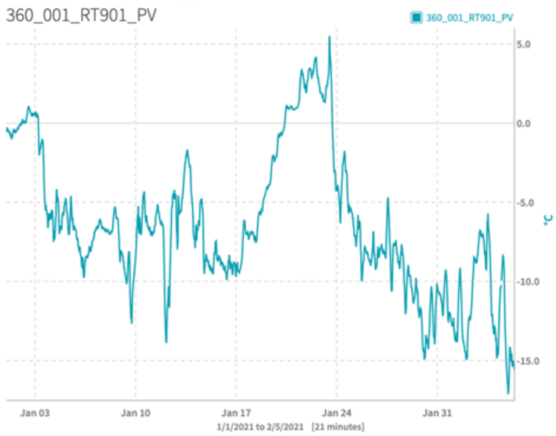
	Total MT	*MT1*	*MT2*	*MT3*
Frequency		49 Hz		
Evaporator capacity [kW]	23.8	8.92	14.84	-
Gas cooler capacity [kW]	96.2	15.42	25.6	-
Power input [kW]	11.78	4.54	7.24	-
Mass flow [kg/h]	565	212	353	-

	Total LT	*LT1*	*LT2*
Frequency		70 Hz	
Evaporator capacity [kW]	11.39	6.67	4.72
Power input [kW]	3.11	1.86	1.26
Mass flow [kg/h]	182.3	106.8	75.5

	Total PC
Frequency	
Evaporator capacity [kW]	28.4
Power input [kW]	11.28
Mass flow [kg/h]	1 057

4.2 Data processing supermarket 1 – winter

The store is located outside Oslo. Since the supermarket has just opened, there are only logging data from January and February 2021. The data shown in the figures below are collected for 04. January 2021. Figure 69 shows the outside temperature for the location (left) and inside temperature (right) for January 2021.



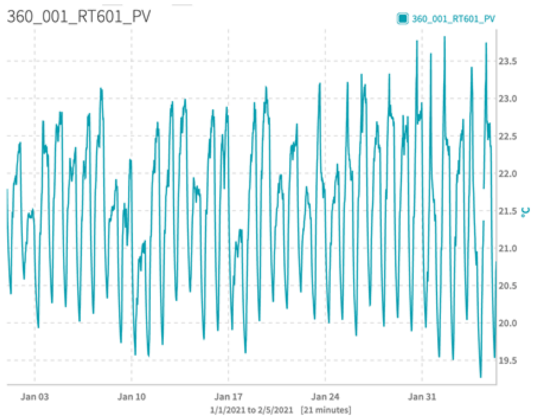


Figure 69: Outside temperature (left) and inside temperature (right) for supermarket 1 January 2021

4.2.1 Results data processing supermarket 1

From Figure 70 one can see that the MT evaporators have good evaporation temperature and regulation. However, one can see that the load on the system is quite low compared to the maximum load. The system has a maximum running capacity of 30% for the given day. Logging for the rest of the month shows similar numbers as for the day given in Figure 70. Figure 71 shows that the LT compressors have some trouble with the regulation. This could be a result of too big compressor. The Compressor turns frequently on and off. The gas cooler runs stable during the day with a pressure of ca 75 bar.

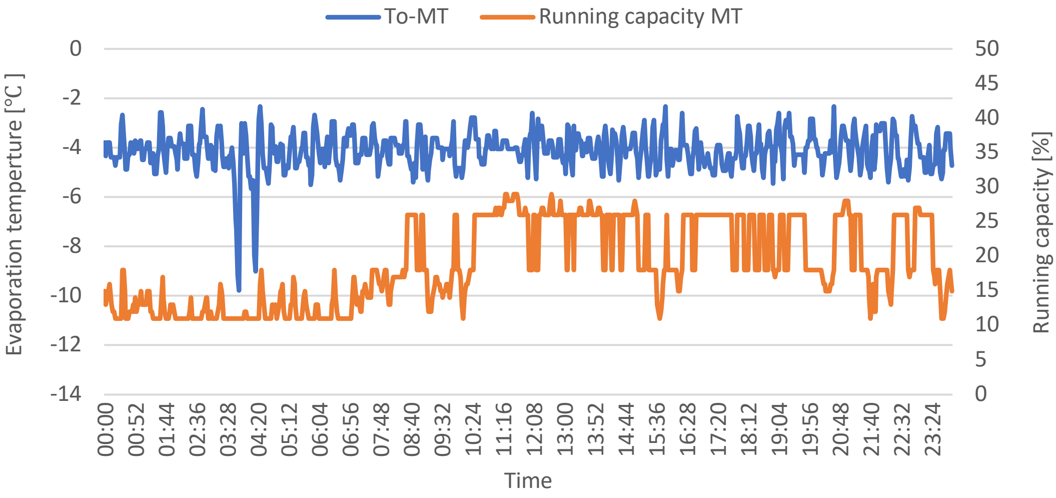


Figure 70: Evaporation temperature and running capacity MT 04.01.2021

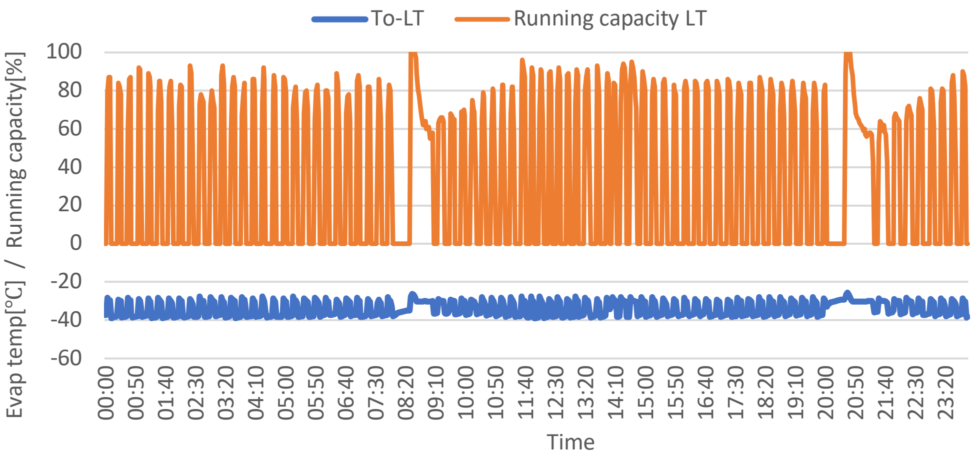


Figure 71: Evaporation temperature and running capacity LT 04.01.2021

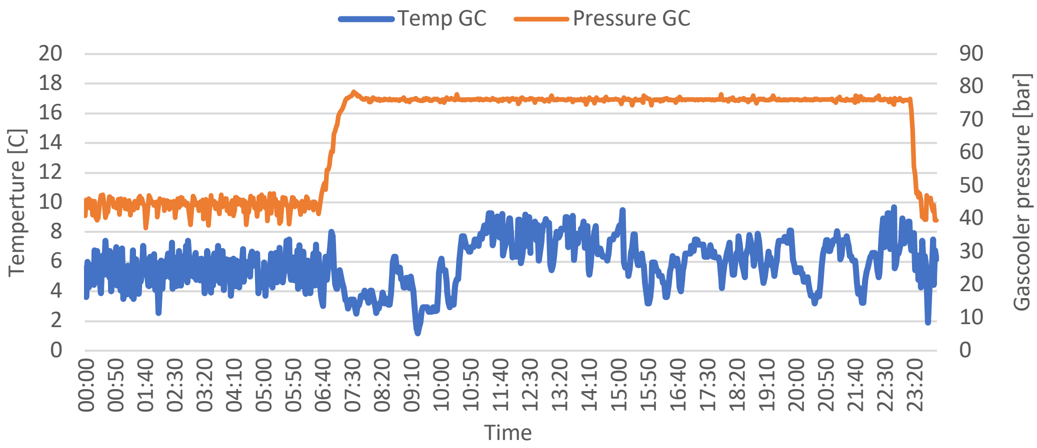


Figure 72: Temperature and pressure gas cooler 04.01.2021

4.3 Data processing supermarket 2 – winter and summer

Data is collected from supermarket 2 which has logging data for an entire year. This is important to see how the store operates in the summer months when the load is at its highest. Supermarket 2 has a little bit different system solution than supermarket 1. However, one can assume that the loads on the stores will be similar during the summer months and that supermarket 2 will be representative for the loads on supermarket 1.

4.3.1 Method for calculating output values for supermarket 2

Supermarket 2 has logging data for a lot of the components but not all of them. The logging data received was inadequate to produce load curves for simulations. The PC did not have logging and further calculations were needed to obtain the data needed. The method for calculation and production of data will be supplied in the following sub chapter. The summer day with the highest outside temperature was chosen for further calculations. The day chosen was 19. July 2020. This day will be representative for the maximum load on the facility. All calculations are done per hour of the given day.

Bitzer software simulations

The Bitzer software mentioned in chapter 4.1 is used to obtain information about the different compressors. One simulation is performed for each compressor and data like maximum stroke volume is collected for further calculations. There is also performed one simulation for the total system. Results from the simulation are shown in chapter 4.1.2.

Calculating AC load

The airflows and temperatures in the AHU are logged for the given day. AC load is calculated from equation 5. Where $\dot{Q}$ is the energy need in kW, V is airflow in m^3/s, ρ is density in kg/m^3 and Δh is enthalpy difference in kJ/kg. The density is set to 1.2. The density will vary, but the variations are so small that it will not affect the final results significantly [73]. The Mollier diagram in Appendix E is used to obtain the enthalpy difference Δh. The AC load for the different hours is shown in Figure 80.

$$\dot{Q} = V * \rho * \Delta h \; [kW] \tag{5}$$

Calculating cooling capacities and mass flows

The total energy consumption of the heat pump system is known. The running capacity for the LT and MT compressors are given and energy needs can be calculated using equation 6 and equation 7. Where Q is the energy need in kW, ṁ is mass flow in kg/s, c_p is specific heat in J/kgK and ΔT is temperature difference in K. The stroke volume of the different compressors was given from the Bitzer software simulations. The values for density and specific energy for the given state of the refrigerant were collected from an online calculator.

$$\dot{m} = \; stroke\ volume * running\ capacity * \; \rho \tag{6}$$

$$Q = \dot{m} * c_p * \Delta T \tag{7}$$

Calculating contribution from parallel compressor

The running capacity for the parallel compressor is not given from the logging data. The parallel compressor turns on if the opening of the FBV is more than 26%. Then the FBV closes, and the PC compresses the flash gas. The power need for the PC has to be calculated. It can be found by using equation 8 or 9. Using equation 8: The total load on the system is known and by subtracting all loads except for the load on the parallel compressor, one can find the load on the parallel compressor.

Using equation 9: All the CO_2 is evaporated after passing through the AC HX. The mass flow through the parallel compressor is the mass flow through the AC HX and the flash gas after the HPV. When the mass flow is known, equation 10 can be used to obtain the power need for the parallel compressor. The P-h diagram is used to find the Δh. Figure 74 shows an example for the system for one specific hour.

$$Paralell\ compressor = \ total\ energy\ use - LT\ compressors - MT\ compressors - other \tag{8}$$

$$\dot{m}_{parallel\ compressor} \ = \ \dot{m}_{flas\ store} \ + \ \dot{m}_{AC} \tag{9}$$

$$Q = \dot{m} * \Delta h \tag{10}$$

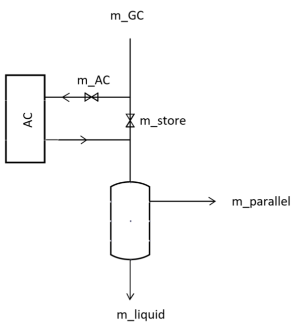


Figure 73: mass flows in system

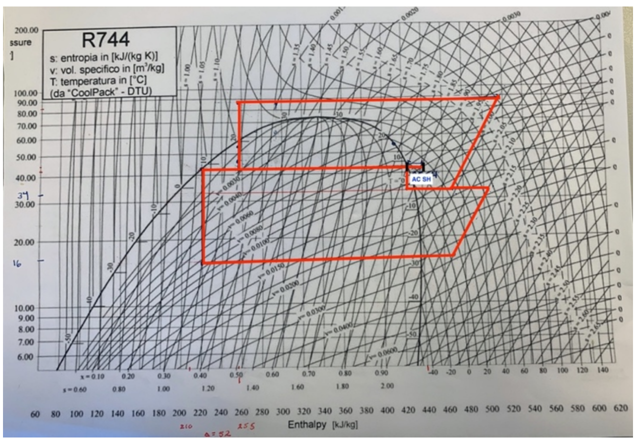


Figure 74: P-h diagram

4.3.2 Results data processing supermarket 2 – winter

This chapter shows the results for data processing for supermarket 2 during a winter day. The day chosen is the same as for supermarket 1 which is 04. January 2021. Figure 75 shows the outside temperature for the given day. Figure 76 shows the running capacity for the LT compressors and Figure 77 shows the running capacity for the MT compressors.

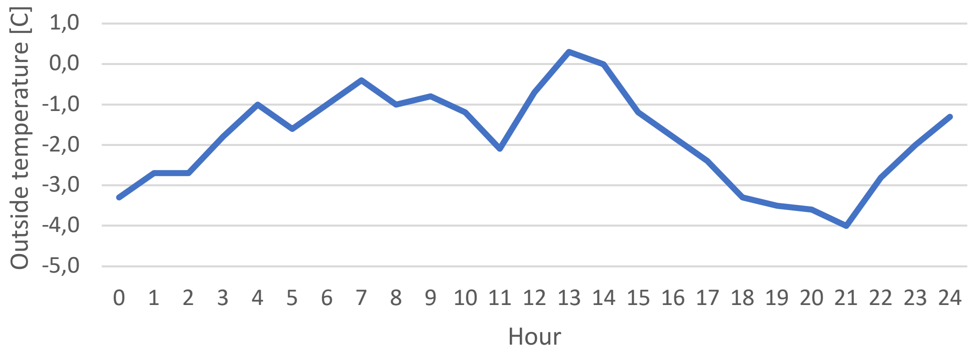


Figure 75: Outside temperature 04.01.21

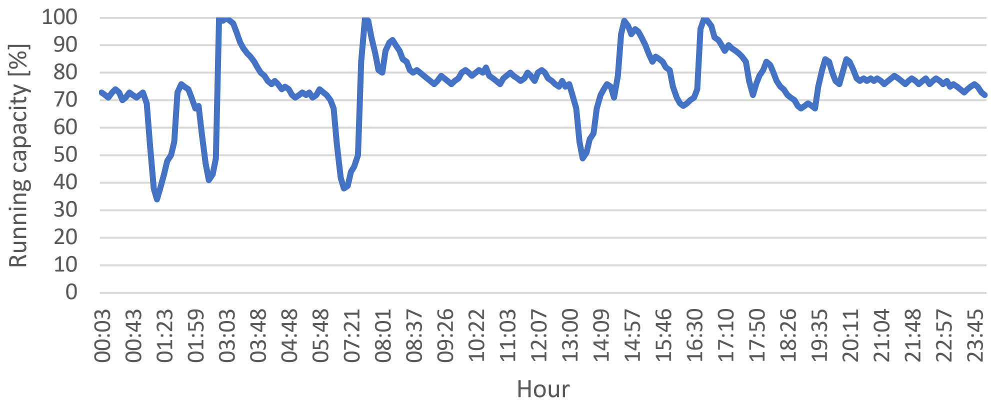


Figure 76: LT compressor running capacity 04.01.21

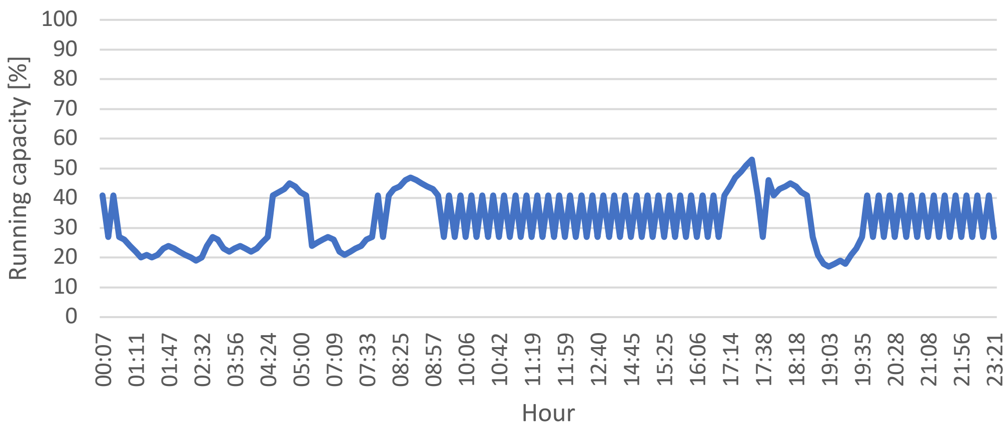


Figure 77: MT compressor running capacity 04.01.21

4.3.3 Results data processing supermarket 2 - summer

This chapter shows the results for data processing for supermarket 2 for 19. July 2020. Figure 78 shows the outside temperature and Figure 79 shows the running capacity for the LT and MT compressors. The MT compressor runs stable between 40 and 50%. Figure 80 shows the AC load curve obtained from the calculations. The AC load increases from 7 to 12 and becomes zero at 24.

Figure 81 shows the load curve for the total system. The different colours indicated different loads. Purple is LT compressors; blue is MT compressors and red is the parallel compressor. For the given day only two of three MT compressors are running. The third MT compressor (MT3) has only 100 running hours during the year. The parallel compressor turns on if the opening of the FBV is more than 26%. The parallel compressor turns on at 7 in the morning and runs until 23 in the evening.

Figure 82 shows the difference between winter and summer load for the system. For the winter load one can see the two peaks as described in chapter 2.5. One in the morning when new goods are loaded and one in the afternoon when people are grocery shopping after work. For the summer the load increases from 7 to 12 and stays high during the day. One can see hints of the two peaks. However the outside temperature will have a strong influence on the load

