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Energy distribution concepts for Urban Supermarkets including energy hubs

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Master of Science in Mechanical Engineering

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MASTER THESIS

for

Student: Håkon Selvnes
Spring 2017

Energy distribution concepts for Urban Supermarkets including energy hubs

Energikonsepter for urbane butikker med energistasjoner

Background and objective

Building plots in urban areas are very valuable, i.e. utilizing the limited area for various purposes is important in the future. Leading Supermarket chains are developing new building concepts with a combination of sales areas, parking lots with energy supply hubs (EL, HY) and flats/apartments in the upper floors.

The entire energy distribution system should be centralized and integrated with respect to minimize the import of primary energy towards the site. The Master Thesis work describes different energy concepts and scenarios for the different energy units and their interaction to optimize the utilization of surplus heat.

The objective of the Master Thesis work is to develop/apply an energy flow simulation tool (Modelica), enabling the evaluation of different energy system concepts.

The following tasks are to be considered:

1. Literature review: energy systems for high performance buildings, hydrogen filling stations, surplus heat recovery concepts, energy storage devices (PCM), etc.
2. Describe and develop the sub-systems for the energy dirstribution and storage systems, including interfaces to external suppliers or clients (building, hydrogen station, etc).
3. Develop/apply an energy flow (hour by hour) simulation tool (Modelica) to evaluate sytem alternatives.
4. Analysis and discussion of simulated scenarios
5. Summary report
6. Draft scientific paper related to the findings of the Master Thesis
7. Proposals for further work

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

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

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

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

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

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

The final report is to be submitted digitally in DAIM. An executive summary of the thesis including title, student's name, supervisor's name, year, department name, and NTNU's logo and name, shall be submitted to the department as a separate pdf file. Based on an agreement with the supervisor, the final report and other material and documents may be given to the supervisor in digital format.

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

Department of Energy and Process Engineering, 15. January 2017

Prof. Dr.-Ing. Armin Hafner
Academic Supervisor

Research Advisor: Prof. Trygve M. Eikevik

I want to dedicate this thesis to the two women in my life.
To my beloved daughter Ida, you are truly the sun that always shines in my life.
To my beautiful Randi. Thank you for all the support at home and being patient when my attention was required elsewhere.
I love you both.

Summary

More and more of the earths population is living in urban areas. This indicates that using urban building plots for combined purposes will be important in the future. For this reason supermarket chains are developing new building concepts that includes supermarket sales area, apartments on the upper floors and parking lots with energy supply hubs for the new generation of hydrogen/electric vehicles. This master thesis emphasizes this concept by designing the complete energy system for such a building and taking into account demands for heating, cooling and refrigeration. A literature review of subsystems and components relevant for the different parts of the building concept was performed. This includes supermarket refrigeration systems, hydrogen refueling stations, energy systems for high performance buildings and thermal storage. Based on the findings from the literature study, a complete design for the building energy system was proposed. The purpose of the energy system is to integrate the different subsystems together to enable heat recovery to use for space heating and domestic hot water, and thereby minimizing the import of primary energy to the building plot.

A key feature of the energy system is that it is organized in three circuits with different temperature levels. They include a high temperature water circuit for domestic hot water, a medium temperature water circuit for space heating and low temperature antifreeze circuit connected to boreholes for heat storage, cooling of ventilation air and as a heat source. Space heating in the building is covered by a low temperature 28/33°C return/supply floor heating system as well as heating of ventilation air. Hour-by-hour energy demand over a year for space heating and domestic hot water for the building was obtained by developing a model of the building envelope in the software SIMIEN. The data obtained was based on assuming a plausible layout and design for this type of building. The supermarket occupies the whole first floor of the building and thirty apartments the three next floors, separated into ten apartments on each floor. The building envelope is designed according to the requirements and specifications to the Norwegian passive house standard. Four weeks were picked from the yearly demand report from SIMIEN to give as input to the simulation of the energy system. The four weeks serves as boundary conditions for a performance investigation of the energy system during spring, summer, fall and winter.