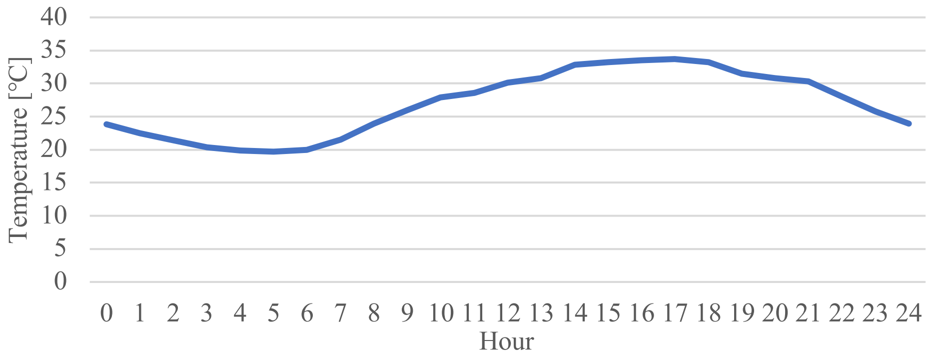


Figure 78:Outside temperature

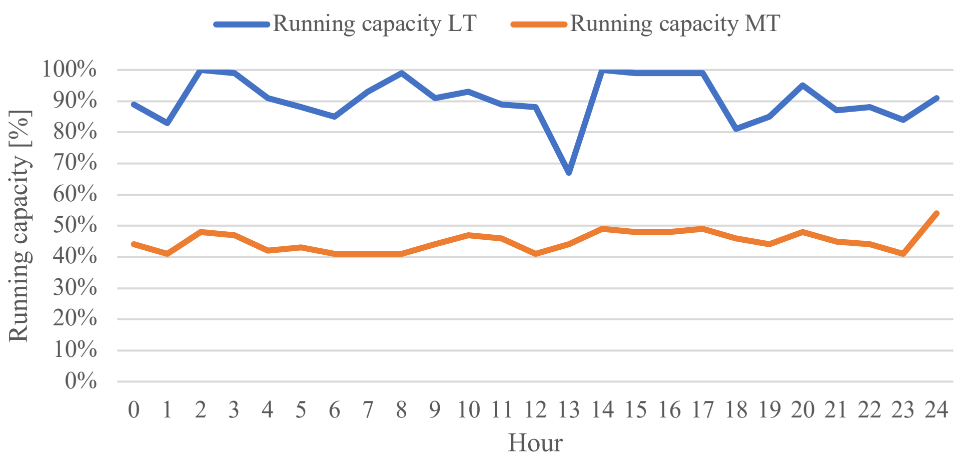


Figure 79: Running capacity LT and MT compressors supermarket 2 19.07.2020

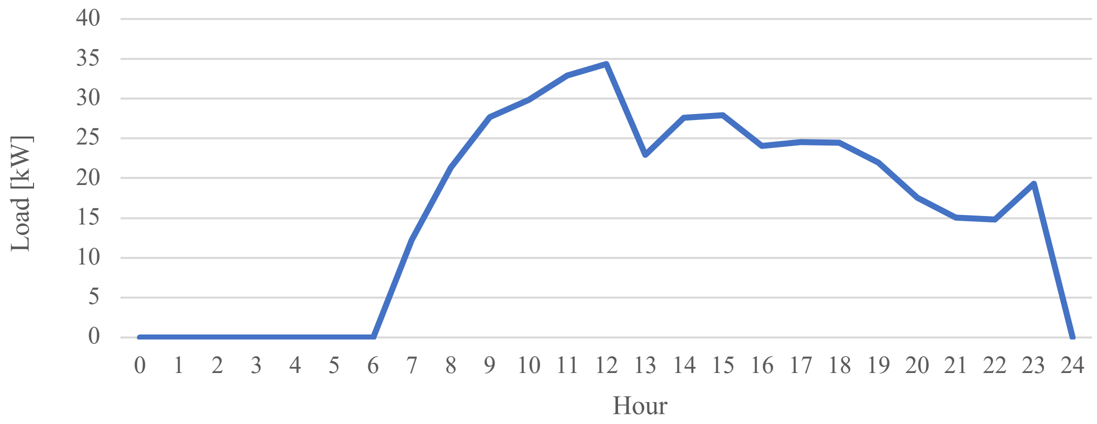


Figure 80: AC load for supermarket 2 19.07.2020

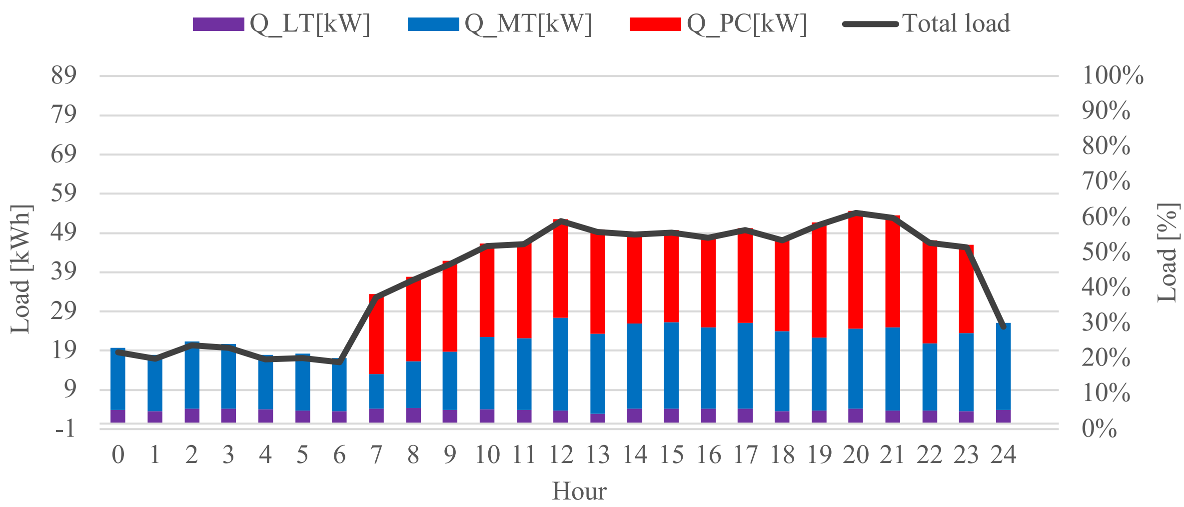


Figure 81: Load curve for supermarket 2 19.07.2020

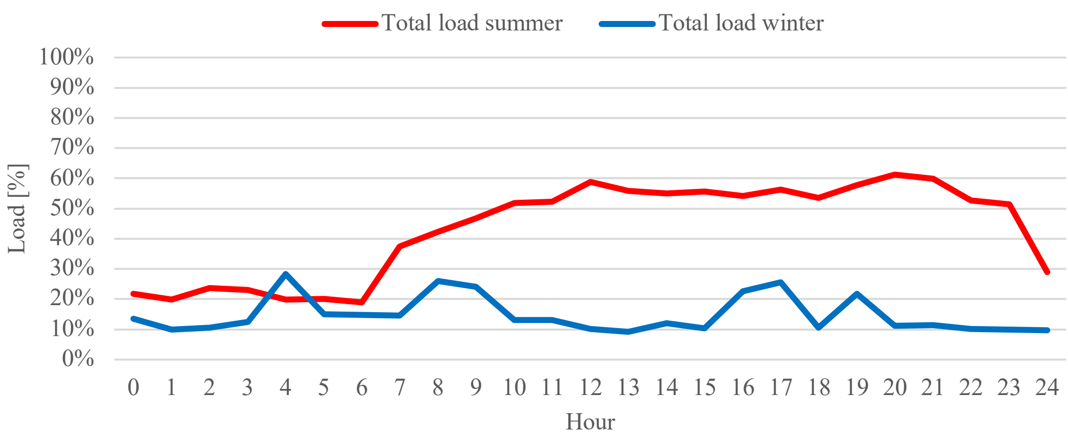


Figure 82: Summer and winter load on system

4.4 Load curves

The figures in this chapter show the load curves that will be used for further calculations and simulations on the system. The load curves are based on information about the system given in Chapter 3, data processing from Chapter 4 and the literature review. Assumptions for the graphs are made in collaboration with Dr. Ángel Álvares Pardiñas at Sintef. The figures show the refrigeration load given in kWh and percentage of full load. There are created three load curves for different periods of the year: January, April, and July. The simulations will be performed for these different periods. The graphs show the load for LT evaporators(purple), MT evaporators (dark blue) and AC (light blue). The grey line also shows the percentage of full load for the system. January and April have no AC load. The opening hours are set from 7:00 to 23:00.

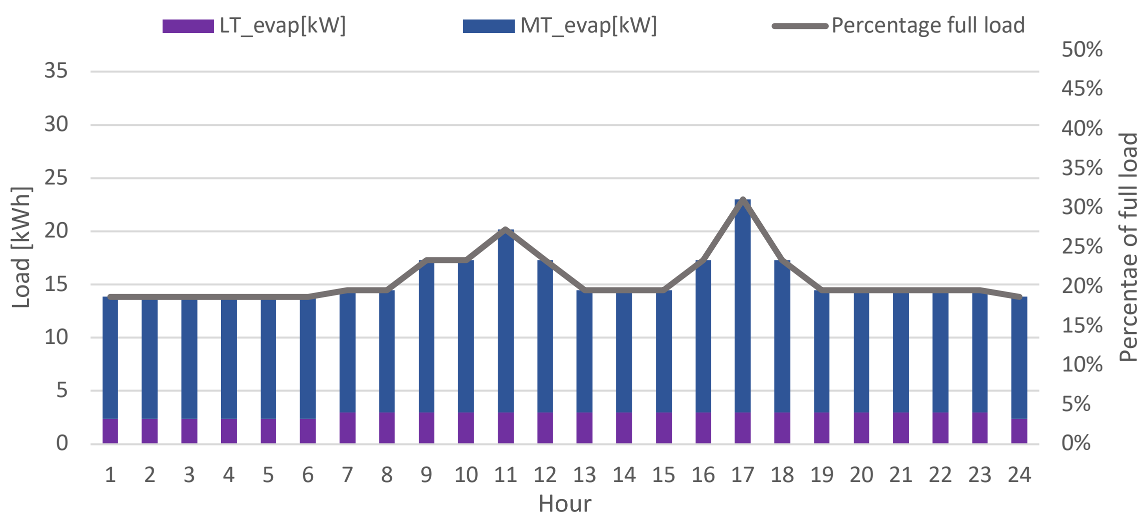


Figure 83: Refrigeration load without storage January

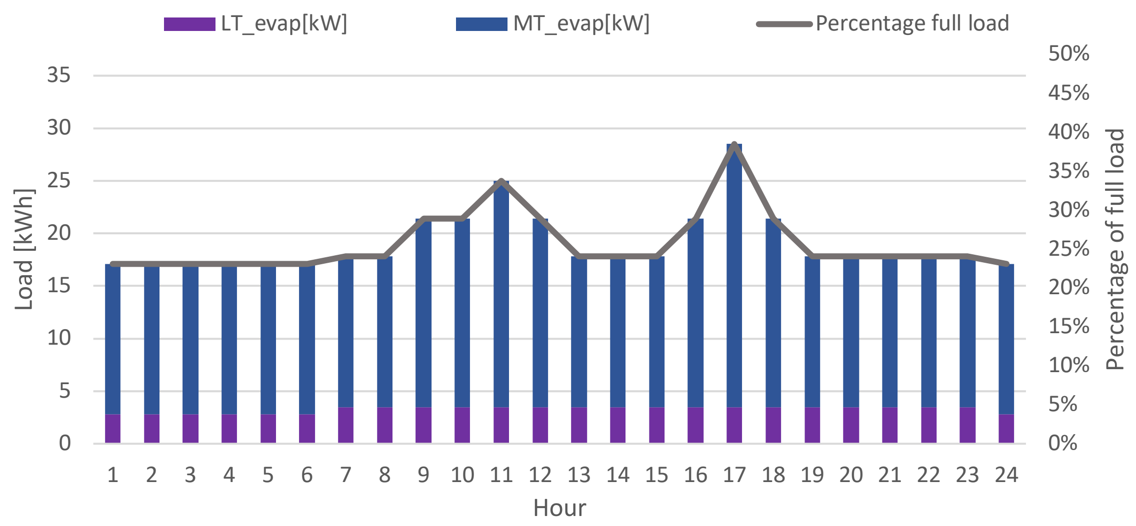


Figure 84: Refrigeration load without storage April

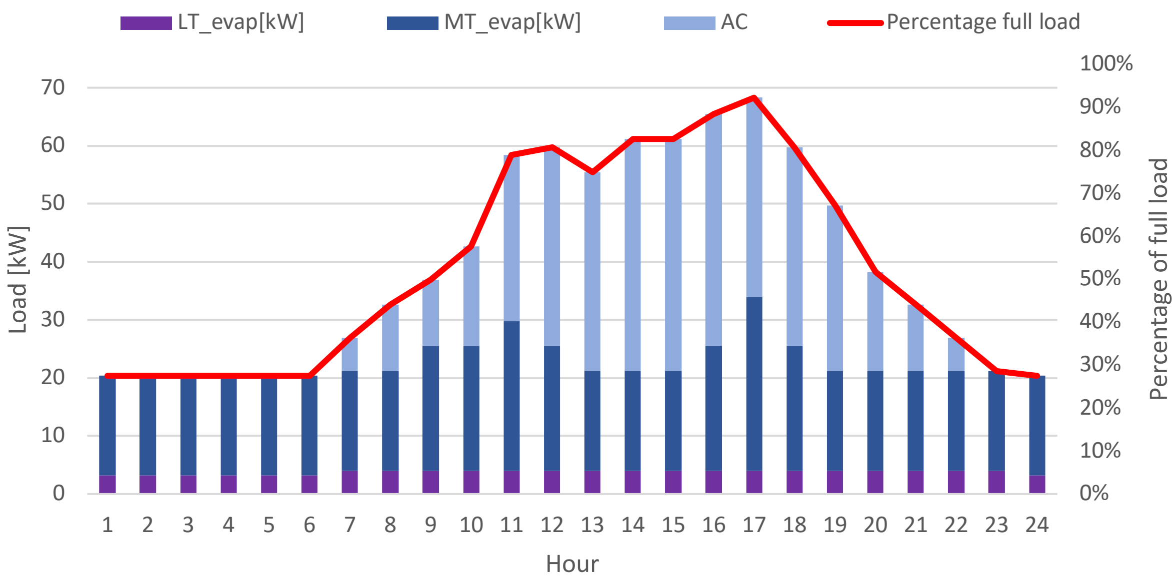


Figure 85: Refrigeration load without storage July

Chapter 5. CTES design and implementation

This chapter shows the new system with implementation of AC storage. Figure 86 and Figure 87 show the existing system (supermarket 1) corrected to fit the new system. The X's indicates the components that are removed or changed. As one can see the heat exchanger for AC will be removed and a lot of the components in connection with that circuit. This includes the glycol loop with MEG30% for the cooling coil in the AHU shown in Figure 87. Also, it may be possible to remove or change one of the compressors. This will be investigated further later in the report.

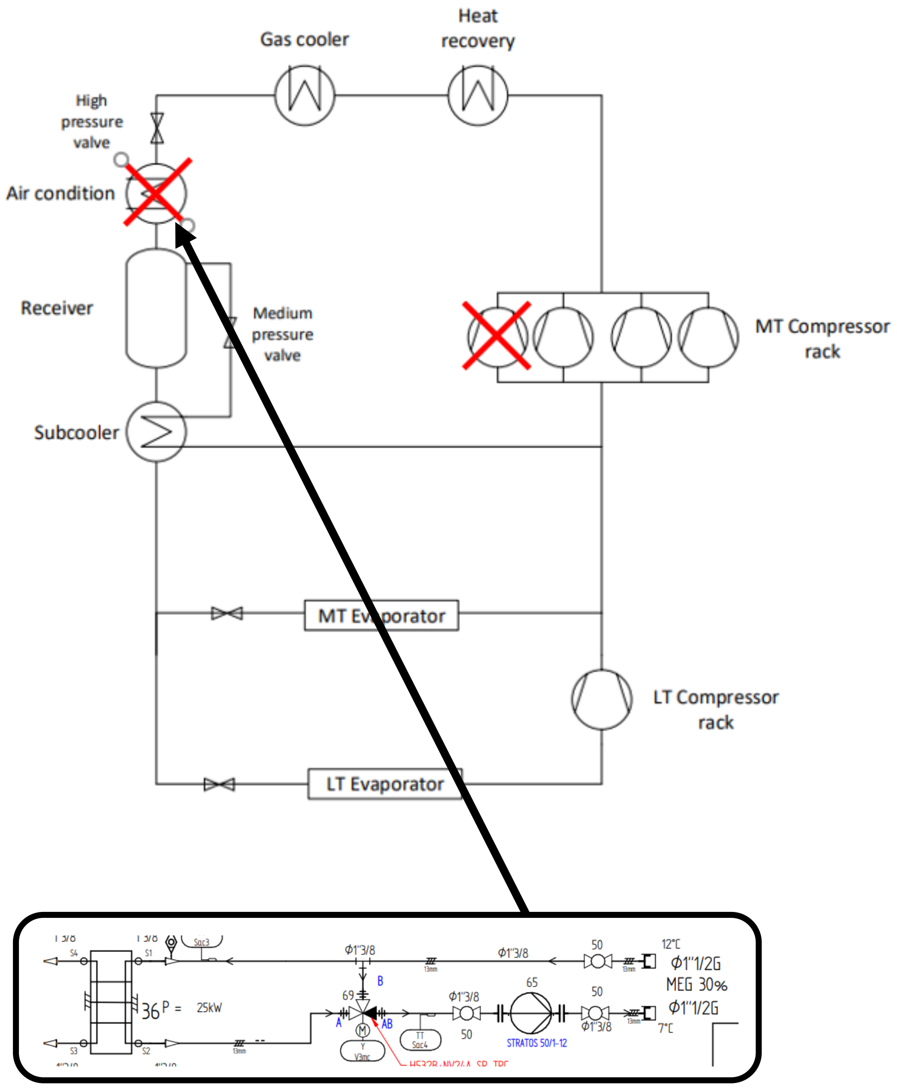


Figure 86: Existing system corrected to fit the new system

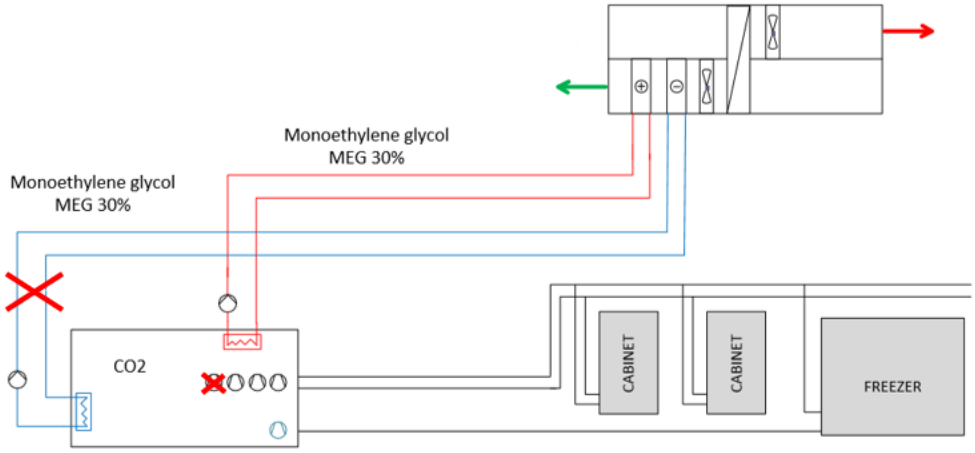


Figure 87: Existing system corrected to fit the new system

Figure 88 shows the new system with PCM storage. The PCM storage and cooling battery will be supplied by the same loop as the MT cabinets. There is no need for a separate glycol loop supplying the cooling battery. During charging the CO_2 circulates through the storage. It is possible to charge the storage and supply the cooling coil at the same time as indicated with the two arrows. During discharging the storage and cooling coil will be disconnected from the rest of the system and will be self-circulating. The self-circulating system is shown in Figure 89.

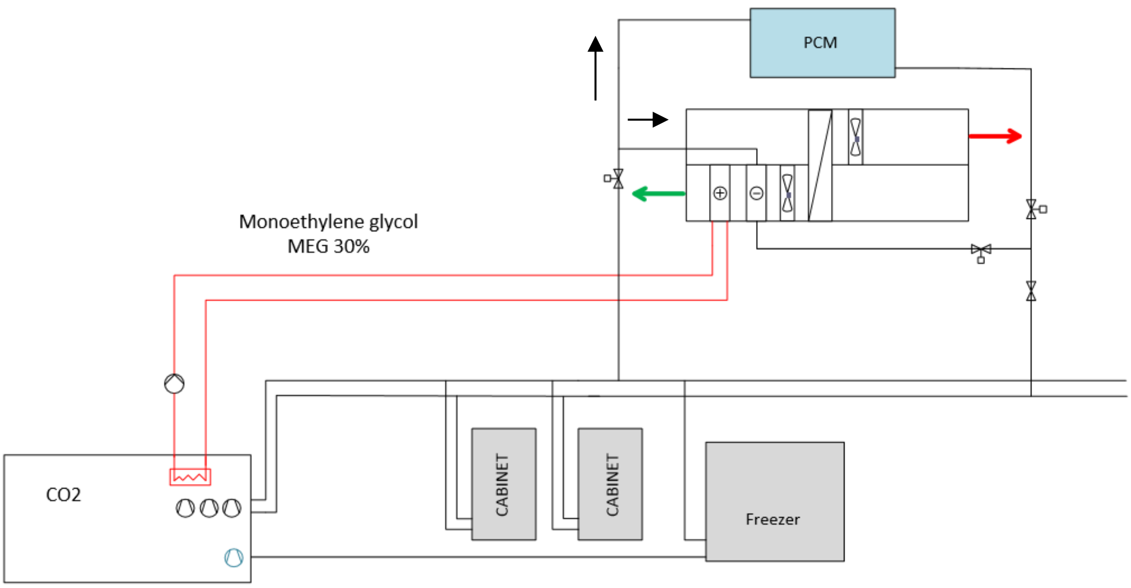


Figure 88: New simplified system with PCM storage

Self-circulating system

The PCM loop can be disconnected from the rest of the system during peak hours. This will reduce the total load on the system during hours of very high demand. The loop is shown in Figure 89. The loop will supply refrigeration for the Air condition system. During discharging the system will be self-circulating using gravity. This is referred to as a natural circulating system in Chapter 2.4. The refrigerant evaporates in the cooling coil and rises up. Then the refrigerant enters the PCM tank and condenses as it is cooled by the melting PCM. The refrigerant passes through the expansion valve and into the cooling coil in AC system and the cycle is repeated.

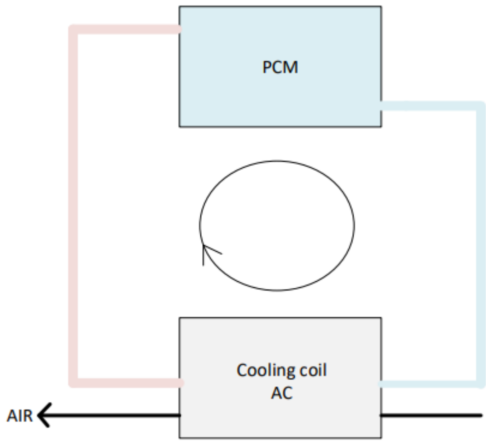


Figure 89: Self circulating PCM system

The system uses the thermosiphon principle that is defined by natural convection. This occurs due to the density difference in the refrigerant. The pressure difference between the two heat exchangers will drive the cycle. [67] Manescu et.al give the following equation for the pressure of the driving force [67]:

$$\frac{\vec{F}_A}{A} = \Delta p = (\rho_l - q_v) * \vec{g} * \Delta z \tag{11}$$

In the formula Δp is the effective pressure of the driving force ($\vec{F}_A$) relative to the cross-sectional area of the pipes (A). Δz = height difference, $\vec{g}$ = gravitational acceleration, $\rho_l - q_v$ = density difference between liquid and gas.

Placement

Placement of the storage will be important for the self-circulation to work. From equation 11 Δz is the height difference. The storage needs to be placed above the AC unit and the Δz has to be big enough. One should also avoid many bends and elbows in the loop. The storage will be placed in a container on the roof of the building. This way the storage will not occupy space inside the supermarket. The placement is shown in Figure 90. The roof is not used for anything specific, so placing the storage here will be a good solution for space reduction inside the supermarket.

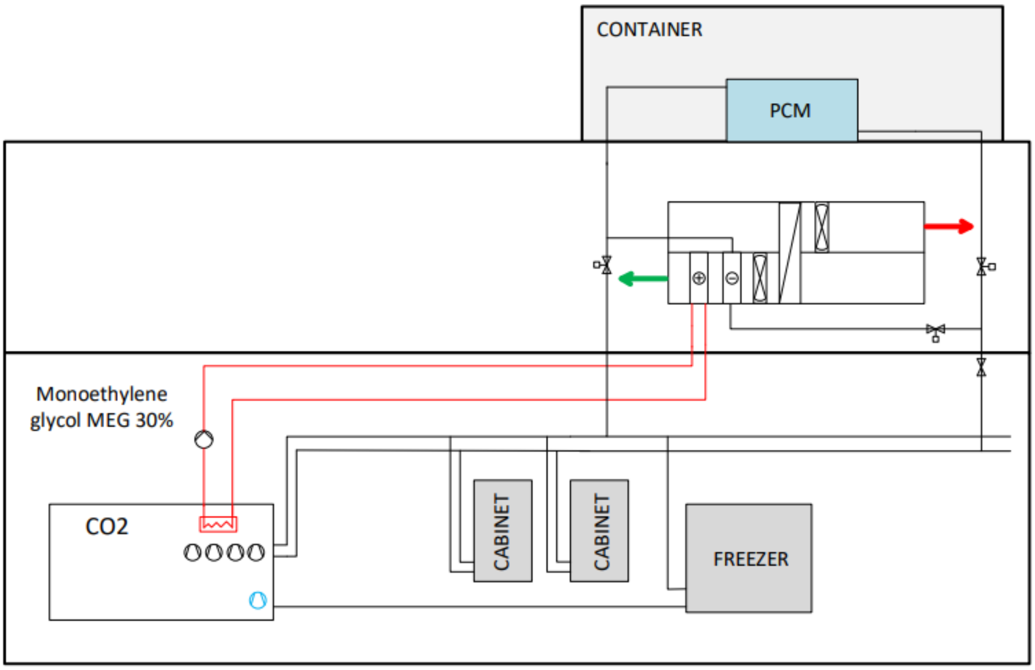


Figure 90: Storage placement

The system will be prefabricated and then placed on the roof. The container will have a coupling point so that it can be connected to the rest of the system inside the building. The container will need some heating, and a radiator is therefore placed inside the container. The minimum size of a container is 20 ft. This size is chosen for the storage solution. Table 17 show the dimensions of the container

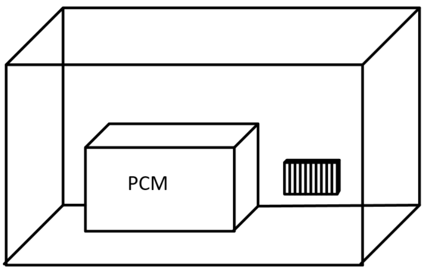


Figure 91: Container with storage

Measurements of container	
External (H x W x L)	2100 x 2100 x 3100 mm
Internal (H x W x L)	1935 x 1975 x 2975 mm
Door opening (H x W)	1890 x 1950 mm

Table 17: Container geometry [82]

PCM

The chosen PCM for the storage is water. Water is available everywhere and is a cheap material. It has good thermal properties with high heat capacity. Further information about water as PCM is given in chapter 2.8.

Storage geometry/construction

The number of pillow plates is adjusted to the calculated cooling load. The container is filled with water as the chosen PCM. The simplified geometry of the PCM storage is shown in Figure 92. The refrigerant enters the pipe in the upper part of the PCM storage tank as indicated in Figure 92. The pipe supplies all the pillow plates with refrigerant. The simplified geometry of the PPs (pillow plates) is shown in Figure 93. The refrigerant enters the PPs in the top as indicated in Figure 93 and exits in the bottom. The pillow plates have a limitation on size of 1.5m x 3m. This is the maximum size that is possible produce today. This will also set limits for the size of the PCM storage.

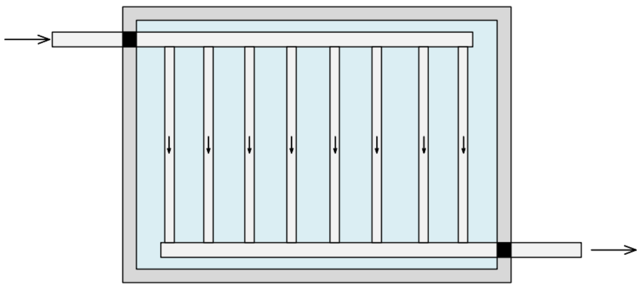


Figure 92: PCM storage

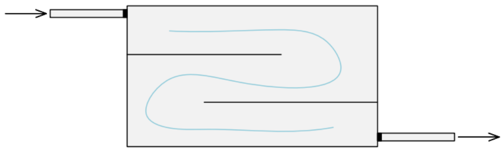


Figure 93: Pillow plate heat exchanger

Figure 94 shows more detailed geometry of the PPs. The wavy structures of the pillow plates will give high heat transfer. Compared to conventional heat exchangers, pillow plates have low manufacturing costs and compact design. [51] When using this type of heat exchanger there is no need for a secondary fluid. The pillow plates can withstand the relatively high operating pressure of CO_2 systems. [45] The refrigerant can both evaporate and condense directly in the storage exchanging heat with the PCM material.

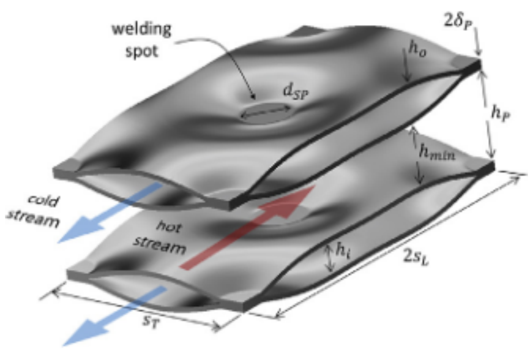


Figure 94:Geometry of a pillow-plate heat exchanger [74]

Figure 95 shows the PCM storage at the NTNU lab from chapter 2.11.5. On the left one can see the entire storage container and on the right one can see the pillow plates. The storage used for our system will have a similar construction. The difference is that the pillow plates will be stacked vertically instead of horizontal. The system at the NTNU lab has horizontal plates, but this system has a pump. The system is going to be self-circulating using gravity and the PPs are therefore placed vertically to help making this possible.

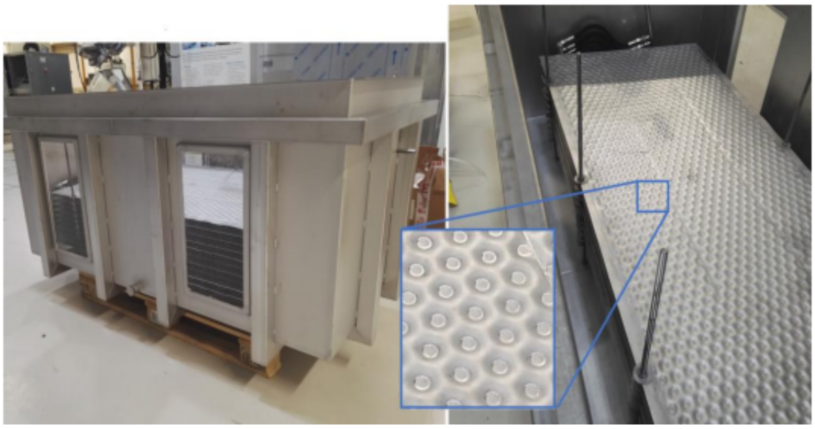


Figure 95: PCM storage at the NTNU lab [70]

Peak shaving and size of the storage

There are different ways to combine storage and chiller. Combining chiller and storage allows for peak shaving of the cooling load. By inserting a thermal storage into the refrigeration system one can correct the mismatch that occurs between supply and demand of energy. [34] Figure 96 shows three different solutions for peak shaving: (a) Full storage, (b) Partial storage with load levelling and (c) Partial storage with demand limit. All three different solutions will be investigated for the new system solution.

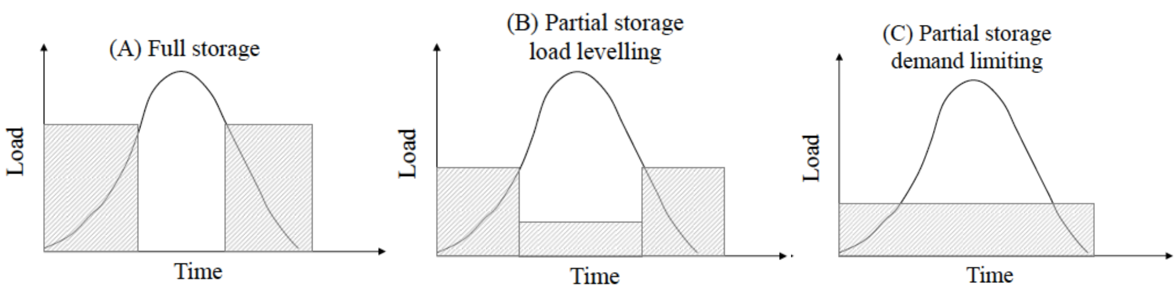


Figure 96: different storage solutions [75]

5.1 Size of storage

The calculations performed on storage size are simplified and a rough approximation. The calculations are done for the "worst case scenario" which is a day in July when the outside temperature is at its highest. There are conducted calculations on different storage solutions. There are made calculations on both full storage and partial storage solutions. The solutions are based on the load curve for July given in Figure 85 in chapter 4.4. The storage is implemented in the excel sheet for the load curves and adjusted until the chosen solutions are obtained. The results are shown in chapter 5.1.1 below.

The calculations on storage size are based on the total volume of the tank being filled with water. The calculations give the size of the ice block. An extra 15% is added to the calculation due to losses in the system. There will be losses in heat exchangers etc. The actual size of the storage will be a little bit bigger than calculated. This is due to extra equipment like heat exchangers, pipes, insulation etc. Table 18 shows data used for calculations of storage capacity and size.

Table 18: Medium characteristics

Medium characteristics	
Medium	water
Temperature[C]	0
*Cp[kJ/kg*K]*	4.182
ρ[kg/m3]	988
*Thermal conductivity [W/m*K]*	2.14

Melting at 0 ℃	
MJ/m3	306
Latent heat of fusion [kJ/kg]	334
ρ (at 0 °C) [kg/m^3]	917
kWh/kJ	2.8E-04
kWh/kg	9.3E-02

The new load curves for the system will give the total energy needed for the storage. Equation 12 is then used to adjust the storage size. Where E is needed energy in kWh, V is volume in m^3, ρ is density in kg/m^{3}and Latent heat of fusion given in kJ/kg. The volume of ice needed can be calculated and then the length, width and height of the storage can be chosen. The geometry of the storage is limited by the maximum length and width of the pillow plates given earlier in Chapter 5. There is made a graph for energy balance for the different solutions. The energy balance is given by equation 13. The energy balance shows the difference between the energy use for the system with and without the storage.

$$E = \frac{V * \rho * Latent\ heat\ of\ fusion}{3600} \tag{12}$$

$$Energy\ balance\ = Energy\ use\ without\ storage - Energy\ use\ with\ storage \tag{13}$$

Compressor combinations

As seen from the compressor combinations in chapter 3.3, the MT load has to be reduced to or below 64kW in order to be able to reduce the system with one compressor. Or including the LT compressor, the total load must be below 68 kW. This has been taken into account when making the new load curves. All solutions will have a maximum load below 68 kW.

It is important to find a load for the system that fits the compressor combinations. Except for the frequency-controlled compressor, the other compressors turn on and off. The new load for the system has to be adapted to fit the possible compressor combinations. Figure 97 shows all the compressors combined. To the left one can see the existing combination of compressors and to the right one can see the combination of compressors without MT2. Table 19 shows the possible compressor steps without MT2. The load curve for the system has to be adapted so that it fits the different compressor steps.

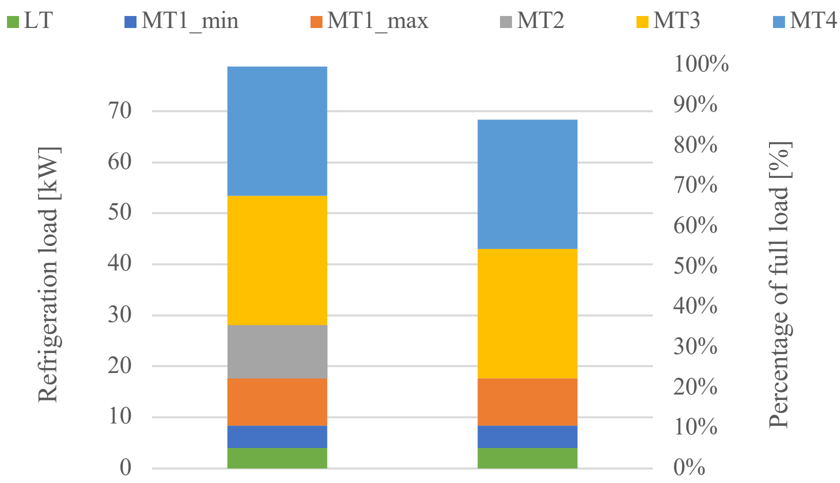


Figure 97: Combination of compressors. Left: Existing system. Right: Existing system without MT2

Table 19: Compressor steps without MT2

Steps	*Interval [kW]*	*Min percentage*	*Max percentage*
LT+MT1	8-18	11 %	22 %
LT+MT3	29	37 %	37 %
LT+MT1+MT3	34-43	43 %	55 %
LT+MT3+MT4	55	69 %	69 %
LT+MT1+MT3+MT4	59-68	75 %	87 %

5.1.1 Results storage size calculations

5.1.1.1 Full storage

The Full storage solution covers 100% of the AC load. The different colours in the graphs represent the different loads on the system: Purple is the LT cabinets; dark blue is the MT cabinets and green indicates the charging of the storage. The black line shows the new load on the system with storage. The red line shows the load without the storage.

Solution 1

For solution 1 there is no production to storage in the hours when the storage is discharging. The production at night is at maximum capacity with three MT compressors running. This is without using the fourth MT compressor. The load at night is 86% of maximum. This is equal to MT1, MT3 and MT4 running. This is the maximum load the system can have. From the energy balance in Figure 99 one can see the energy use relative to the base case without energy storage. The energy use at night is 44 kW higher than for the base case. The energy use in daytime is lower than for the base case. In the middle of the day the energy use is 40 kW lower than the base case. This corresponds to the maximum AC need.

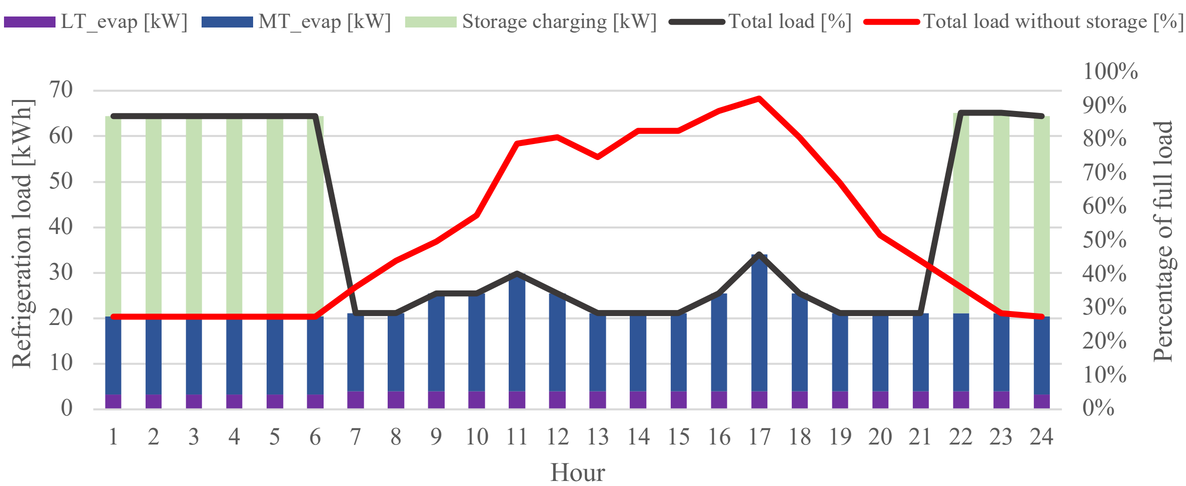


Figure 98: Refrigeration load with full AC storage solution 1

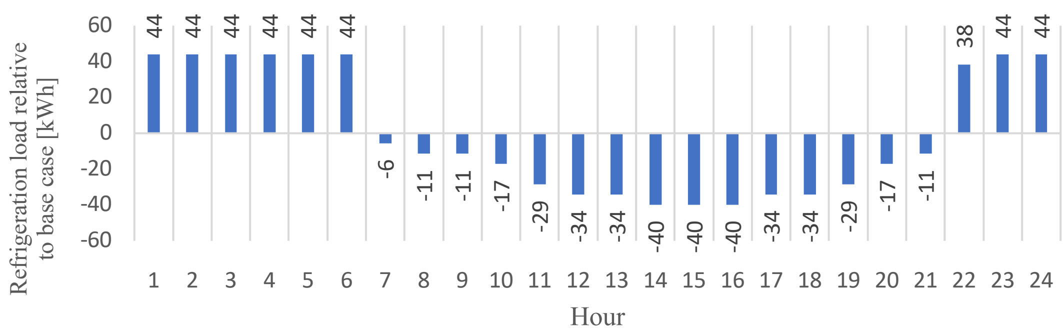


Figure 99: Energy balance solution 1

Solution 2

In solution 2 the system both charges and discharges the storage during the same hour. The numbers are only given per hour. It will be possible to both charge and discharge the storage within the same hour. It is not possible to both charge and discharge at the same time, but one can discharge part of the time and then charge for the rest of the hour. From the energy balance in Figure 101 one can see that the energy use at night is higher than for the base case, but lower than for solution 1. The load at night is 68% of maximum. This is equal to MT3 and MT4 running.

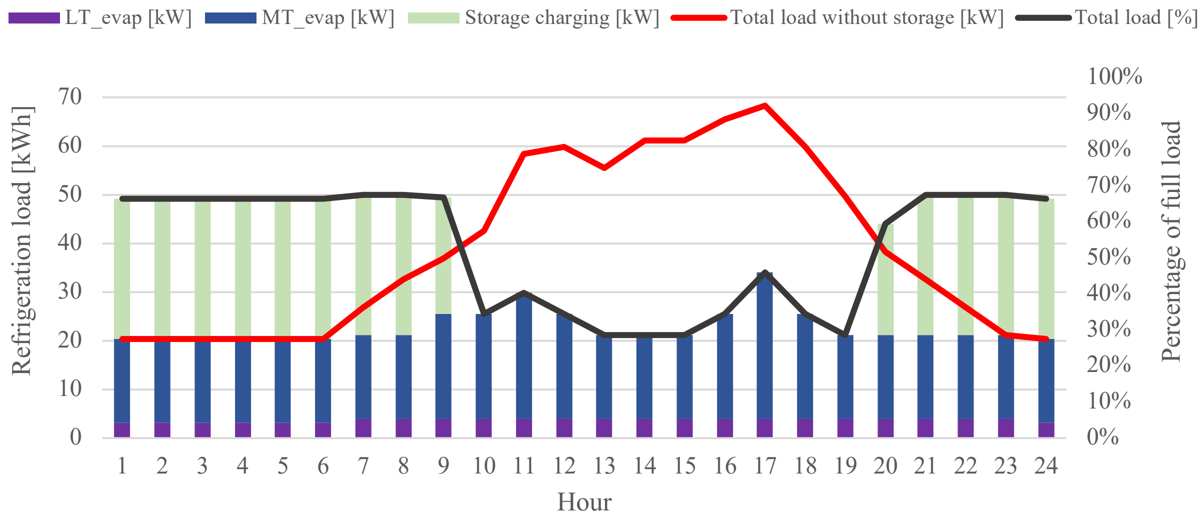


Figure 100: Refrigeration load with full AC storage solution 2

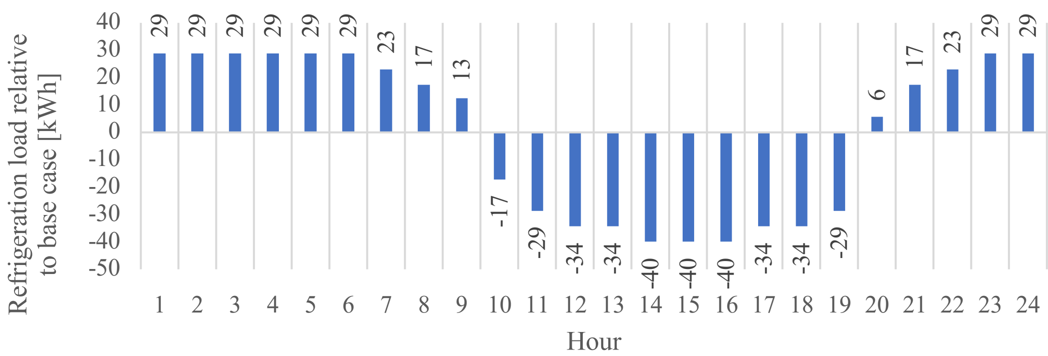


Figure 101: Energy balance solution 2

Solution 3

Solution 3 also has production during the hours of discharging of the storage. From the energy balance in Figure 103 one can see that the energy use at night is higher than for the base case, but lower than for solution 1. The load at night is 77% of maximum. This is equal to MT1, MT3 and MT4 running. The load is then reduced to 45% which equals MT and MT3 running.

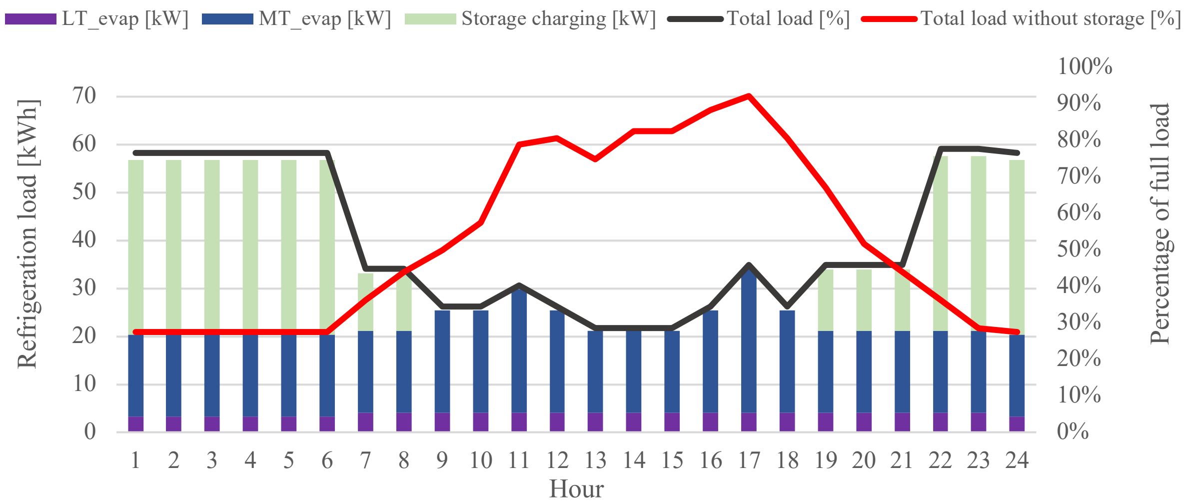


Figure 102: Refrigeration load with full AC storage solution 3

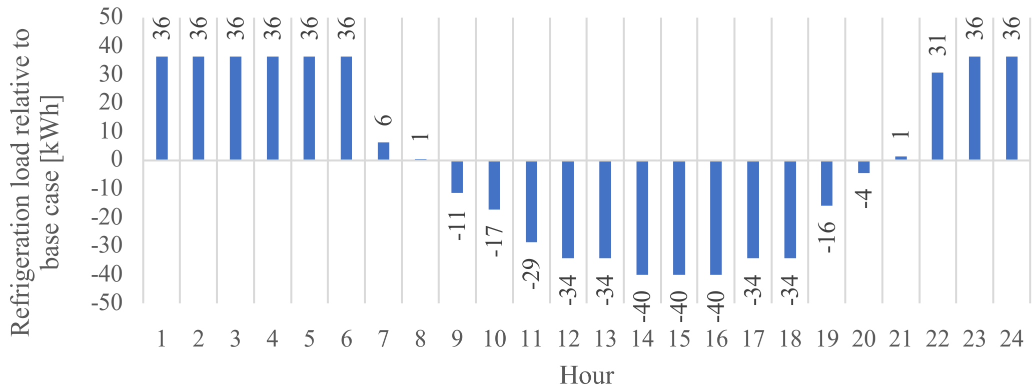


Figure 103: Energy balance solution 3

Storage size with full storage solution

The total AC need for an entire day is 394 kWh. Both solution 1, solution 2 and solution 3 cover the entire AC need. Results for storage capacity and size is presented in Table 20. As one can see, a storage that covers the entire AC load will be heavy. The storage would be 4 264 kg. In addition to this the storage size will be bigger due to the implementation of other components like heat exchangers, pipes and other storage components. Further calculations will be conducted in 5.2 to find the number of pillow plates required to supply the given cooling.

Table 20: Capacity of full PCM storage

Needed AC [kWh]	394
Factor due to losses	1.15
Actually needed storage [kWh]	453
Total amount of ice if nothing else[kg]	4 264
Length[m]	1.5
Width[m]	1.6
Height[m]	2.0

5.1.1.2 Partial storage

Figure 104 shows the load curve for the partial storage solution on a day in July. The different colours in the graphs represent the different loads on the system: Purple is the LT cabinets, dark blue is the MT cabinets, light blue is the AC load from the grid and green indicates the charging of the storage. The black line shows the new load on the system with storage. The red line shows the load without the storage. The partial storage will only cover the load during peak hours and not the entire load. The AC load will be supplied by both the storage and heat pump system.