To evaluate the chosen design of the energy system three different variants of the system, called case 1, 2 and 3, was developed in the dynamic simulation software Dymola. The three models were simulated for all four weeks and the results compared. Case 1 involves integrating the supermarket refrigeration system to the centralized heating system in the building and delivering waste heat from the gas coolers to space heating and domestic hot water. Simulations show that by operating the system in trans-critical mode, heat recovery could cover the full demand of domestic hot water in the building for all weeks. In addition, the full space heating demand could be covered by heat recovery for the summer and spring week, and reaching a share of 75.4% and 31.4% for the fall and winter

week. Remaining heat demand in the building was covered by a R290 ground source heat pump connected to the energy wells on the evaporator side.

Case 2 and 3 investigate heat recovery from the hydrogen refueling station in two different ways in addition to heat recovery from the supermarket refrigeration system. The waste heat at the hydrogen station is due to the operation of an electrolyser, compression of hydrogen gas and precooling of hydrogen gas during filling of vehicles. The available waste heat/refrigeration load in case 2 and 3 was linked to the daily production level of hydrogen gas in the electrolyser. The production was assumed constant through the day, and equal for all days of the week, giving a constant waste heat availability. In case 2, an antifreeze cooling circuit from the hydrogen station brings waste heat at 45°C to the centralized heating system of the building, and the heat is recovered to space heating by a heat exchanger. In case 3, the hydrogen station is integrated entirely with the CO_2 booster refrigeration system in the supermarket, and the refrigeration demand at the station is covered by it. The first comparison between case 1, 2 and 3 was based on a daily production of 50 kg of hydrogen fuel, corresponding to 25% of maximum installed capacity. Results showed that case 2 and 3 displayed similar performance for space heating heat recovery, where both could satisfy the space heating demand for summer, spring and fall week, and about 60% during a winter week. Due to high uncertainty in the demand for hydrogen fuel in the area, a parameter study of the hydrogen production level was carried out. The production level was varied from 10% to 100% and simulated for the winter week. For both case 2 and 3, a daily production level of 75% (150 kg) and higher could cover all space heating demand. In addition, case 3 showed a large potential to supply domestic hot water, up to 5.5 times the demand of the building at maximum daily production of 200 kg hydrogen. Case 1 is considered the minimum integration that should be carried out for the building concept, as large energy savings can be achieved with small modifications to a standard CO_2 booster system. If the hydrogen station is to be integrated, the case 3 design is the better option. It can deliver similar performance to space heating as case 2, but can in addition deliver a large supply of domestic hot water. If case 3 design of the system is chosen, delivering domestic hot water to neighbouring buildings should be considered to use the full potential of waste heat that is available.

Sammendrag

En stadig større andel av verdens befolkning lever i urbane områder. Dette medfører et stort press på arealer i byer og tettbebygde områder. Dette tilsier at kombinert bruk av urbane arealer vil bli viktig i fremtiden. På grunn av dette utvikler nå ledende dagligvarekjeder ulike bygningskonsepter som inneholder dagligvarebutikk, leiligheter i etasjene over butikken og parkeringsplasser med lade- og fyllestasjoner for den neste generasjonen av elektriske- og hydrogendrevne biler. Denne masteroppgaven tar tak i dette konseptet ved designe et komplett energisystem for en slik bygning som skal ta hensyn til varme-, kjøle- og frysebehov i bygningen. Et grundig litteraturstudie av delsystemer som er relevanet for de ulike delene av bygningen ble gjennomført. Dette inkluderer kjøleanlegg til dagligvarebutikker, fyllestasjoner for hydrogen, energisystemer for høy-ytelsesbygninger og termisk lagring. Hensikten med energisystemet i bygningen er å integrere de ulike delsystemene sammen til et sentralt energisystem for å kunne bruke varmegjenvinning. Spillvarmen skal kunne brukes til oppvarming av bygningen og varmt tappevann, slik at import av primærenergi til tomten minimeres.