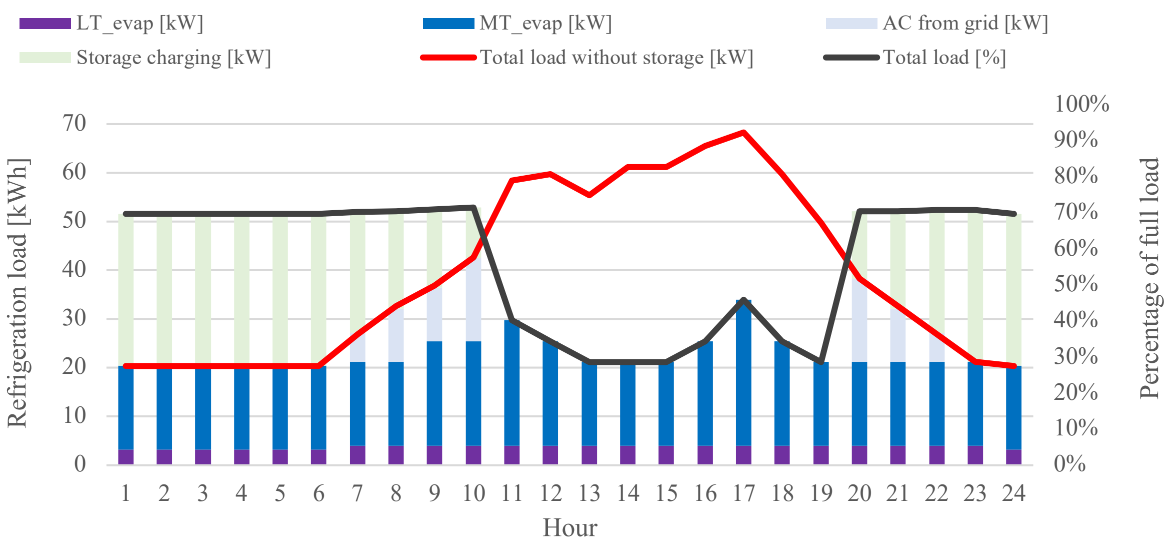


Figure 104: Refrigeration load with partial AC storage

The storage and cooling coil in the AHU are coupled in parallel. As see from Figure 88 it is possible to both charge the storage and supply the cooling coil in the AHU at the same time. The storage will cover the entire load from 11-19. From 7-10 and 20-22 the refrigeration load will be covered by the heat pump system that requires electricity from the grid. During these hours the heat pump system will provide both charging of the storage and cooling for the AHU. This will keep the load at the same level as for nighttime. The load at night is a little bit over 70% of maximum. This is equal to MT1, MT3 and MT4 running. From the energy balance in Figure 101 one can see that the energy use at night is higher than for the base case, but lower than for the full storage solution.

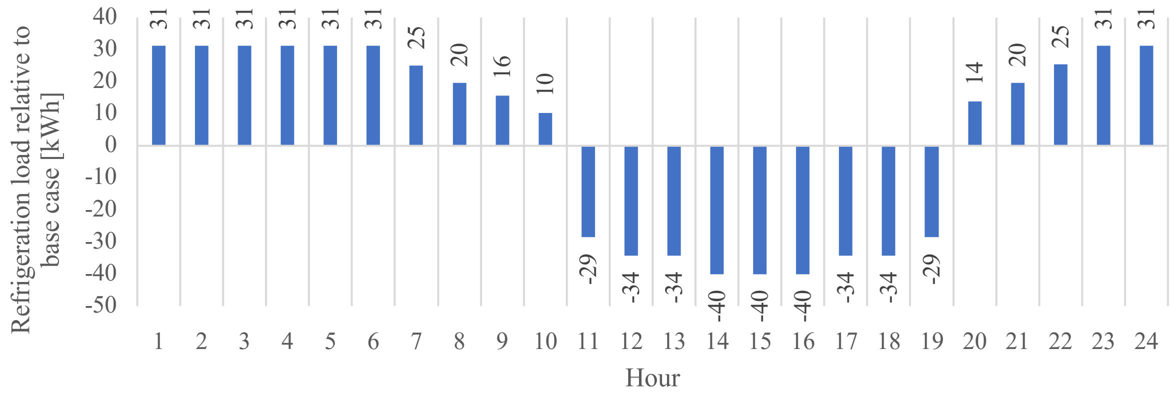


Figure 105: Energy balance [kWh]

The partial storage solution will cover an AC need of 314 kWh. Results for storage capacity and size is presented in Table 21. This storage will be approximately 100 kg lighter than the full storage. The storage would be 4 127kg.

Table 21: Capacity of partial PCM storage

Needed AC [kWh]	314
Factor due to losses	1.15
Actually needed Storage [kWh]	361
Total amount of ice if nothing else[kg]	4 127
Length[m]	1.5
Width[m]	1.5
Height[m]	2.0

5.2 Heat exchanger size

The calculations performed on heat exchanger size are simplified and a rough approximation. Calculations of the number of pillow plates are based on research and experiments performed on the storage at the NTNU lab from chapter 2.11.5. The storage has a similar setup as the storage for this report. It has pillow plates in a tank with water as PCM. The main difference is that the plates in his setup are horizontal, while for this report the plates are vertical. It is assumed that this will not affect the calculation significantly.

Some of the results from the experiments are shown in chapter 2.11.5. Based on the results it is possible to calculate the kW/m^2 for the pillow plates. The average discharge rates for three different mass flows are shown in Figure 55. During discharging the system will be self-circulating. The self-circulation can limit the flow in the system. Therefore the calculations are based on the smallest mass flow from the experiment, which is 4 kg/min.

The pillow plates have a limitation on the size of 1.5m x 3m. This is the maximum size that is possible to produce today. As shown in chapter 2.11.5 the discharge rate of the storage will change during the cycle. The discharging rate will be high in the beginning and then decrease. The discharge rate has to be adapted to fit the load curve of the system. During peak hours the system must be able to supply the maximum AC load which is 40 kW.

During charging the CO_2 circulating through the PPHEs will have the same temperature as the MT evaporators which is -7°C. Research performed by Selvnes et.al [76] [71] shows that this will be a good temperature when using a system like this. The charging time decreases with decreasing temperature. The charging time will also decrease with decreasing distance between the pillow plates.

5.2.1 Results heat exchanger size calculations

Table 22 shows the number of plates needed for different inlet temperatures. The power per m^2 is calculated from the experiments on the storage at the NTNU lab from chapter 2.11.5 [71] [70]. Higher temperatures give higher pressure fall and higher proportion of evaporated CO_2 (gas quality) entering the CTES. There is a big difference in number of pillow plates. The solution with lowest inlet temperature have two times as many pillow plates as the solution with highest temperature.

Table 22: Calculated number of pillow plates needed

	4kg/min and 5,5°C	***4kg/min and 7°C***	***4kg/min and 10°C***
Maximum effect AC [kW]	40	40	40
Power pr m2 [kW/m2]	0.3	0.4	0.6
Area per pillow plate [m^2]	6	6	6
Power pr plate [kW/plate]	1.8	2.4	3.8
Number of plates	22	16	11

Chapter 6. Modelica simulations

This chapter shows the different simulation tools, simulation models and results from the simulations. There is created one model without storage to use as a reference model/base case. This way one can compare the different simulations to the reference model without storage.

6.1 Simulation tools

The system is modelled in Dymola using the Modelica language. The simulation file is then imported to the program DaVE where simulation data is processed and presented in diagrams.

6.1.1 Modelica

Modelica is intended to serve as a standard format so that models arising in different domains can be exchanged between tools and users [77].The physical phenomenon's are modelled with equations. The Modelica language has been developed to allow different tools to generate code automatically. This reduces the effort needed for modelling. [77]

6.1.2 Dymola

The system is modelled in Dymola using the Modelica language. Dymola stands for Dynamic Modelling Laboratory. The system allows for modelling of many different kinds of systems and for the use of modelling libraries. There are many different libraries allowing to reuse components and connectors that already exist. The program architecture is shown in Figure 106. [77]

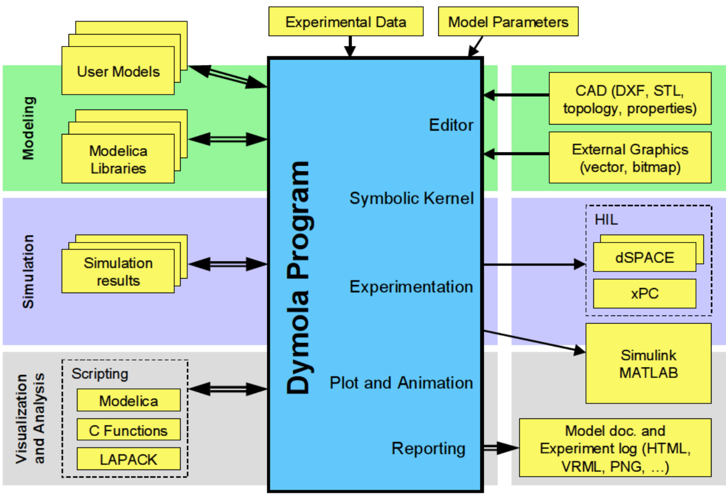


Figure 106: Dymola program architecture [77]

6.1.3 DaVE

Simulations performed in Dymola are presented in graphs using the program DaVE. The simulation file from Dymola is imported to DaVE as a Modelica result file. This makes it possible to use the data to perform calculations and make graphical illustrations. There are made tables for different parameters: MT load, heat recovery (gas cooler load), COP and energy use (electrical input for compressors). The electrical input of the system is given by equation 14. The COP of a system is given by energy output divided by the energy input. The COP of the system is given by equation 15.

$$Electrical\ input = MTComp1 + MTComp2 + MTComp3 + MTComp4 + LTComp1 \qquad (14)$$

$$COP = \frac{Q_{MT} + Q_{LT} + Q_{AC}}{Electrical\ input} \qquad (15)$$

6.2 Simulation model without storage

Refrigeration systems for supermarkets are big and complicated and to be able to perform calculations and simulations a simplified model was needed. The P&ID in Figure 60 is the basis for the simulations and the values in Table 7 are used as inputs. The total simulation model without storage is shown in Figure 107. Further description of the system is presented in the following chapters. For the simulation all unintentional pressure drops are neglected.

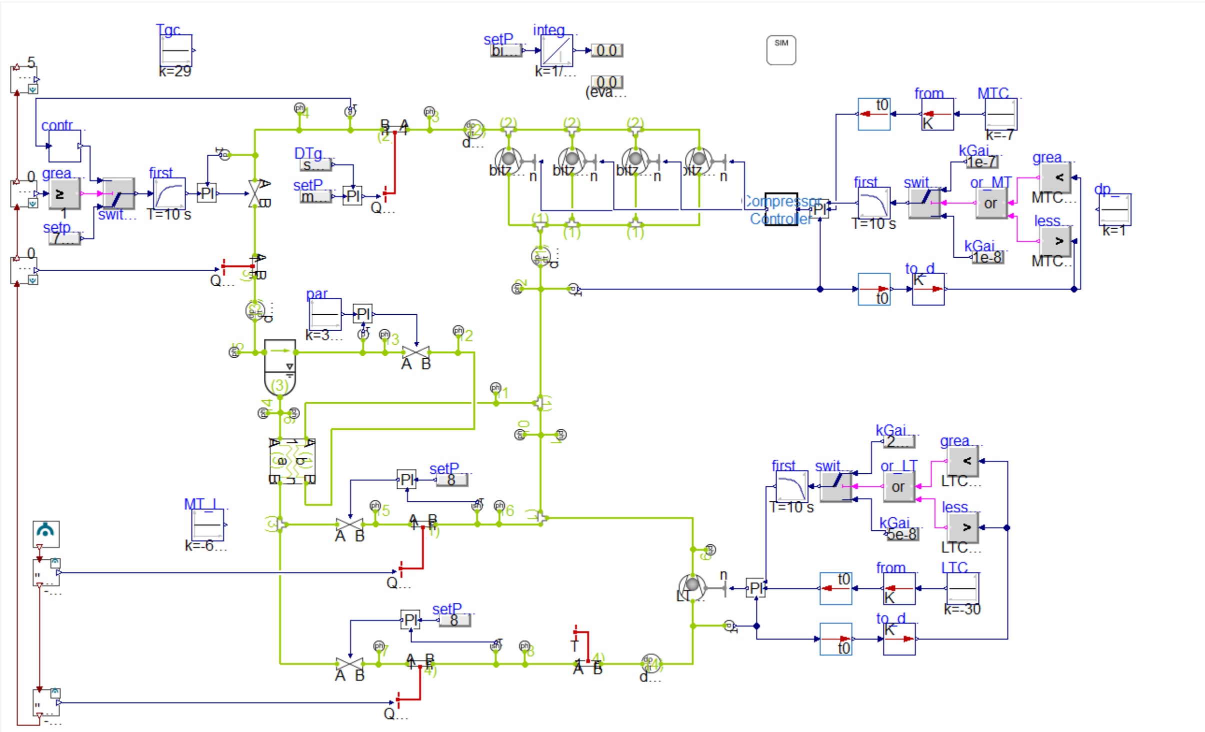


Figure 107: Simulation model in Dymola without storage

Compressors

The compressors that are added in Dymola are the compressors listed in Table 8. There are implemented one compressor at the LT level and four compressors at the MT level. One of the compressors at each level is modelled with speed control of the compressor. In order to adjust the capacity of the system the compressors are modelled with PI control. The controller uses the pressure as the controlling value. If the pressure drops, this indicates that the compressors should reduce the speed. If the pressure increase it indicates that the compressors should increase the speed. Figure 108 shows the LT compressor model and Figure 109 shown the MT compressor model. They are modelled with an isentropic efficiency of 70%

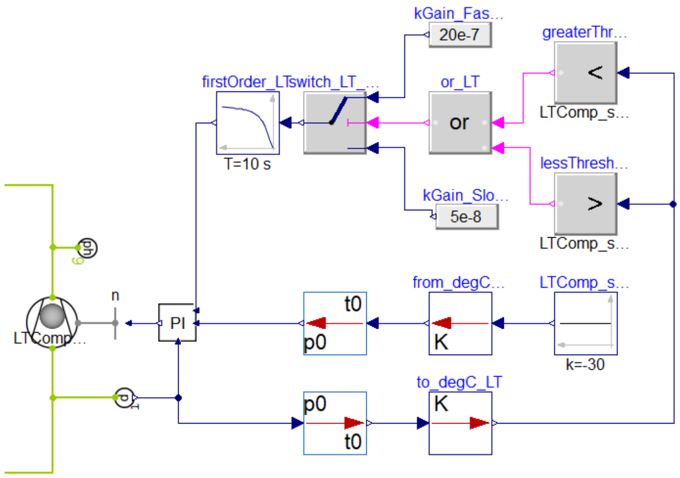


Figure 108: LT compressor model

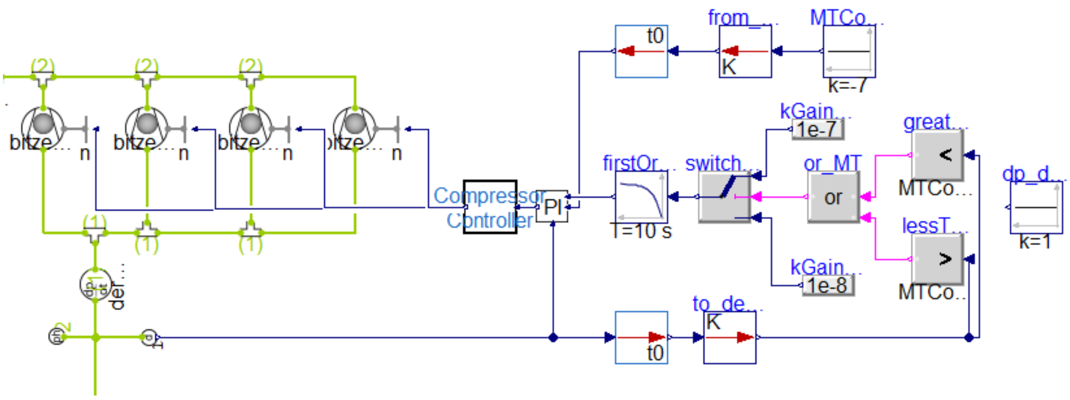


Figure 109: MT compressor model

The compressors are modelled with lights as shown in Figure 110. The lights indicate which of the compressors that are running. In Figure 110 the green dots indicate the compressors that are running, and the grey dots indicates the compressors that are off.

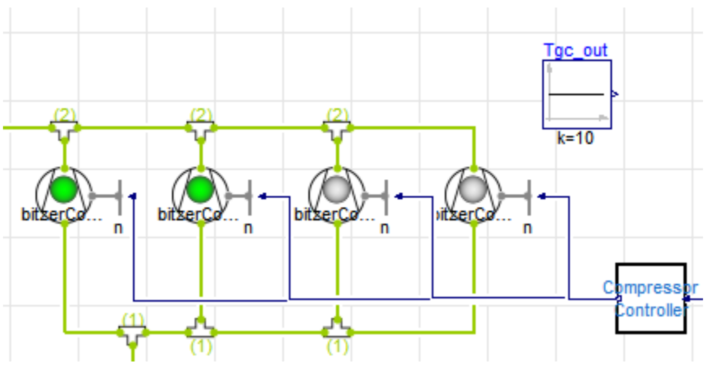


Figure 110: Compressor lights

Input values for the different components

The input values were initially given by pressure (bar). The modelling of the input values is changed, so that the input values can be given in temperatures. The reason for this is that the industry normally gives values in temperatures instead of pressure. This will make it easier to communicate and present data for the people involved in the project. One must convert from Celsius to kelvin and from kelvin to pressure(bar). Figure 111 shows the modelling of conversion. This is done for LT compressors, MT compressors and gas cooler.

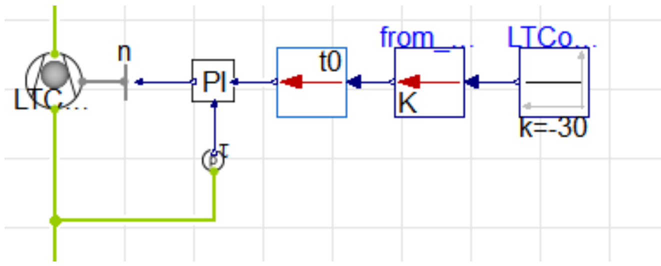


Figure 111: Converting input values from temperature to pressure

TILfilreader

Dynamic data input is implemented using TILfilereader. The input values are saved in an excel file and the excel file is then used as an input for the TILfilrereader. The excel file has the setup shown in Figure 112. The time is listed with the corresponding load. The load is given in kW and the time is given in seconds. Positive values indicate load required from the grid. Negative values indicate energy supplied to the facility (e.g. from the PCM storage that is implemented later in the report). Further description of the implementation of the TILfilereader can be found in chapter 6.4.

	A	B	C	D
1	Time	Load	Flow	
2				
3	0	20		
4	1000	25		
5				

Figure 112: Excel input file to Dymola

Evaporators/Cabinets

The evaporators/cabinets are modelled as shown in Figure 113. The evaporator load is implemented using the TILfilereader. The values are listed in the excel sheet which gives values for the Q_flow. The expansion valve is modelled with a PI control that is regulated by the superheat. A setpoint of 20K is chosen. Defrosting of evaporators is not considered for the simulations.

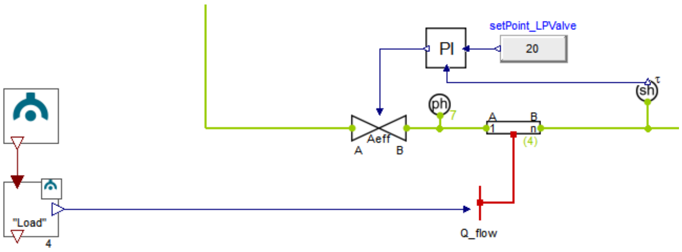


Figure 113: Evaporator model Dymola

Gas cooler and HPV

The gas cooler has inputs for outdoor temperature from the excel sheet. The graphs for the different months are shown in Figure 118. The HPV is modelled with a switch. The valve receives input values using the TILfilereader. The HPV receives a value of either 1 or 0. 1 indicates that there is heating demand (January) and 0 indicates that there is no heating demand.

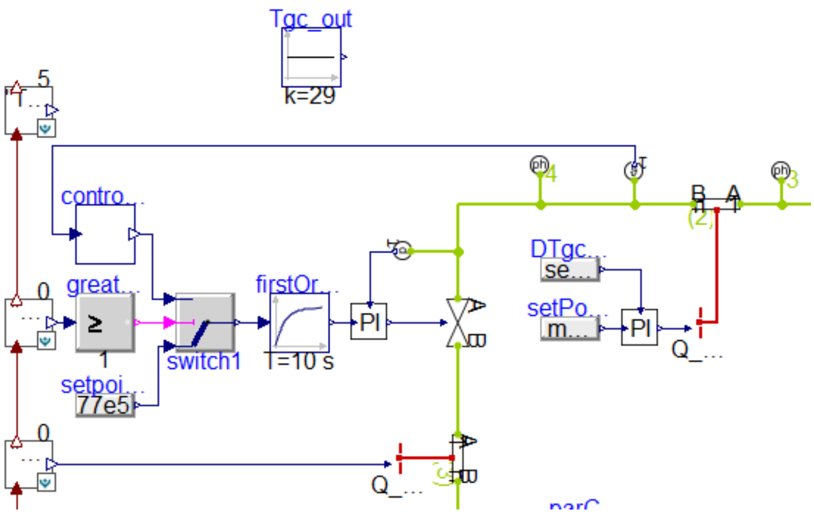


Figure 114: Gascooler2

6.3 Simulation model with AC storage

The simulation model with storage is shown in Figure 115. Except for the implementation of the CTES the simulation model is the same as in chapter 6.2. There is not made a separate heat exchanger for the storage in the system. The storage is supplied by the same loop as the MT cabinets as shown in Figure 88. The AC load is therefore implemented in the input values for the MT load. In periods where the storage is charging, the MT load on the system is increased. The simulations are performed for full storage not partial storage.