Et viktig særtrekk til det valgte designet for energisystemet er å organisere det i tre kretser med ulike temperaturnivå. Dette inkluderer en høytemperatur vannkrets for varmt tappevann, en vannkrets med mellomtemperatur for oppvarming og en lavtemperaturkrets med kjølemiddel koblet til energibrønner for varmelagring, kjøling av ventilasjonsluft og som varmekilde. Oppvarming av bygningen gjøres med lavtemperatur vannbåren varme med tur/retur temperatur på 33/28°C til gulvvarme og oppvarming av ventilasjonsluft. Tappevann- og oppvarmingsbehov for et år i bygningen ble kartlagt ved å bygge en simuleringsmodell av bygningen i programmet SIMIEN. Resultatene fra simuleringen var basert på å anta en sannsynlig layout og struktur til slik bygning. Dagligvarebutikken okkuperer hele første etasje, og i de tre etasjene over er tretti leiligheter er fordelt med ti i hver etasje. Bygningskroppen ble designet etter krav og spesifikasjoner i den norske passivhusstandarden NS3700. Fire uker fra hver sin årstid i årssimuleringen ble plukket ut for å gi som input til simulering av energisystemet. De fire ukene skal representere grensebetingelser for temperatur og oppvarmingsbehov i en kartlegging av ytelsen til energisystemet gjennom vinter, vår, sommer og høst.

For å evaluere det valgte designet av energisystemet ble tre ulike variasjoner av systemet, case 1, 2 og 3, bygget i det dynamiske simuleringsprogrammet Dymola. De tre modellene ble simulert for alle de fire valgte ukene og resultatene ble sammenlignet. I case 1 er CO_2 kjøleanlegget i dagligvarebutikken integrert med det sentraliserte oppvarmingsanlegget i bygningen, og gasskjølerne i anlegget leverer spillvarme til romoppvarming og varmt tappevann. Resultat fra simuleringene viser at ved å kjøre kjøleanlegget i transkritisk modus på 85 bar, kan varmegjenvinning dekke hele tappevannsbehovet i bygningen for alle ukene. I tillegg kan alt oppvarmingsbehov dekkes av varmegjenvinning i vår- og sommeruken, og systemet lever en oppvarmingsandel på 75.4% og 31.4% for høst- og

våruken. Resterende varmebehov dekkes av en R290 propanvarmepumpe koblet til energibrønnkretsen på fordampersiden.

I case 2 og 3 blir varmegjenvinning fra fyllestasjonen for hydrogen undersøkt på to ulike måter i tillegg til varmegjenvinning fra dagligvarebutikkens kjøleanlegg. Spillvarme fra hydrogenstasjonen kommer fra drift av en elektrolysør, hydrogenkompressorer og forkjøling av gassen ved fylling av biler. Tilgjengelig spillvarme/kjølebehov ved stasjonen i case 2 og 3 ble knyttet til den daglige produksjonen av hydrogengass i elekrolysøren. Den ble antatt konstant gjennom dagen og lik for alle dager gjennom uken. I case 2 kobles en krets med kjølemiddel fra hydrogenstasjonen til det sentrale oppvarmingssystemet. Kretsen leverer spillvarme på 45°C som varmeveksles med det vannbårne oppvarmingsanlegget i bygningen. I case 3 integreres hydrogenstasjonen helt med CO_2 kjøleanlegget i dagligvarebutikken, og kjølebehovet dekkes av dette. For den første sammenligningen mellom case 1, 2 og 3 ble det antatt en daglig produksjon på 50 kg hydrogen, tilsvarende 25% av maksimal kapasitet. Resultatene viser at case 2 og 3 leverer omtrent samme mengde spillvarme til oppvarming, og begge variantene av systemet kan tilfredstille behovet for oppvarming i høstuken og levere omtrent 60% av varmebehovet i vinteruken. På grunn av høy usikkerhet i etterspørselen etter hydrogendrivstoff ble en parameterstudie av daglig hydrogenproduksjon gjennomført. Den ble variert fra 10% til 100% av installert kapasitet og simulert for vinteruken. For både case 2 og 3 var en daglig produksjon av hydrogen på 75% (150 kg) og høyere nok til å dekke oppvarmingsbehovet i bygningen med spillvarme. I tillegg viste case 3 et stort potensiale til å levere varmt tappevann, opp til 5.5 ganger høyere enn behovet i bygningen ved maksimal produksjon av hydrogen på 200 kg/dag. Fra resultatene er det konkludert med at case 1 er et minimum av integrering som bør gjennomføres i energisystemet for bygningskonseptet. Store energibesparelser kan oppnås med små modifikasjoner av et standard butikkjøleanlegg. Hvis hydrogenstasjonen skal integreres er case 3 det beste alternativet. Case 3 leverer sammenlignbar ytelse til oppvarming som case 2, og i tillegg er mye varme tilgjengelig til oppvarming av tappevann. Hvis løsningen med case 3 velges for bygningens energisystem, bør man levere varmt tappevann til omkringliggende bygninger for å utnytte det fulle potensialet av spillvarme som er tilgjengelig.

Preface

This Master Thesis summarize my work during the spring of 2017 at the Norwegian University of Science and Technology, Department of Energy and Process Engineering. The topic for the thesis is formulated on the basis of relevant problems and challenges associated with energy consumption and area use in urban areas. The interaction between different energy systems are studied to maximize the heat recovery and minimize import of primary energy to a building.

I wish to express my gratitude to my supervisor, professor Dr.ing Armin Hafner at NTNU, for good discussions and advise during my work with this thesis. I will also like to thank Dr. Ángel Álvarez Pardiñas for helpful guidance concerning the modelling software Dymola during startup and troubleshooting during the work. To fellow student Silje Marie Smitt, thank you for good discussions during the work. Finally I would like thank my dear Randi, my daugther Ida, rest of my family and friends for all your support during my study at NTNU.

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Abbreviations

DHW	=	Domestic hot water
HT	=	High temperature
MT	=	Medium temperature
LT	=	Low temperature
COP	=	Coefficient of Performance E
=	energy	
m	=	mass
h	=	specific enthalpy
V	=	volume
ρ	=	density
Q	=	Heat flow

Chapter 1

Introduction

Global warming and climate change have led researchers to search for sustainable solutions to reduce the emission of greenhouse gases by focusing on developing energy efficient buildings, industrial plants and agriculture.

Supermarkets have one of the highest specific energy consumption per m^2 of all commercial buildings. According to Nordtvedt and Hafner (2012), the energy consumption of supermarkets are in the range of 300-600 kWh/m^2 per year. Other commercial buildings, such as offices, demonstrate significantly lower numbers, typically in the range of 150-200 kWh/m^2 per year. Tassou et al. (2011) reports that supermarkets are responsible for approximately 3% of the annual electricity consumption in the UK. Between 30-60% of this electrical energy is used by the refrigeration plant in the supermarket. This shows that installing energy efficient refrigeration systems in supermarkets are important when addressing energy savings and efficient energy use in the sector.

In addition to the high usage of electrical energy, supermarkets are also a large source of direct emissions of greenhouse gases. Yearly leakages of refrigerants are in the range of 15-20% of the total installed charge in Europe, and refrigerants with high global warming potential (GWP) values are used in a large majority of the systems (Hafner et al., 2012). To reduce the environmental impact of leakages from supermarket refrigeration systems and other sources where refrigerants are used, the European Union revised the F-gas regulation and put it into action from 1st of January 2015 (European Union, 2014). The revised regulation includes ban and restrictions on use and service of systems using high GWP refrigerants. The aim is to reach a total phase-out of HFCs and other strong greenhouse gases as refrigerants. This has led scientists to focus research on natural working fluids like CO_2, ammonia and hydrocarbons. These working fluids have low GWP values, are stable compounds and their behaviour once released to the atmosphere is well known. Using natural working fluids help reduce the consequence of direct emission of refrigerants from refrigeration systems and heat pumps.

The conventional fossil fuels are becoming more and more expensive as the worlds oil and gas reserves are depleting. In addition to the increase concern of climate change, this has led researchers to focus on finding sustainable alternatives to replace fossil fuels. Many believe that hydrogen will play an important role in the future as an energy carrier along with electricity (Rosen and Koohi-Fayegh, 2016). As an action towards more sustainable transport sector, several automotive companies are developing a wide range of non-fossil fuel vehicles for personal transport, i e. electric and hydrogen fuel cell electric vehicles. Honda, Toyota and Hyundai are the leading automotive companies on fuel cell vehicles, but also European manufacturers are planning to release fuel cell vehicles in the period until 2020. Toyota and Honda has already started exporting their commercial fuel cell vehicles to the US, and sales as of April 2017 were 1523 of the Toyota Mirai, and 134 of the Honda Clarity (Carsalesbase.com, 2017).