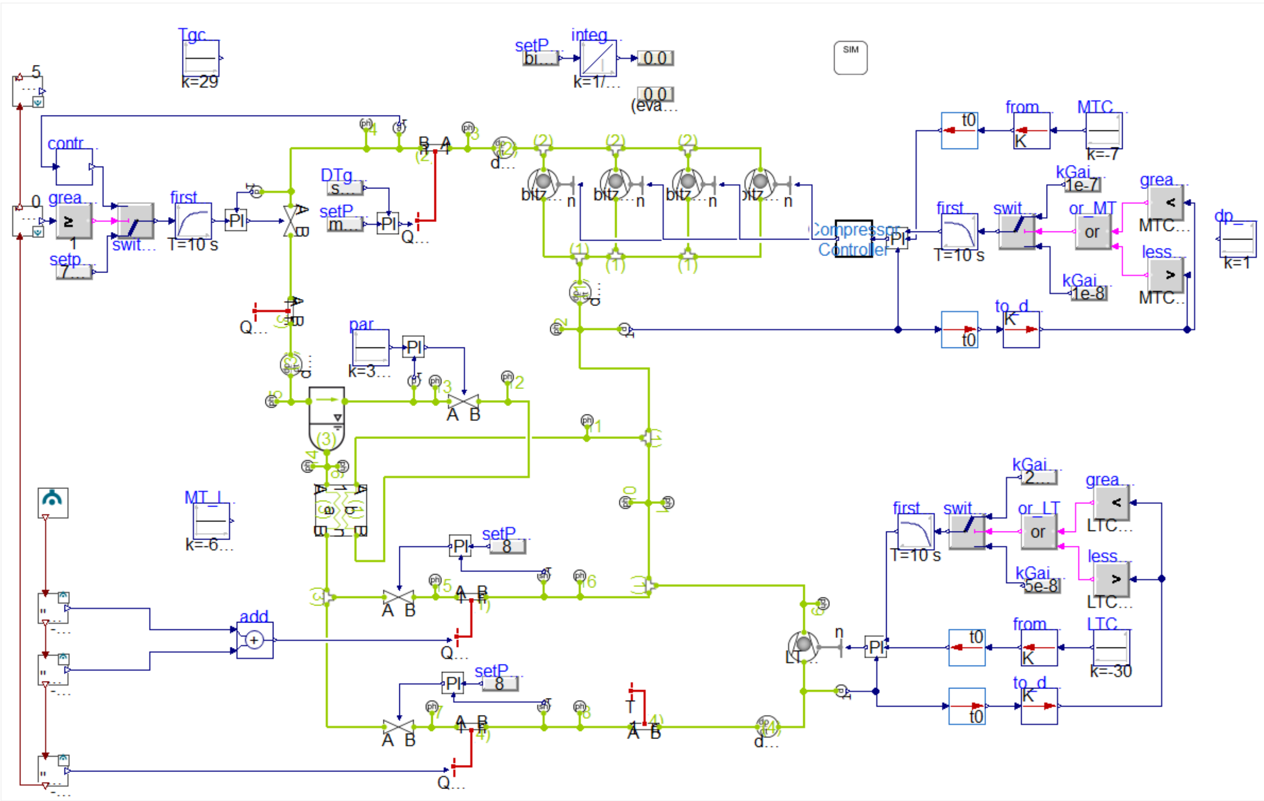


Figure 115: Simulation model in Dymola with storage

MT load full storage

The MT load is modelled as shown in Figure 116. It is modelled using a Modelica block that adds together two loads. The load is modelled with equation 16. The two loads that are added together are the MT cabinet load and the charging of the storage. For the full storage solution there is no AC load required from the grid, all AC load is supplied by the storage.

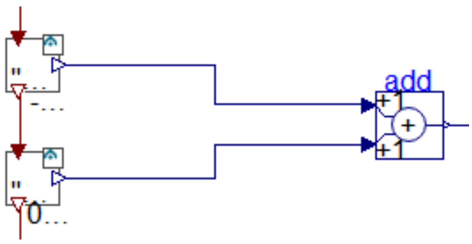


Figure 116: Modelling MT load with full storage

$$Q_{MT} = Q_{MT_{evap}} + Q_{Storage_{charge}} \tag{16}$$

6.4 Input data for the Til filereader

There is created one excel file used for the entire simulation. All input data is collected in this excel sheet. The load curves used for simulations are shown in chapter 4.4. In the excel sheet the load curves are given in percentage of full load. The load curve in percentage will be the same for all three months, but the maximum load is changed to fit the given month. The maximum load for the different months is given in Table 23. This makes simulation easier. The load curve can be the same, but the maximum load can be changed to fit the different months.

Table 23: Maximum load for different months

Month	*January*	*April*	*July*
MT max [kW]	20	25	30
LT max [kW]	3	3.5	4
AC max [kW]	0	0	40

Figure 117 shows an example of input for the MT load. The “variable name” refers to which column the data should be collected from. In this case the column is called “CapMT”. The value given in the box called “factor” gives the maximum load that should be multiplied with the data from the given column in the excel sheet. In this case the maximum load for MT is 30kW. The number is given in Watts (30e3W). The negative sign indicates that energy is required from the system. A positive sign would indicate that energy is supplied to the system. An example would be cooling supplied from the energy storage.

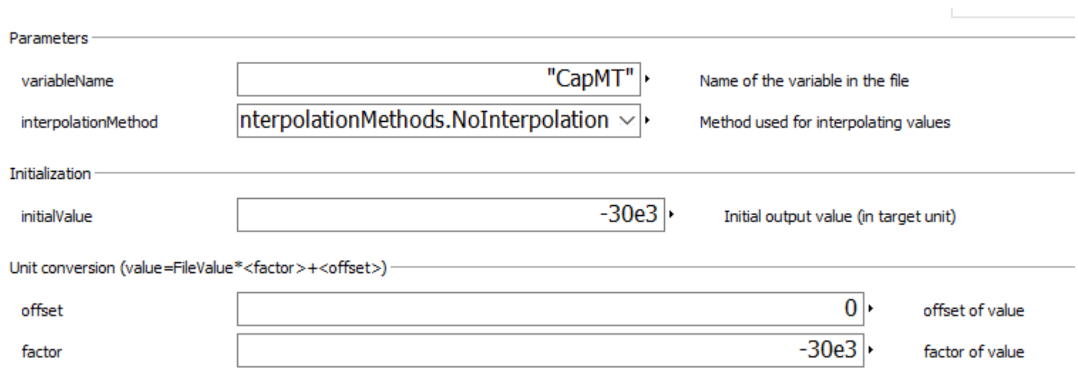


Figure 117: TilfileReader

Outside temperature

Figure 118 shows the outside temperatures for the different months used in the simulations. This is used as an input for the gas cooler. The different colours show temperatures for the different months. Grey is July, orange is April and blue is January.

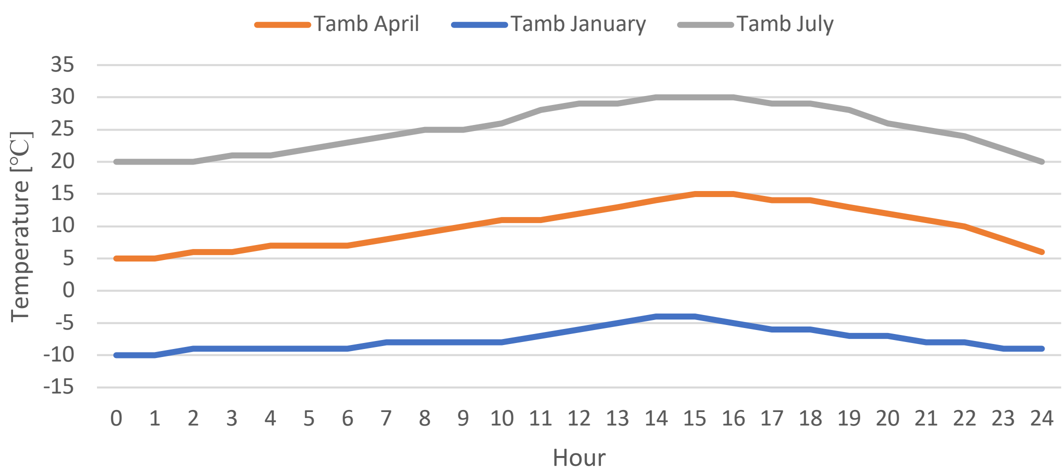


Figure 118:Outside temperatures for the simulation months

6.5 Results simulations

This chapter shows simulation results both with and without storage.

6.5.1 Simulation with excel file

Data are presented in graphs to be able to compare the results There are made four different simulation files: January, April, July without storage and July with storage. The simulation is performed with the given load curves in chapter 4.4 and a gas cooler temperature that varies with the outdoor temperature. The excel file used for the simulation is shown in appendix C.

MT load

From Figure 119 one can see the MT load simulated for the different months (load curves) given in chapter 4.4. The curves have the same profile in January, April and July (without storage), but with different maximum load. For the model without storage the AC load is implemented before the receiver and is therefore not part of the MT load. For the simulation with storage the profile is changed. For the model with storage the AC load is implemented in the MT load. The storage is implemented to reduce the maximum load during peak hours. The MT load is increased outside peak hours.

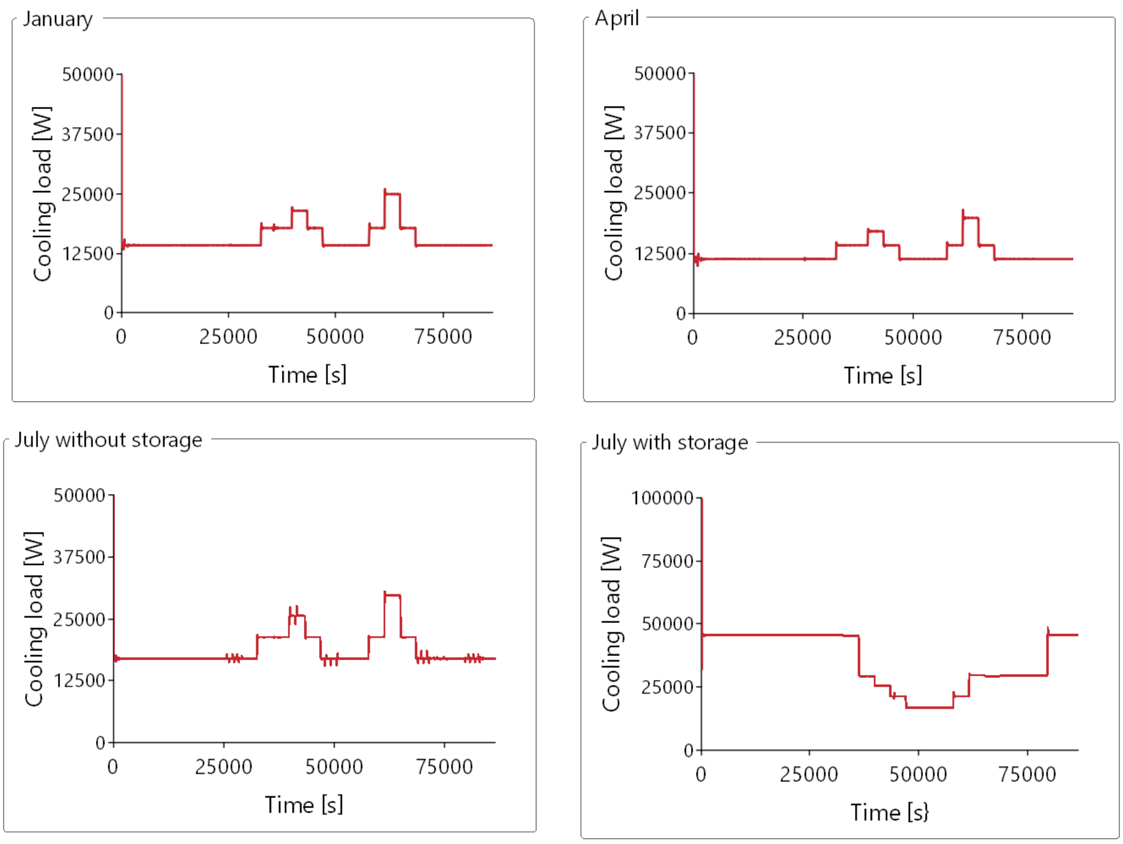


Figure 119: MT load. From upper left: January, April, July without storage, July with storage

Gas cooler

Figure 120 shows the gas cooler load. January and April show similar load curves. July (without storage) has a higher load and shows a bigger potential for heat recovery from the gas cooler. The possible heat that can be recovered increase with decreasing need for heat. In the winter the heat can be recovered and used for snow melting or other operations that require heat. For July with storage the gas cooler load is higher in the night. This is when the refrigeration load on the system is highest.

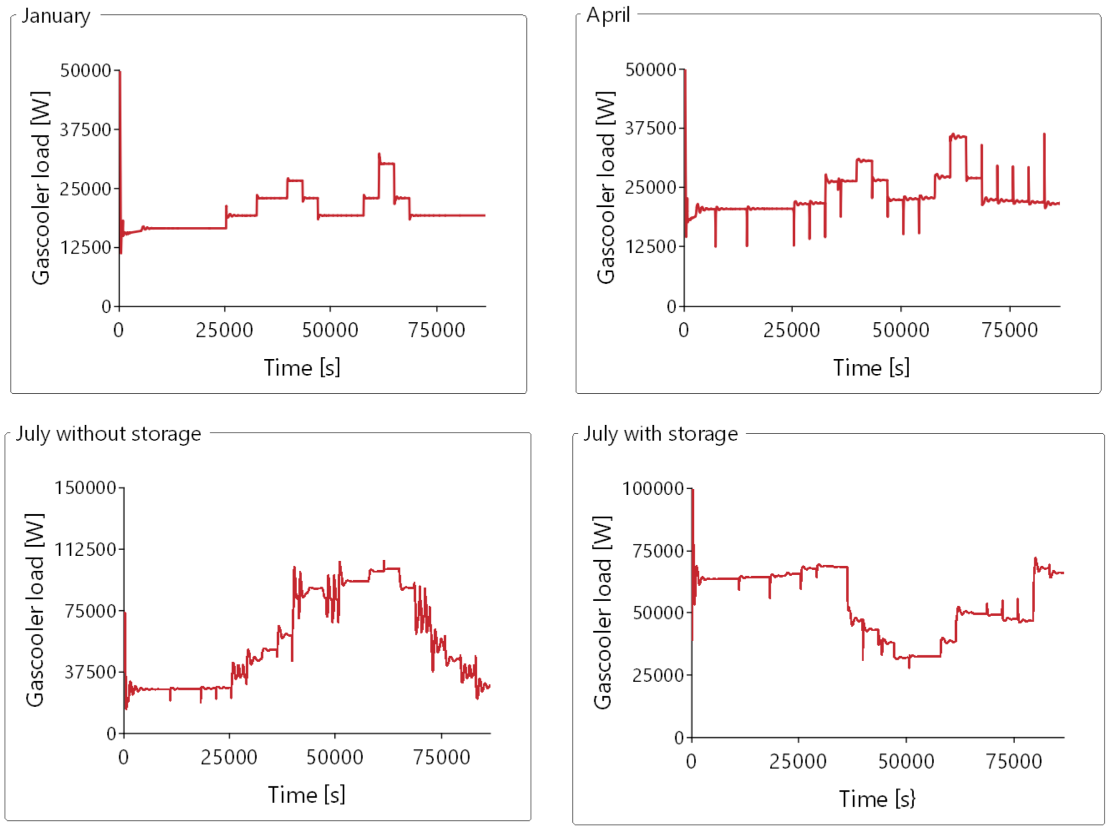


Figure 120: Gas cooler load. From upper left: January, April, July without storage, July with storage

Compressor

Figure 121 shows compressor on/off and Figure 122 shows the energy input for the compressors. Both figures show simulation for one day (24 hours). In January there are only one compressor that are running at all times MT1. MT2 turns on during the peak hours at the middle of the day. In April there is still only one compressor that runs continuously throughout the day MT1. The second compressor MT2 turns on and off more frequently than in January.

In July without storage the compressors turn on and off more frequently. MT1 and MT2 are running continuously throughout the day, while MT3 and MT 4 turns on and off during the peak hours in the middle of the day. For the simulation with storage the last compressor MT4 is not running at all. In addition to this the on and off cycles are reduced, and the system runs more stable than without storage.

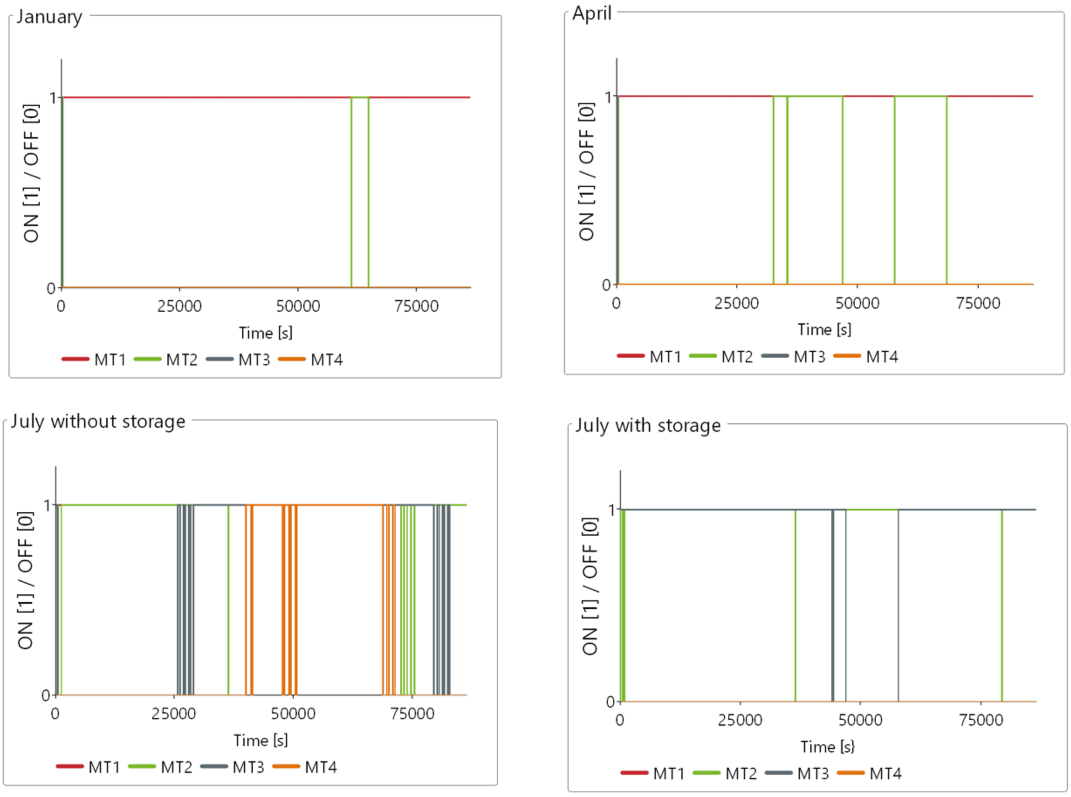


Figure 121: Compressors On/Off. From upper left: January, April, July without storage, July with storage

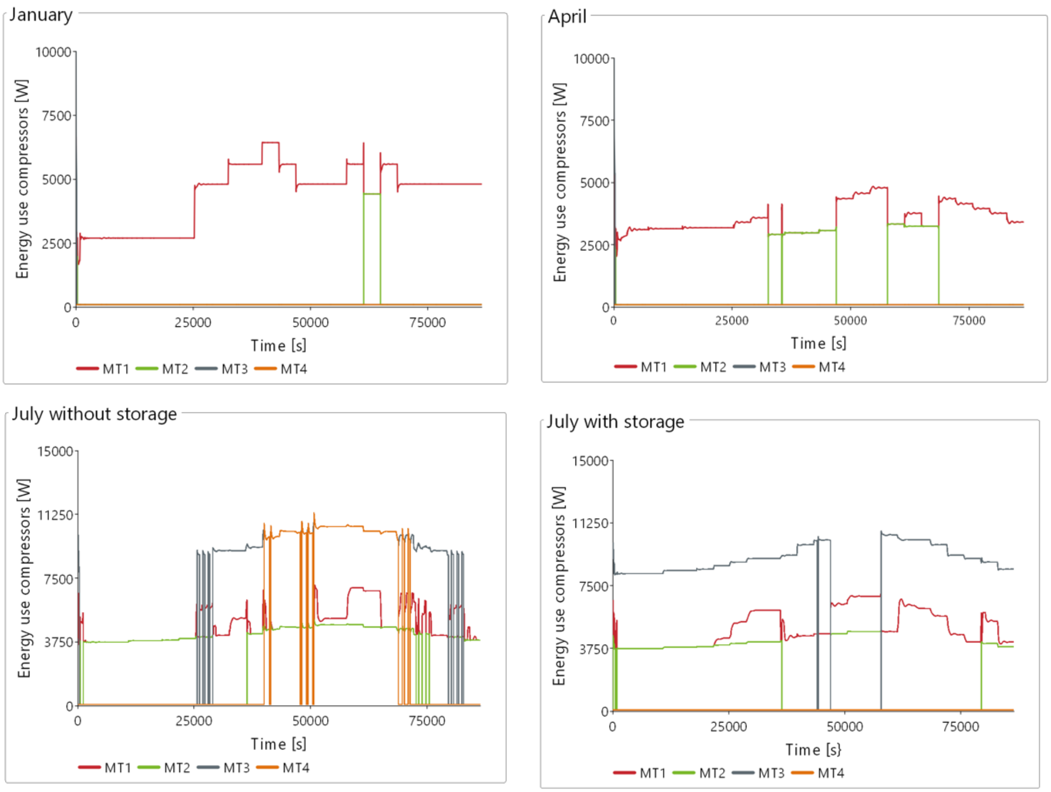


Figure 122: Energy use for compressors. From upper left: January, April, July without storage, July with storage

Energy use and COP

Figure 123 shows the energy use for the different months and Figure 124 shows the COP. Red is January, Green in April, grey is July without storage and orange is July with storage. As one can see the energy use for January and April will be much lower than in July. January and April have a very linear graph. These months does not have AC need. As one can see there will be difference in energy use for July with storage and without storage. The simulation without storage has a steeper slope in the middle of the day, which is the peak hours. This shows that the energy use is higher during the peak hours. For the simulation with storage the energy use has a more linear graph, indicating that the energy use is more even during the day (constant slope number).

The total energy use with and without storage will also be different. The system with storage will produce energy at night with more favourable conditions, hence a better COP. The required energy input will therefore be lower. Figure 124 shows the COP for the different months. January and April have the highest COP. July with storage has higher COP during most parts of the day compared to July without storage.

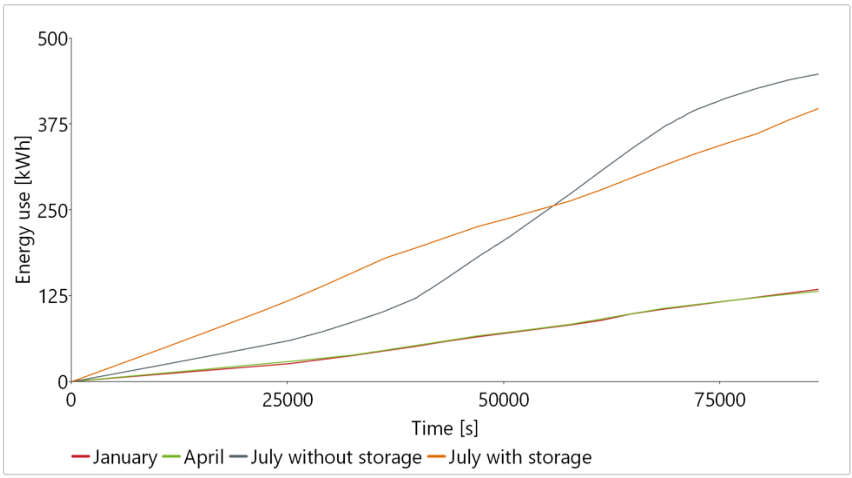


Figure 123: Energy use for the different months

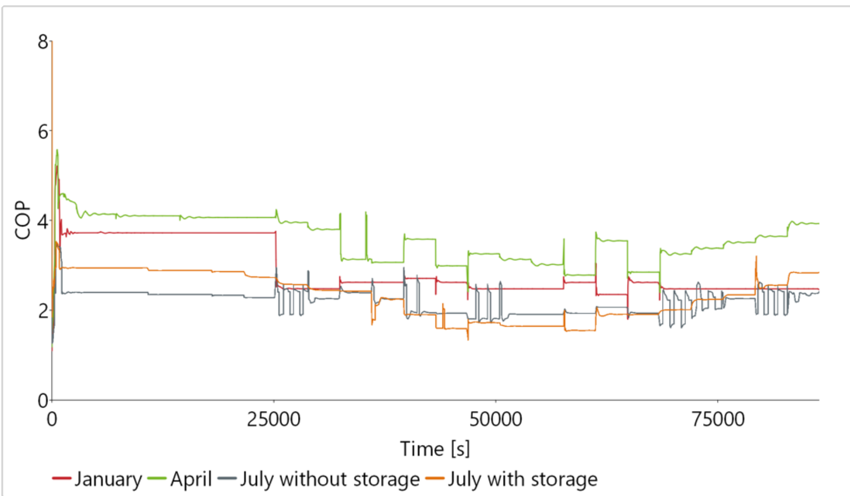


Figure 124: COP for the different months

Summary of results

Table 24 shows a summary of the simulation results. The different compressors are listed. A value of 1 indicates that the compressor is running and a value of 0 indicates that the compressor is off. A value of 1 does not mean that the compressor is running continuously but indicate that the compressor turns on at least one time during the simulation. As one can see the difference in energy use between July with and without storage is 50 kWh.

Table 24: Summary of simulation results

	January	*April*	*July*	*July with storage*
Total load [kWh]	135	132	448	398
MT1	1	1	1	1
MT2	1	1	1	1
MT3	0	0	1	1
MT4	0	0	1	0
LT	1	1	1	1
MT load max [kW]	20	25	30	30
LT load max [kW]	3	3.5	4	4
AC load max [kW]	0	0	40	40
Heating	Yes	No	No	No

6.5.2 Simulation for different gas cooler temperatures

Table 25-28 shows results for simulations with different gas cooler temperatures. They show the compressors that are running with different temperatures out of the gas cooler. The simulation uses the load curves in chapter 4.4. The gas cooler temperature is changed for the different simulations. Value of 1 indicates that the compressor is running. A value of 0 indicates that the compressor is off.

From the graphs one can see that for January compressors MT3 and MT4 is off for all the simulations. For April MT4 is off for all simulations and MT3 turns on when the gas cooler outlet temperature reaches 23 degrees. For the simulations for July without storage MT4 turns on when the gas cooler outlet temperature reaches 19 degrees. For July with storage the compressor MT4 never turns on. The load for the system is reduced significantly and the compressor therefor never turns on.

Table 25: Compressors that are running January

Tgc_out [°C]	*LT*	*MT1*	*MT2*	*MT3*	*MT4*
11	1	1	1	0	0
13	1	1	1	0	0
15	1	1	1	0	0
17	1	1	1	0	0
19	1	1	1	0	0
21	1	1	1	0	0
23	1	1	1	0	0
25	1	1	1	0	0
27	1	1	1	0	0
29	1	1	1	0	0

Table 26: Compressors that are running April

Tgc_out [°C]	*LT*	*MT1*	*MT2*	*MT3*	*MT4*
11	1	1	1	0	0
13	1	1	1	0	0
15	1	1	1	0	0
17	1	1	1	0	0
19	1	1	1	0	0
21	1	1	1	0	0
23	1	1	1	1	0
25	1	1	1	1	0
27	1	1	1	1	0
29	1	1	1	1	0

Table 27: Compressors that are running July without storage

Tgc_out [°C]	LT	MT1	MT2	MT3	MT4
11	1	1	1	1	0
13	1	1	1	1	0
15	1	1	1	1	0
17	1	1	1	1	0
19	1	1	1	1	1
21	1	1	1	1	1
23	1	1	1	1	1
25	1	1	1	1	1
27	1	1	1	1	1
29	1	1	1	1	1

Table 28: Compressors that are running July with storage

Tgc_out [°C]	LT	MT1	MT2	MT3	MT4
11	1	1	1	1	0
13	1	1	1	1	0
15	1	1	1	1	0
17	1	1	1	1	0
19	1	1	1	1	0
21	1	1	1	1	0
23	1	1	1	1	0
25	1	1	1	1	0
27	1	1	1	1	0
29	1	1	1	1	0

Chapter 7. Investment Analysis

This analysis looks at both price for the storage system and also reduced price for the total system. For the optimal system the price of the storage would be less than the savings for the rest of the system. The requirement for payback time is three years. Prices from the contractor that will be building the storage in the lab has been requested. The results from the analysis were sent to contractor earlier in the semester, but response was unfortunately not received before delivery.

7.1 Investment new system

The main components needed for the storage is: PCM, storage tank, container, heat exchanger, valves and piping. In addition to all these components the storage also needs control system and extra measurement equipment. Except for main components there will also be cost for other things like reinforcement of the roof, installation etc.

PCM: Water is cheap. Compared to other mediums one can say that it is free. One can open the tap in the sink and fill the storage. Using water as PCM will therefore not lead to any major costs for the system.

Storage tank: The main costs of the storage is the storage tank and heat exchanger equipment. The storage tank will use steel as main material. Several different heat exchanger types can be used for the system. Pillow plate heat exchangers are chosen for this storage system. The container and the storage will be prefabricated and will be placed on the roof as explained in Chapter 5.

Piping and valves: During charging of the storage the refrigerant will pass through the CTES heat exchanger. During discharging the system will be self-circulating. There will be need for some extra piping and a few valves. But there will not be any major cost related to extra piping and valves. Since the storage is coupled in parallel with the cabinets, some of the piping will be the same as for the existing system. The extra piping is the piping from the branching and to the storage. An approximation of required pipe length is the distance from the branching and to the storage multiplied with two. This will give 16 meters of pipes. Number of valves is set to 4

Control system: The storage will need a control system and measuring equipment and sensors. The sensors need to be able to detect when the storage need charging and when there is need for discharging. The control system will not be looked at when calculating price for the system.

It is challenging to calculate costs of the system. Prices will vary a lot depending on the number of components. If one were to order 1 or 100 heat exchangers the price will be different. Large quantity will give lower prices. It is also dependent on whether the contractor that is going to build the system is experienced on PCM storages or not. An experienced contractor will use less hours than an inexperienced one. The first time the storage system is created the cost of engineering hours will be high. However, when the first storage system is built the cost of the other systems will be lower. The system will have a standard solution. For the analysis it is assumed that the contractor is experienced and that this is not the first system to be built. It is also assumed that the developer buys many systems not just one. Table 29 shows cost of components and total cost of the system. The total cost of the storage system is 229 400 NOK.

Table 29: Total cost of the storage system

	Total cost [NOK]
Piping	6 400
Valves	8 000
Steel tank	50 000
Heat exchanger	30 000
PCM	0
Container	15 000
Installation of piping	50 000
Increasing stability of roof	20 000
Engineering	50 000
Total price of storage	229 400

7.2 Reduced investment costs existing system

Inserting a thermal storage will not only give extra cost for the thermal storage. It can also lead to reduced costs of the total system. Smaller pipes and components will reduce the cost of the system. It can also be possible to reduce the system by one compressor. Figure 86 in Chapter 5 shows the components that are removed from the existing system when the storage is implemented. Table 30 shows the cost of components that can be removed from the existing system when the storage is implemented and the total reduced costs. The reduced costs are 321 900 NOK. The CO_2 pipes are much smaller than the glycol pipes. The cost of installation for the glycol pipes are therefor higher.

Table 30: Total reduced costs

	Total cost [NOK]
Monoethylene glycol MEG 30%	900
Piping glycol circuit	3 000
Heat exchanger AC	5 000
Three-way valve	2 000
Pump	31 000
Compressor	70 000
Reduced size of gas cooler	10 000
Reduced use of space in machine room	
Installation of piping	200 000
Total	321 900

7.3 Reduced operating cost

In addition to the reduced size of the initial system, the new system solution will have reduced operating cost. Implementation of the storage will lead to reduced energy consumption due to production at favourable conditions. Producing energy at night requires less energy than procuring the same amount of energy at daytime. Therefor the energy need for the system will be reduced. This will reduce operation costs. The energy price is also lower at night. Equation 17 is used to calculate energy savings for the system. Where $E_{basecase}$ is energy use without storage and E_{CTES} is energy use with storage. The price of energy is set to 0.98NOK/kWh, the energy need for the base case is 448kWh and energy need for new system is 398kWh. The energy savings per day is 50 kWh corresponding to 49 NOK/day.

$$E_{saved} = E_{base\ case} - E_{CTES} \tag{17}$$

7.4 Payback

The payback method is the easiest way to evaluate if an investment is profitable. The method does not take into account the time value of money. The payback value is given in years. Since the calculations are rough approximation it should only be used in early phase calculations. The required payback time for this project is 3 years. Equation 18 shows the formula for payback time.

$$PB = \frac{Initial\ investment}{cashflow\ per\ year} \tag{18}$$

Assuming that the total savings for the system would be 321 900 NOK and the total cost of the storage is 229 400 NOK. The payback time will be zero. However, one has to take into account that these calculations are very simplified and further calculations will be needed. The system will also give reduced operating costs. The storage operates at night which gives better COP. Thereby the solution will be profitable based on the calculation.

Chapter 8. Further work

Further development of the system

This chapter show further developments that has been done to the system. These upgrades were not a part of the simulation model. These upgrades should be further investigated. There is created a system where the storage supplies both the AC unit and the MT evaporators. Figure 125 shows the simplified system. One or several of the MT cabinets can be connected to the PCM loop. The solution does not require a lot of extra equipment.

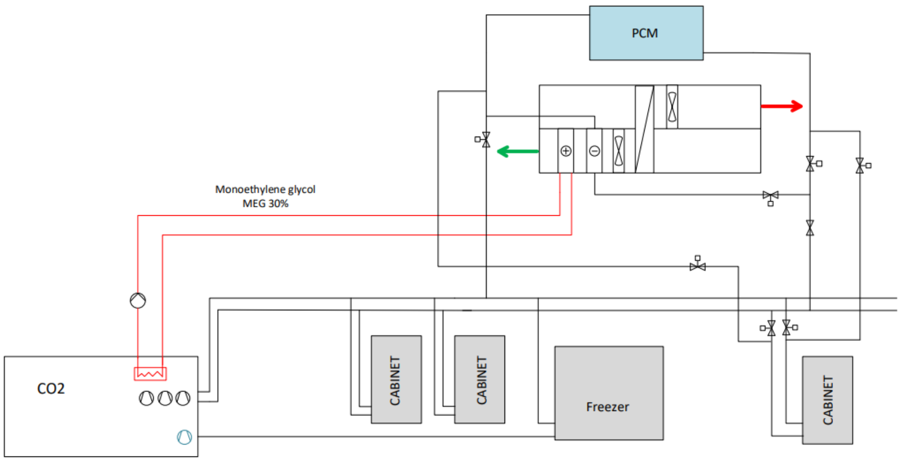


Figure 125: New simplified system solution with PCM storage for both AC and MT

Figure 126 shown the load curve for the system. As one can see this solution gives good possibilities of load levelling. The two tops in the MT evaporator load can be flattened out. One cabinet is approximately 3.5kW. One can decide how many cabinets that should be connected to the PCM loop. The number of cabinets can be decided so that the MT load becomes even. In theory one will then only need two different compressor levels. One in the night and one during the day.

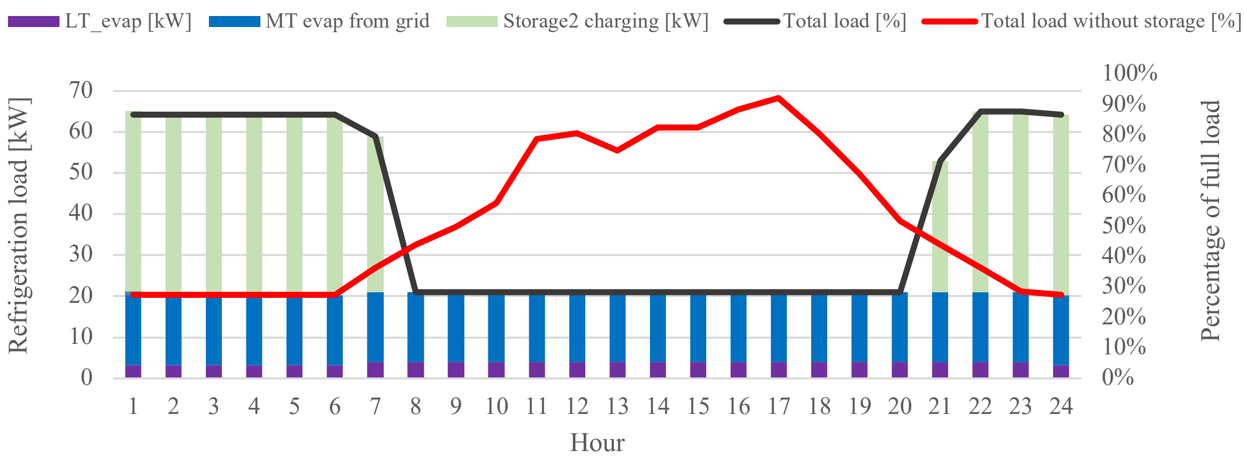


Figure 126: Full storage for both AC and MT load

Experiments in the LAB

Further work will be to perform experiments on the storage that will be built in the NTNU lab. Investigation of the charging and discharging cycles will be important to see how the storage operates in real life. It is important to test how the self-circulation works in real life. Experiments can be performed with different numbers of pillow plates, different plate distances, different mass flows and different inlet temperature for charging and discharging.

Cost analysis

Prices from the contractor that will be building the storage in the lab has been requested. The results from the analysis were sent to contractor earlier in the semester, but response was unfortunately not received before delivery. The cost of the system has to be adapted when data from the contractor is received. It is assumed that they might add some components and adjust some of the prices.

Chapter 9. Discussion and summary

This chapter will give a discussion of the topics presented in the report. The discussion is divided into subchapters.

9.1 Modelica simulations

Setting up an accurate model is challenging. Simplification of the system has to be made in order to be able to perform simulations. From the simulations one can clearly see differences between the simulations for July with and without storage. For the model without storage the compressors turn on and off more frequently. MT1 and MT2 are running continuously throughout the day, while MT3 and MT4 turns on and off during the peak hours in the middle of the day. For the simulation with storage the last compressor MT4 is not running at all. In addition to this the on and off cycles are reduced, and the system runs more stable than without storage.

As one can see from Figure 123 the energy use for January and April will be much lower than in July. As one can see there will be difference in energy use for July with storage and without storage. The simulation without storage has a steeper slope in the middle of the day, which is the peak hours. This shows that the energy use is higher during the peak hours. For the simulation with storage the energy use has a more linear graph, indicating that the energy use is more even during the day (constant slope number). The energy use is a little lower in the middle oof the day. The total energy use with and without storage will also be different. The system with storage will produce energy at night with more favourable conditions, hence a better COP. The required energy input will therefore be lower. July with storage has higher COP during most parts of the day compared to July without storage.

As one can see from the simulation with different gas cooler temperatures there is a clear difference for the simulation with and without storage. For the simulations for July without storage MT4 turns on when the gas cooler outlet temperature reaches 19 degrees. For July with storage the compressor MT4 never turns on. The load for the system is reduced significantly and the compressor therefor never turns on.

9.2 Compressor layout and regulation

It is important for an efficient system to have a good compressor combination and capacity regulation. When choosing compressor combinations, it is important to choose a configuration where the different steps "overlap" or starts/begins at the same point. In order to have an efficient system the maximum value of one step should be close to the minimum value of the next step.

The system has several compressors. The frequency-controlled compressor adjusts the load by turning up and down the speed. When the speed is at max the next compressor turns on and the speed is reduced to minimum. The cooling load varies during the day and compressors are turned on and off many times. The speed of frequency-controlled compressor is also constantly changing. By integrating a storage into the system, the system can be run at a more constant load. This way one will avoid frequent on and off for the compressor and changes in speed and compressor combinations. This will lead to increased lifetime of the compressors.

When implementing a storage one can reduce the number of on/off cycles on the compressor. The load can be kept at a specified load for a longer time. Instead of turning off the compressor when the need for cooling is reduced, the system can charge the storage. Similarly, when the need for cooling increases one can avoid turning on more compressors by discharging the storage. The new load for the system with storage has to be adapted to fit the possible compressor combinations.

9.3 Implementation of storage

The calculations conducted on the storage are rough approximations. As shown in Chapter 5 the storage can be implemented several ways. The storage can cover either full load or partial load. It can cover just AC load, just the MT load or a combination of both. For this report the investigated solution is a storage for AC load. The cooling coil in the air handling unit will be connected in the PCM loop. Towards the end of the semester, it was also investigated implementing some of the MT cabinets to the loop. This was not a part of the simulation but was added to Chapter 8 Further work. This is a solution that can be investigated further for the next master students.

Most systems are over dimensioned and will be able to run even though one compressor were to faile. Productivity of equipment will decrease in periods of reduced demand. The system normally runs on 20-35% and rarely runs on maximum capacity (only few days in summer). Integrating a storage will make it possible to dimension the system for the base load and not the peak load. The chosen solution will make it possible to reduce the system by several components. The glycol circuit in the cooling coil can be removed and the cooling coil can be supplied directly from the CO_2-circuit. Also, the heat exchanger for AC will be removed reducing losses due to heat exchanging.

A seen from chapter 3.3 the maximum load for the system before having to use the fourth compressor is 68kW. With the new system the maximum load can be reduced to well below this number. Implementation of the storage will lead to reduced energy need during peak hours and thereby reduce the maximum peak load on the system. This shows that it can be possible to reduce the system with one compressor. However, these calculations are simplified, and some assumptions are made. Further calculation and experiments will be needed. Especially regarding charging and discharging cycles (losses, charging/discharging time etc). The heat transfer is complex due to the phase change. The storage is planned to be built in the NTNU lab. This will make it possible to run test (can be done for master students during the fall).

There is a lot of money in form of products stored in supermarkets. If the system were to faile a lot of products will have to be thrown out and the economic losses can be big. The storage will give the system a backup in case of failure. The self-circulating systems can operate while the main system can be fixed.

9.4 Storage size and heat exchangers

The calculations performed on storage size and number of heat exchangers are very simplified. Storage size is calculated based on the load curves. It is mentioned by the people in charge of the supermarket that the load during the summer months may be even lower than the load curves made for the system. The storage can then be even smaller than calculated.

The charging and discharging rate of the heat exchanger will vary with distance between the plates, time of discharging etc. The number of pillow plates is calculated based on research on the storage at the NTNU lab performed by Håkon Selvnes. The number of pillow plates was adapted to fit the maximum load. Further experiments need to be performed on the storage when it is built at the NTNU lab.

9.5 Economy and savings

Inserting a thermal storage will not necessarily give extra cost for the system. It can lead to reduced costs of the total system. Smaller pipes and components will reduce the cost of the system. The given solution makes it possible to remove several components like heat exchanger for AC and glycol circuit for the cooling coil and one compressor. The total cost of the storage is calculated to 229 400 NOK. This cost is lower than the saving for the rest of the system which is 321 900 NOK. Thereby the solution will be profitable based on the calculation and satisfies the required payback time of 3 years. However, one has to take into account that these calculations are very simplified and further calculations will be needed.

When considering costs of a system with thermal storage it is important to look at both investment cost and benefits during operation. Reduced electricity bills can be one of the advantages of thermal storage. There are also becoming more and more focus on environmental friendly facilities and technology in the world. BREEAM and other classifications of the buildings are becoming more important in addition to regulations from the government. Cost of the system will therefore not be the only factor when considering investments for the future.

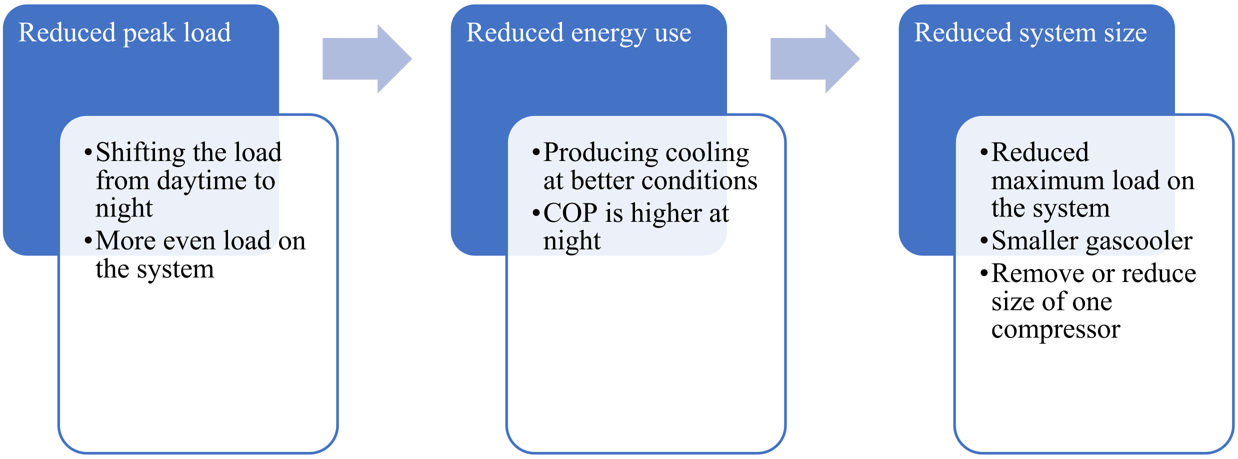
Implementation of the storage will lead to reduced energy consumption due to production at favourable conditions. Procuring energy at night will give better production conditions (colder at night). Producing energy at night requires less energy than procuring the same amount of energy at daytime. Therefor the energy need for the system will be reduced. This will give reduce operation costs. For the given day in July the saving is 49 NOK per day. However, this is on a warm summer day and there are very few days with these high temperatures in Norway. The savings will be bigger for countries with more warm days.

Norway has low electricity prices compared to other countries. This is due to the Norwegian hydropower. Big lakes are filled with water, making it possible to extract energy when we need it. This energy source is more stable than e.g. wind power that are only available when the wind is blowing. For countries who have more unstable energy sources the use of thermal energy storage is a good way to correct the mismatch that occur between supply and demand of energy. In many countries the electricity price is lower in the night than during the day. The price can vary as much as six times between night and day. Charging a storage at night when the price is low and using the energy at day when the price is high is therefore beneficial. Peak shaving will reduce the consumption during peak hours and thereby reduce the costs.

In Norway the electricity customers consuming large amount of electricity have different power agreement than other customers buying electricity for apartments, houses etc. For this customer the peak consumption will make up a large proportion of the electricity cost. Staubo AS shows that the peak consumptions can account for as much as 30-50% of the electricity costs. [78] However electricity prices are still low compared to other countries. Companies consuming large amount of electricity also have own agreements making electricity price lower. Peak shaving will probably be more profitable in the future. Norway recently connected their electricity grid to Europe. A giant cable extends from Norway to Germany. As electricity starts to flow across borders the electricity price can become more even for the different countries.

In industrialized countries things are becoming more electrified. Electrifying cars, boats etc will add a big load to the electricity grid. Peak shaving methods will help even out the demand from the grid during peak hours. This can be beneficial in areas where load on the grid is high and further development of shops requires development of the grid. If part of the building mass in an area where to install storages this can make it possible to avoid big developments in grid.

Chapter 10. Conclusion



Shifting the load from daytime to nigh time gives lower maximum power use in the peak hours. Reducing the maximum system load allows for downsizing of the system. The given solution will make it possible to reduce the system by several components. The glycol circuit to the cooling coil can be removed and the cooling coil can be supplied directly from the CO_2-circuit. Also, the heat exchanger for AC will be removed reducing losses due to heat exchanging. The maximum load for the system before having to use the fourth compressor is 68kW. With the current solution the maximum load can be reduced to well below this number. This shows that it can be possible to reduce the system with one compressor.

As shown from the investment analysis the implementation of the storage will not require any extra investment cost. The cost of the system is lower than the reduced cost of the existing system. However, one has to take into account that the calculations are simplified and that further calculations will be needed. Implementing a storage will also lead to reduced energy consumption and thereby lower operating costs. Producing energy at night requires less energy than procuring the same amount of energy at daytime. The energy costs will be reduced due to lower energy use at night and lower energy cost during off-peak periods.

Implementation of the storage will give more stable load on the system. The load can be kept at a specified load for a longer time. This will reduce on and off cycles for the compressors. This will lead to increased lifetime for components. The system will also act as a backup in case of system failure.

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Appendix A – Bitzer simulation supermarket 1

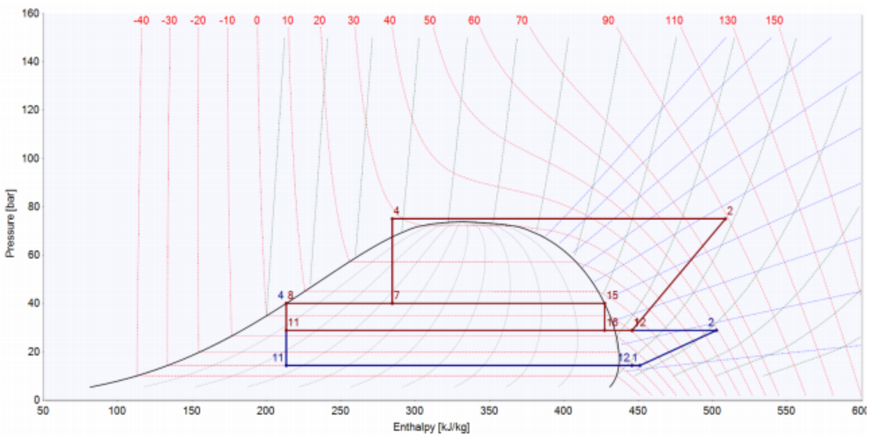
A.1 Input values and log P-h diagram

Selection: CO2 Systems

COP/EER Evaporator: 2,41

Input Values	**LT-Stage**	**MT-Stage**
System	**Flashgas**	
Series	ME (high standstill pressures)	Standard
Operating mode	Subcritical	Transcritical
Number compressors	1	4
Evaporating SST	-30,00 °C	-7,00 °C
Evaporator superheat	8,00 K	8,00 K
Suction line superheat	5,00 K	5,00 K
High pressure		Auto
Gas cooler outlet		29,0 °C
Intermed. pressure		40,0 bar(a) / 5,30 °C
Power frequency	50Hz	
Power voltage	400V	

p,h Diagram



LT-Stage

1 - 2	Compression
4 - 11	Expansion
11 - 12	Evaporation
12 - 1	Superheat suction line

MT-Stage

1 - 2	Compression
2 - 4	Gas cooling/Condensation
4 - 7	Expansion to intermediate pressure
7 - 8	Intermediate receiver: liquid outlet
8 - 11	Expansion to evaporation pressure
11 - 12	Evaporation
12 - 1	Total superheat
7 - 15	Intermediate receiver: gas outlet
15 - 16	Expansion to evaporation pressure

A.2 All compressors 50 Hz

Result

Compressor	LT-Stage	2MME-07K
Frequency compressor	--	50,0 Hz
Evaporator capacity	3,06 kW	3,06 kW
Ratio	--	100,0 %
Power input	0,68 kW	0,68 kW
Current	1,32 A	1,32 A
Voltage range	--	380-420V
Mass flow	47,4 kg/h	47,4 kg/h
Total superheat	12,90 K	12,90 K
Discharge gas temp. w/o cooling	49,0 °C	49,0 °C

Compressor	MT-Stage	4PTE-6K	4PTE-6K	4KTE-10K	4KTE-10K
Frequency compressor	--	50,0 Hz	--	--	--
Evaporator capacity	65,8 kW	9,60 kW	9,61 kW	23,3 kW	23,3 kW
Ratio	--	14,58 %	14,59 %	35,4 %	35,4 %
Gas cooler capacity	99,4 kW	14,49 kW	14,50 kW	35,2 kW	35,2 kW
Power input	27,9 kW	4,31 kW	4,29 kW	9,63 kW	9,63 kW
Current	48,4 A	7,70 A	7,93 A	16,37 A	16,37 A
Voltage range	--	380-420V	380-420V	380-420V	380-420V
Mass flow	1597 kg/h	233 kg/h	233 kg/h	566 kg/h	566 kg/h
Flashgas mass flow	532 kg/h	--	--	--	--
Total superheat	7,80 K	7,80 K	7,80 K	7,80 K	7,80 K
Discharge gas temp. w/o cooling	88,6 °C	91,5 °C	91,2 °C	87,5 °C	87,5 °C
optimal high pressure	75,0 bar(a)	--	--	--	--

: Tentative Data.
: Discharge gas temperature at least 50°C (122°F)
: Power consumption at compressor inlet.
MT-Stage: Tentative Data.
MT-Stage: Power consumption at compressor inlet.
MT-Stage: Total superheat smaller than 10K / 18°F.

A.3 Frequency controlled compressors 25/30Hz

Result

Compressor	LT-Stage	2MME-07K
Frequency compressor	--	30,0 Hz
Evaporator capacity	1,75 kW	1,75 kW
Ratio	--	100,0 %
Power input	0,43 kW	0,43 kW
Current	1,34 A	1,34 A
Voltage range	--	380-420V
Mass flow	27,1 kg/h	27,1 kg/h
Total superheat	12,90 K	12,90 K
Discharge gas temp. w/o cooling	54,2 °C	54,2 °C

Compressor	MT-Stage	4PTE-6K	4PTE-6K	4KTE-10K	4KTE-10K
Frequency compressor	--	25,0 Hz	--	--	--
Evaporator capacity	61,8 kW	4,39 kW	9,80 kW	23,8 kW	23,8 kW
Ratio	--	7,11 %	15,87 %	38,5 %	38,5 %
Gas cooler capacity	91,6 kW	6,51 kW	14,54 kW	35,3 kW	35,3 kW
Power input	25,8 kW	2,27 kW	4,29 kW	9,63 kW	9,63 kW
Current	48,4 A	7,74 A	7,93 A	16,37 A	16,37 A
Voltage range	--	380-420V	380-420V	380-420V	380-420V
Mass flow	1473 kg/h	104,7 kg/h	234 kg/h	567 kg/h	567 kg/h
Flashgas mass flow	490 kg/h	--	--	--	--
Total superheat	7,40 K	7,40 K	7,40 K	7,40 K	7,40 K
Discharge gas temp. w/o cooling	88,5 °C	99,8 °C	90,7 °C	87,1 °C	87,1 °C
optimal high pressure	75,0 bar(a)	--	--	--	--

: Tentative Data.
: Power consumption at compressor inlet.
MT-Stage: Tentative Data.
MT-Stage: Power consumption at compressor inlet.
MT-Stage: Total superheat smaller than 10K / 18°F.

A.3 Frequency controlled compressors 70/75Hz

Result

Compressor	LT-Stage	2MME-07K
Frequency compressor	--	75,0 Hz
Evaporator capacity	4,66 kW	4,66 kW
Ratio	--	100,0 %
Power input	1,09 kW	1,09 kW
Current	1,77 A	1,77 A
Voltage range	--	380-420V
Mass flow	72,2 kg/h	72,2 kg/h
Total superheat	12,90 K	12,90 K
Discharge gas temp. w/o cooling	51,8 °C	51,8 °C

Compressor	MT-Stage	4PTE-6K	4PTE-6K	4KTE-10K	4KTE-10K
Frequency compressor	--	70,0 Hz	--	--	--
Evaporator capacity	68,5 kW	13,68 kW	9,37 kW	22,7 kW	22,7 kW
Ratio	--	19,97 %	13,67 %	33,2 %	33,2 %
Gas cooler capacity	106,0 kW	21,2 kW	14,50 kW	35,2 kW	35,2 kW
Power input	29,7 kW	6,14 kW	4,29 kW	9,63 kW	9,63 kW
Current	50,3 A	9,66 A	7,93 A	16,37 A	16,37 A
Voltage range	--	380-420V	380-420V	380-420V	380-420V
Mass flow	1696 kg/h	339 kg/h	232 kg/h	563 kg/h	563 kg/h
Flashgas mass flow	565 kg/h	--	--	--	--
Total superheat	8,30 K	8,30 K	8,30 K	8,30 K	8,30 K
Discharge gas temp. w/o cooling	89,3 °C	91,0 °C	92,0 °C	88,3 °C	88,3 °C
optimal high pressure	75,0 bar(a)	--	--	--	--

: Tentative Data.
: Discharge gas temperature at least 50°C (122°F)
: Power consumption at compressor inlet.
MT-Stage: Tentative Data.
MT-Stage: Power consumption at compressor inlet.
MT-Stage: Total superheat smaller than 10K / 18°F.

Appendix B – Bitzer simulation supermarket 1

B.1 Input values and log P-h diagram

Selection: CO2 Systems

COP/EER Evaporator: 2,43

Input Values	LT-Stage	MT-Stage	Parallel-Stage
System		Flashgas	
Series	ME (high standstill pressures)	Standard	Standard
Operating mode	Subcritical	Transcritical	Transcritical
Number compressors	2	2	1
Evaporating SST	-27,20 °C	-1,90 °C	
Evaporator superheat	8,00 K	8,00 K	
Suction line superheat	25,6 K	14,90 K	5,00 K
High pressure		Auto	
Gas cooler outlet		33,0 °C	
Intermed. pressure		43,2 bar(a) / 8,30 °C	
Air Conditioning			28,4 kW
IHX Flashg. - Gas c.			8,50 K
Power frequency	50Hz		
Power voltage	400V		

B.2 Results simulation for 19.07.21

Result

Compressor	LT-Stage	2KME-1K	2KME-1K
Frequency compressor	--	70,0 Hz	--
Evaporator capacity	11,39 kW	6,67 kW	4,72 kW
Ratio	--	58,6 %	41,4 %
Power input	3,11 kW	1,86 kW	1,26 kW
Current	4,93 A	3,01 A	1,92 A
Voltage range	--	380-420V	380-420V
Mass flow	182,3 kg/h	106,8 kg/h	75,5 kg/h
Total superheat	33,5 K	33,5 K	33,5 K
Discharge gas temp. w/o cooling	82,4 °C	83,5 °C	80,9 °C

Compressor	MT-Stage	4PTE-7K	4MTE-10K
Frequency compressor	--	49,0 Hz	--
Evaporator capacity	23,8 kW	8,92 kW	14,84 kW
Ratio	--	37,6 %	62,4 %
Gas cooler capacity	96,2 kW	15,42 kW	25,6 kW
Power input	11,78 kW	4,54 kW	7,24 kW
Current	21,9 A	8,94 A	12,97 A
Voltage range	--	380-420V	380-420V
Mass flow	565 kg/h	212 kg/h	353 kg/h
Total superheat	41,4 K	41,4 K	41,4 K
Discharge gas temp. w/o cooling	134,5 °C	136,2 °C	133,6 °C
optimal high pressure	81,5 bar(a)	--	--

Compressor	Parallel-Stage	4JTE-15K
Frequency compressor	--	57,0 Hz
Evaporator capacity	28,4 kW	--
Ratio	--	100,0 %
Power input	11,28 kW	11,28 kW
Current	19,17 A	19,17 A
Voltage range	--	380-420V
Mass flow	1057 kg/h	1057 kg/h
Total superheat	13,50 K	13,50 K
Discharge gas temp. w/o cooling	79,0 °C	79,0 °C

: Tentative Data.
: Power consumption at compressor inlet.

Appendix C - Modelica simulations

The tables show the input values for the Modelica simulations in January, April and July

C.1 January

Hour	*Time*	*CapMT*	*CapLT*	*Heating*	*Tamb January*
0	0	0,572	0,8	0	-10
1	3600	0,572	0,8	0	-10
2	7200	0,572	0,8	0	-9
3	10800	0,572	0,8	0	-9
4	14400	0,572	0,8	0	-9
5	18000	0,572	0,8	0	-9
6	21600	0,572	0,8	0	-9
7	25200	0,572	1	1	-8
8	28800	0,572	1	1	-8
9	32400	0,716	1	1	-8
10	36000	0,716	1	1	-8
11	39600	0,86	1	1	-7
12	43200	0,716	1	1	-6
13	46800	0,572	1	1	-5
14	50400	0,572	1	1	-4
15	54000	0,572	1	1	-4
16	57600	0,716	1	1	-5
17	61200	1	1	1	-6
18	64800	0,716	1	1	-6
19	68400	0,572	1	1	-7
20	72000	0,572	1	1	-7
21	75600	0,572	1	1	-8
22	79200	0,572	1	1	-8
23	82800	0,572	1	1	-9
24	86400	0,572	0,8	0	-9

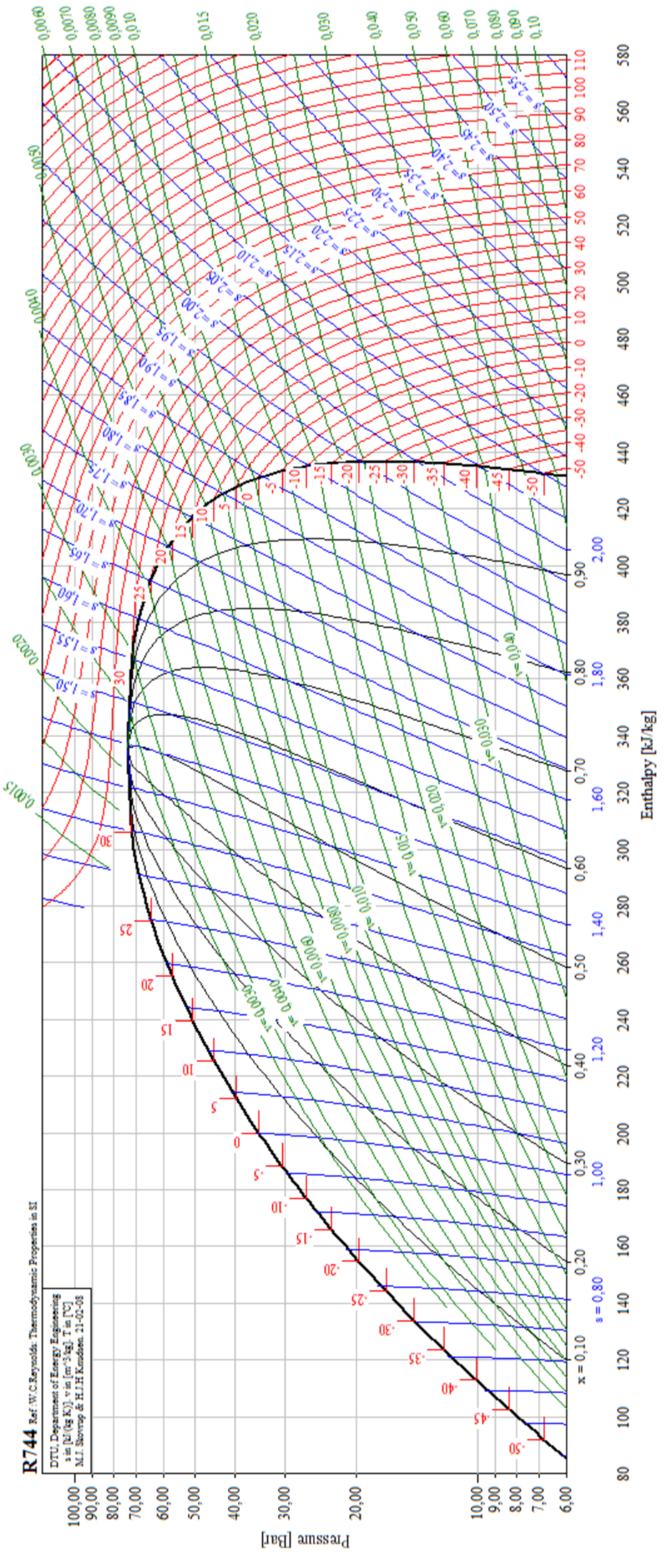
C.2 April

Hour	*Time*	*CapMT*	*CapLT*	*Heating*	*Tamb April*
0	0	0,572	0,8	0	5
1	3600	0,572	0,8	0	5
2	7200	0,572	0,8	0	6
3	10800	0,572	0,8	0	6
4	14400	0,572	0,8	0	7
5	18000	0,572	0,8	0	7
6	21600	0,572	0,8	0	7
7	25200	0,572	1	0	8
8	28800	0,572	1	0	9
9	32400	0,716	1	0	10
10	36000	0,716	1	0	11
11	39600	0,86	1	0	11
12	43200	0,716	1	0	12
13	46800	0,572	1	0	13
14	50400	0,572	1	0	14
15	54000	0,572	1	0	15
16	57600	0,716	1	0	15
17	61200	1	1	0	14
18	64800	0,716	1	0	14
19	68400	0,572	1	0	13
20	72000	0,572	1	0	12
21	75600	0,572	1	0	11
22	79200	0,572	1	0	10
23	82800	0,572	1	0	8
24	86400	0,572	0,8	0	6

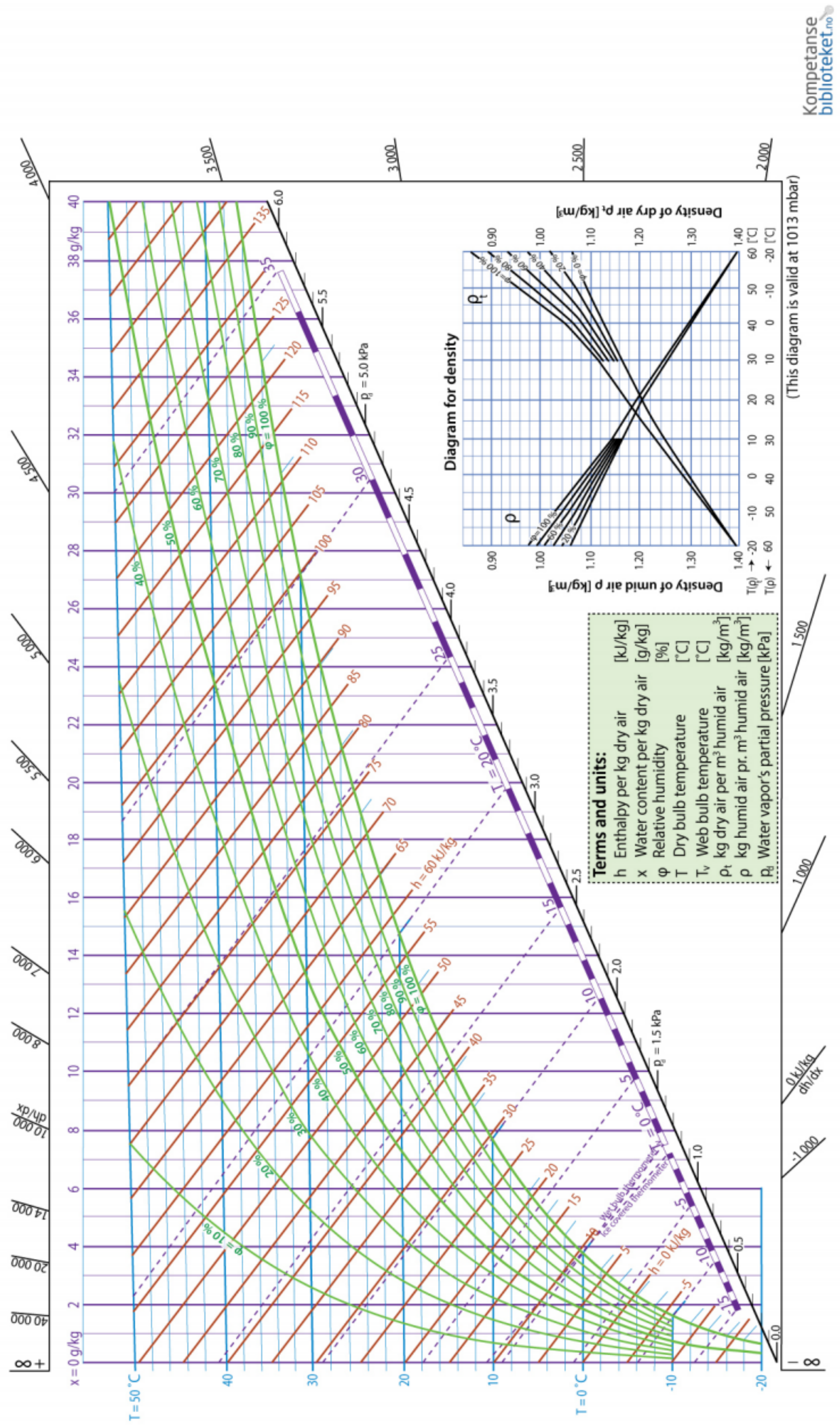
C.3 July

Hour	*Time*	*CapMT*	*CapLT*	*Heating*	*Tamb*	*AC_Need*	*AC_charge*
0	0	0,57	0,8	0	20	0	0,72
1	3600	0,57	0,8	0	20	0	0,72
2	7200	0,57	0,8	0	20	0	0,72
3	10800	0,57	0,8	0	21	0	0,72
4	14400	0,57	0,8	0	21	0	0,72
5	18000	0,57	0,8	0	22	0	0,72
6	21600	0,57	0,8	0	23	0	0,72
7	25200	0,57	1	0	24	0,14	0,72
8	28800	0,57	1	0	25	0,29	0,72
9	32400	0,72	1	0	25	0,29	0,6
10	36000	0,72	1	0	26	0,43	0,2
11	39600	0,86	1	0	28	0,71	0
12	43200	0,72	1	0	29	0,86	0
13	46800	0,57	1	0	29	0,86	0
14	50400	0,57	1	0	30	1	0
15	54000	0,57	1	0	30	1	0
16	57600	0,72	1	0	30	1	0
17	61200	1	1	0	29	0,86	0
18	64800	0,72	1	0	29	0,86	0,2
19	68400	0,57	1	0	28	0,71	0,32
20	72000	0,57	1	0	26	0,43	0,32
21	75600	0,57	1	0	25	0,29	0,32
22	79200	0,57	1	0	24	0,14	0,72
23	82800	0,57	1	0	22	0	0,72
24	86400	0,57	0,8	0	20	0	0,72

Appendix D - Log P-h CO_2



Appendix E - Mollier



Appendix F – Draft version of scientific report

See next page

INVESTIGATION OF R744 REFRIGERATION SYSTEM AND THE POSSIBILITY OF INTEGRATING A CTES

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ABSTRACT

This article summarize work done for master thesis spring 2021. The topic for the thesis is CO2 refrigeration systems for supermarkets and the possibility of implementation a thermal storage. The system to be investigated is an existing CO_2-only refrigeration unit. A simplified version of an existing system will be the basis for the simulation model that will be developed.

The report will look at the possibility of implementing a storage for the AC unit. Implementation and operation of the storage will be illustrated and explained. It will also be looked at how the implementation of the storage will affect the rest of the system, both size and cost will be considered.

Assuming that one install equal/bigger amount of storage capacity as the capacity of one of the compressors it can be possible to reduce the number of compressors for the system. It also may be possible to reduce the size of the existing system (components, piping, size of gas cooler etc). Simplified calculations shows that the cost of the storage can be equal or lower than the saving for the rest of the system. Thereby integrating a storage will be profitable based on the calculations. However, one has to take into account that these calculations are simplified, and further calculations will be needed.

Keywords: CO_2, commercial refrigeration, PCM storage, CTES

NOMENCLATURE

CO_2	Carbon dioxide
CTES	Cold thermal energy storage
PCM	Phase change material
PP	Pillow plate

1 INTRODUCTION

There is a large transition in supermarket refrigeration with a strong focus on reducing the energy demand and the installation cost. Highly efficient system configurations with R744 are introduced in various locations throughout Europe; however further improvements are necessary and possible, for example with the use of ejector-based expansion work recovery, pivoting compressor arrangements, and the implementation of local cold storages, etc.

Supermarket refrigeration systems have a significant energy demand to preserve the quality of valuable food products. Thus, several approaches to reduce the energy demand, and to solve other challenges a supermarket refrigeration system faces have been considered in the past years. One of the main challenges that has been addressed is the temperature instability of the food products in the display cabinets/freezers in general and during the defrost period. Other issues that have been considered are high electricity costs during peak demand, due to the grid dependency of the refrigeration unit, and the power consumption irregularity of the system.

Implementation of cold storages in combination with the R744 refrigeration system can have enough benefits which enables an implementation in the market. As cold thermal storage reduces the peak load and allows for shifting it to periods with low electricity cost or high electricity production with renewables (e.g. solar panels). These units can also lead to a radical downsizing of the compressor packs. The first approach to thermosyphon-driven, local cold storages has been experimentally investigated at NTNU's laboratory, but the concept needs to be refined and deeply investigated related to the impact on the total system architecture.

2 REFRIGERATION SYSTEM

System without storage

Figure 1 shows the existing system. The system is a centralized transcritical CO_2 booster system for a supermarket located in Norway. The system has one LT compressors and four MT compressor. The CO_2 heat pump supplies the cabinets and freezer with refrigeration. It also supplies the air handling unit (AHU) with cooling and heating for the ventilation air. The heating and cooling for the AHU is supplied with separate pipes with Monoethylene glycol (MEG 30%). The solution requires a lot of piping.

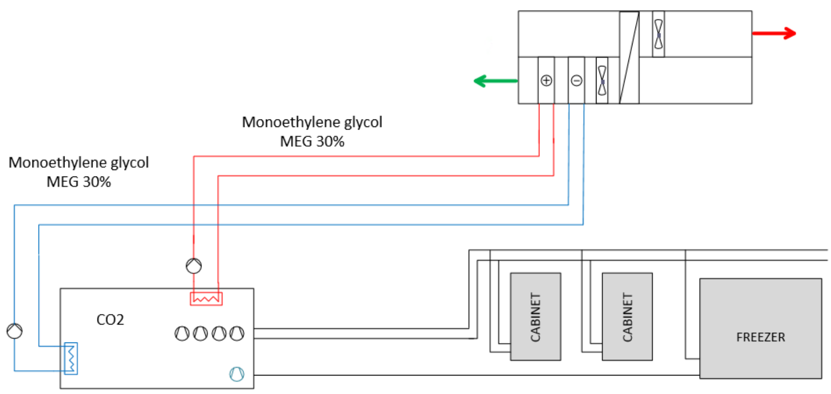

Figure 1: Existing system

System with storage

Figure 2 shows the new system with PCM storage. The PCM storage and cooling battery will be supplied by the same loop as the MT cabinets. There is no need for a separate glycol loop supplying the cooling battery. During charging the CO_2 circulates through the storage. It is possible to charge the storage and supply the cooling coil at the same time as indicated with the two arrows. During discharging the storage and cooling coil will be disconnected from the rest of the system and will be self-circulating

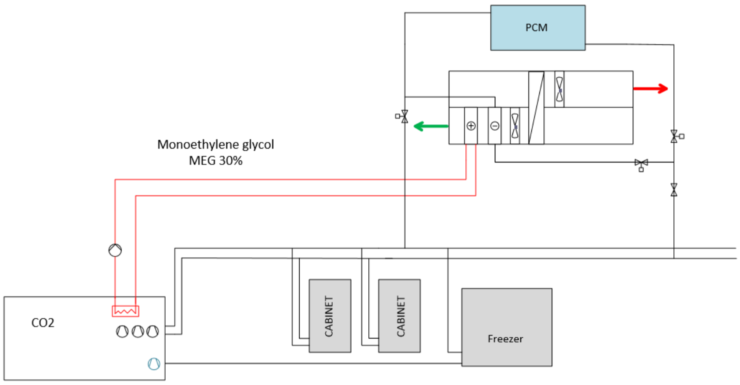

Figure 2: System with PCM storage

Self-circulating system

The PCM loop can be disconnected from the rest of the system during peak hours. This will reduce the total load on the system during hours of very high demand. The loop is shown in Figure 3. The loop will supply refrigeration for the Air condition system. During discharging the system will be self-circulating using gravity. This is referred to as a natural circulating system. The refrigerant evaporates in the cooling coil and rises up. Then the refrigerant enters the PCM tank and condenses as it is cooled by the melting PCM. The refrigerant passes through the expansion valve and into the cooling coil in AC system and the cycle is repeated. The system uses the thermosiphon principle that is defined by natural convection. This occurs due to the density difference in the refrigerant. The pressure difference between the two heat exchangers will drive the cycle. [1]

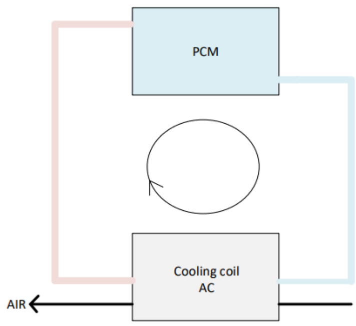

Figure 3: Self circulating PCM system

Placement

Placement of the storage will be important for the self-circulation to work. The storage needs to be placed above the AC unit and the height has to be big enough for the self-circulation to work. One should also avoid many bends and elbows in the loop. The storage will be placed in a container on the roof of the building. This way the storage will not occupy space inside the supermarket. The placement is shown in Figure 4. The roof is not used for anything specific, so placing the storage here will be a good solution for space reduction inside the supermarket.

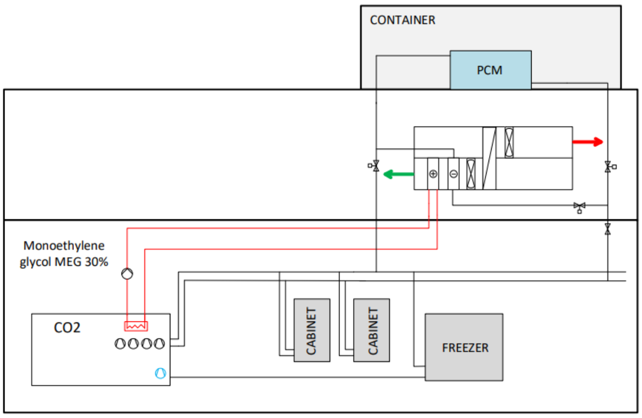

Figure 4: Storage placement

Storage geometry

The storage will consist of a tank with pillow plates stacked together. The number of pillow plates is adjusted to the calculated cooling load. The container is filled with water as the chosen PCM. The simplified geometry of the PCM storage is shown in Figure 5. The refrigerant enters the pipe in the upper part of the PCM storage tank as indicated in Figure 5. The pipe supplies all the PPs (pillow plates) with refrigerant. The system is going to be self-circulating using gravity and the PPs are therefore placed vertical to help making this possible. The simplified geometry of the PP is shown in Figure 6. The wavy structures of the pillow plates will give high heat transfer. Compared to conventional heat exchangers, pillow plates have low manufacturing costs and compact design. [2] When using this type of heat exchanger there is no need for a secondary fluid. The pillow plates can withstand the relatively high operating pressure of CO_2 systems. [3] The refrigerant can both evaporate and condense directly in the storage exchanging heat with the PCM material.

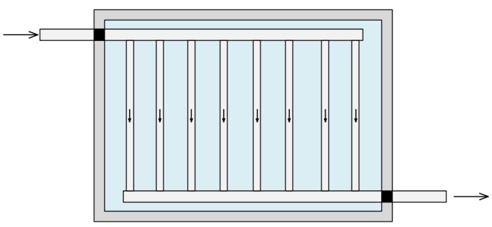

Figure 5: PCM storage

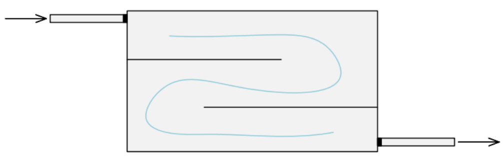

Figure 6: Pillow plate heat exchanger

2.1 Storage size and peak shaving

Figure 7 shows the storage solution. The different colours in the graphs represent the different loads on the system: Purple is the LT cabinets, dark blue is the MT cabinets and green indicates the charging of the storage. The black line shows the new load on the system with storage. The red line shows the load without the storage. The Full storage solution covers 100% of the AC load. The production at night is at maximum capacity with three MT compressors running. This is without using the fourth MT compressor. The load at night is 86% of maximum. This is equal to MT1, MT3 and MT4 running. This is the maximum load the system can have.

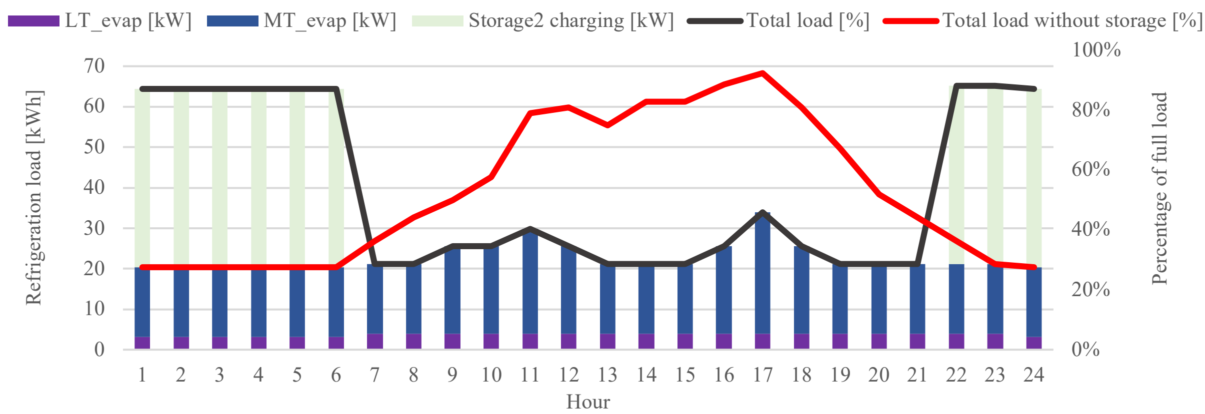

Figure 7: Refrigeration load with full AC storage solution 1

Storage size with full storage

The total AC need for an entire day is 394 kWh. Results for storage capacity and size is presented in Table 1. As one can see, a storage that covers the entire AC load will be heavy. The storage would be 4 264 kg. In addition to this the storage size will be bigger due to implementation of other components like heat exchangers, pipes and other storage components.

Table 1: Capacity of full PCM storage

Needed AC [kWh]	394
Factor due to losses	1.15
Actually needed storage [kWh]	453
Total amount of ice if nothing else[kg]	4 264
Length[m]	1.5
Width[m]	1.6
Height[m]	2.0

2.2 Heat exchanger size

Calculation has been conducted to find required number of pillow plates to supply the given cooling in Chapter 2.1. Table 2 shows the number of plates needed for different inlet temperatures. The power per m^2 is calculated based on experiments performed by Selvnes et.al [4] [5] on the storage at the NTNU lab. The discharge rate has to be adapted to fit the load curve of the system. During peak hours the system must be able to supply the maximum AC load which is 40 kW. The self-circulation can limit the flow in the system. Therefore the calculations are based on the smallest mass flow from the experiment, which is 4 kg/min. Higher temperatures give higher pressure fall and higher proportion of evaporated CO_2 (gas quality) entering the CTES. There is big difference in number of pillow plates. The solution with lowest inlet temperature have two times as many pillow plates as the solution with highest temperature.

Table 2: Calculated number of pillow plates needed

	4kg/min and 5,5°C	***4kg/min and 7°C***	***4kg/min and 10°C***
Maximum effect AC [kW]	40	40	40
Power pr m2 [kW/m2]	0.3	0.4	0.6
Area per pillow plate [m^2]	6	6	6
Power pr plate [kW/plate]	1.8	2.4	3.8
Number of plates	22	16	11

3 INVESTMENT

It is challenging to calculate costs of the system. Prices will vary a lot depending on the number of components. If one were to order 1 or 100 heat exchangers the price will be different. Large quantity will give lower prices. It is also dependent on whether the contractor that is going to build the system is experienced on PCM storages or not. An experienced contractor will use less hours than an inexperienced one. The first time the storage system is created the cost of engineering hours will be high. However, when the first storage system is built the cost of the other systems will be lower. The system will have a standard solution. For the analysis it is assumed that the contractor is experienced and that this is not the first system to be built. It is also assumed that the developer buys many systems not just one. The total cost of the storage system is calculated to 229 400 NOK. Total investment analysis can be found in report for the master thesis [6].

Inserting a thermal storage will not only give extra cost for the thermal storage. It can also lead to reduced costs of the total system. Smaller pipes and components will reduce the cost of the system. It can also be possible to reduce the system by one compressor. The total reduced costs are 321 900 NOK. In addition to the reduced size of the initial system, the new system solution will have reduced operating cost. Implementation of the storage will lead to reduced energy consumption due to production at favourable conditions. Producing energy at night requires less energy than procuring the same amount of energy at daytime. Therefor the energy need for the system will be reduced. This will reduce operation costs. The energy savings per day is 50 kWh corresponding to 49 NOK/day.

The required payback time for this project is 3 years. Assuming that the total savings for the system would be 321 900 NOK and the total cost of the storage is 229 400 NOK. The payback time will be zero. Thereby the solution will be profitable based on the calculation. However, one has to take into account that these calculations are very simplified and further calculations will be needed. The system will also give reduced operating costs. The storage operates at night which gives better COP.

4 SIMULATION RESULTS

Figure 8 shows compressor on/off and Figure 9 shows the energy input for the compressors. In January there are only one compressor that are running at all times MT1. MT2 turns on during the peak hours at the middle of the day. In April there is still only one compressor that runs continuously throughout the day MT1. The second compressor MT2 turns on and off more frequently than in January.

In July without storage the compressors turns on and off more frequently. MT1 and MT2 are running continuously throughout the day, while MT3 and MT 4 turns on and off during the peak hours in the middle of the day. For the simulation with storage the last compressor MT4 is not running at all. In addition to this the on and off cycles are reduced, and the system runs more stable than without storage.

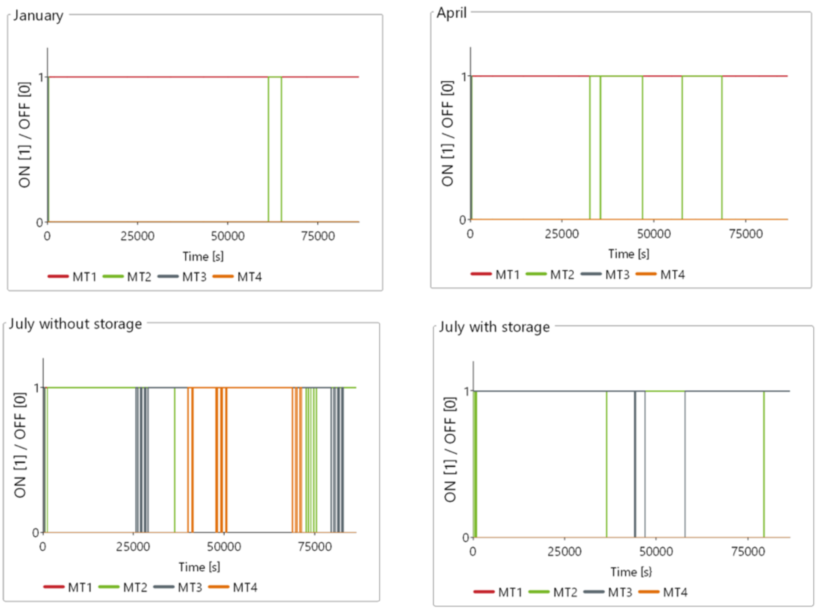

Figure 8: Compressors On/Off. From upper left: January, April, July without storage, July with storage

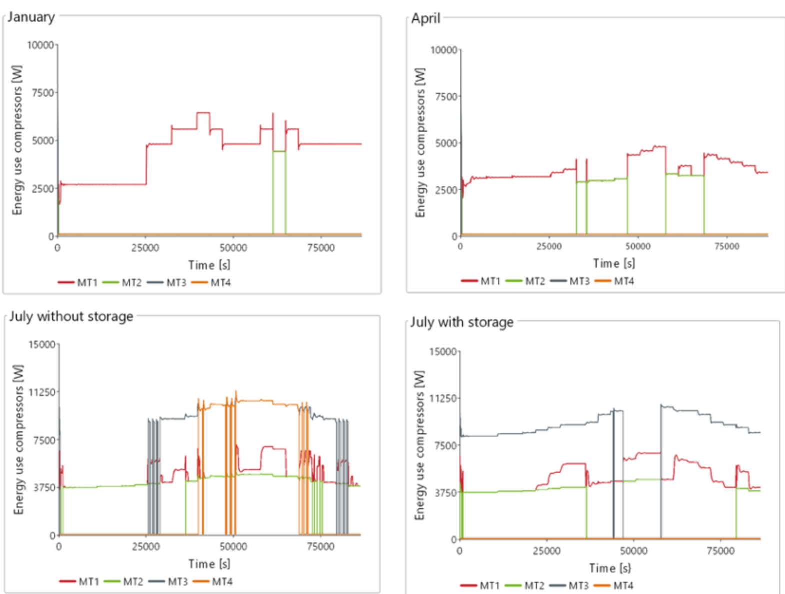

Figure 9: Energy use for compressors. From upper left: January, April, July without storage, July with storage

Figure 10 shows the energy use for the different months. As one can see the energy use for January and April will be much lower than in July. January and April has a very linear graph. These months does not have AC need. As one can see there will be difference in energy use for July with storage and without storage. For the simulation with storage the energy use has a more linear graph, indicating that the energy use is more even during the day (constant slope number). The simulation without storage has a steeper slope in the middle of the day, which is the peak hours. The total energy use with and without storage will also be different. The system with storage will produce energy at night with more favourable conditions, hence a better COP. The required energy input will therefore be lower.

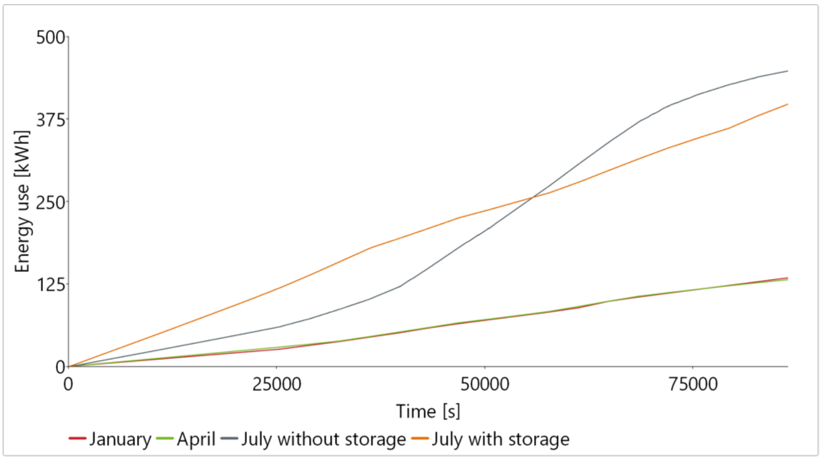

Figure 10: Energy use for the different months

5 CONCLUSION

Shifting the load from daytime to nigh time gives lower maximum power use in the peak hours. Reducing the maximum system load allows for downsizing of the system. The given solution will make it possible to reduce the system by several components. The glycol circuit to the cooling coil can be removed and the cooling coil can be supplied directly from the CO_2-circuit. Also, the heat exchanger for AC will be removed reducing losses due to heat exchanging. The maximum load for the system before having to use the fourth compressor is 68kW. With the current solution the maximum load can be reduced to well below this number. This shows that it can be possible to reduce the system with one compressor.

As shown from the investment analysis the implementation of the storage will not require any extra investment cost. The cost of the system is lower than the reduced cost of the existing system. However one have to take into account that the calculations are simplified and that further calculations will be needed. Implementing a storage will also lead to reduced energy consumption and thereby lower operating costs. Producing energy at night requires less energy than procuring the same amount of energy at daytime. The energy costs will be reduced due to lower energy use at night and lower energy cost during off-peak periods.

Implementation of the storage will give more stable load on the system. The load can be kept at a specified load for a longer time. This will reduce on and off cycles for the compressors. This will lead to increased lifetime for components. The system will also acts as a backup in case of system failure.

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