More and more of the earths population lives in urban areas and cities. In the European Union, the percentage of the population which lives in urban areas has increased from approximately 70 to 75 % from 1990 to 2015 (World Bank Data, 2017). This means that building plots in urban areas are becoming more and more valuable, and good utilization of urban areas for combined purposes will be important in the future.

This master thesis aims to develop a building concept for an urban supermarket where the entire energy distribution system is integrated and centralized. The building concept includes a supermarket with sales area, parking lot with energy supply hubs for hydrogen vehicles and apartments on the upper floors of the building. The system solutions proposed for the energy system and building will be based on an extensive literature study of different energy subsystems that are found on the property. The overall goal of the thesis is to minimize the import of primary energy to the building plot. A simulation model of the building energy system will be developed to study and evaluate the feasibility of the proposed system design, and investigate the potential for using heat recovery to cover energy demands.

Chapter 2

Literature Review

2.1 Supermarket refrigeration systems

The refrigeration plant is a vital part of any supermarket, as it provides chilling and freezing of food and goods in display cabinets located in the sales area, and in cold rooms. The refrigeration plant is also the largest consumer of electrical energy in a supermarket, accounting for 30-60% of the annual electrical energy consumption (Tassou et al., 2011; Ducoulombier et al., 2006). This number varies based on a number of parameters, such as type of refrigeration system, size of the store, geographical location and local climate.

A large majority of the refrigeration plants in Europe employ HFC gases as refrigerant, and R404a in particular (Hafner et al., 2012). There are inevitable leakages from a refrigeration plant and because of this, the refrigeration systems are directly emitting strong greenhouse gases into the atmosphere. To reduce the environmental impact of refrigeration systems, a revision of the EU f-gas regulation concerning HFC gases from 2006 was put in to action from 1st of January 2015 (European Union, 2014). The f-gas regulation includes different bans and restrictions on use and service of systems with high GWP refrigerants, including domestic and commercial systems. For commercial refrigeration systems over 40 kW, a maximum limit of 150 GWP is set in action from 1st of January 2022. As a reaction, the use of natural refrigerants has received attention as a sustainable and competitive alternative to HFC refrigerants (Gullo et al., 2016; Tassou et al., 2011). Natural working fluids are substances that are naturally present in the biosphere. Examples of natural working fluids are ammonia, hydrocarbons and carbon dioxide.

R744/CO_2 is the most promising natural refrigerant for commercial refrigeration. High operating pressure and low critical temperature are at first glance the drawbacks of the CO_2, but on the contrary it is interesting because of very favorable thermo-physical properties. These include higher latent heat, density, specific heat, thermal conductivity and volumetric cooling capacity than HFC refrigerants. In addition, the fluid has low viscosity, it is non-flammable, non-toxic and has very low global warming potential.

An arrangement for a supermarket refrigeration system typically consists of two circuits, one for medium temperature (MT) and one for low temperature (LT). The medium temperature circuits are used for chilled food cabinets and displays, while the low temperature circuits are used for frozen food. There are three main system solutions where CO_2 is used in a refrigeration plant for a supermarket (Sawalha, 2008):

- Indirect system
- Cascade system
- Trans-critical system

In a indirect system, CO_2 is used as a two-phase secondary fluid for low temperature applications. In this application CO_2 replaces conventional one-phase secondary fluids, i e. antifreeze or brine. CO_2 used as a secondary refrigerant for supermarket application has proven to have several advantages, including low pumping work and excellent heat transfer properties (Ge and Tassou, 2011). But still, the primary refrigerant in the refrigeration system will very often be a HFC like R404a.

In a cascade system CO_2 is most often used in the lowest temperature part of the cascade and is evaporated in the freezing cabinets. A cascade system solution requires that a cascade heat exchanger exists in the system for condensing the CO_2 against the refrigerant in the higher temperature part of the cascade. Because of this, there is a need for a temperature difference in the cascade heat exchanger to ensure heat transfer between the two refrigerants. This means lowering the evaporation temperature of the high side refrigerant. The result is a larger required pressure lift for the high side compressors and a lower COP for the system (Ge and Tassou, 2011). The type of high side refrigerant can vary, but commonly used refrigerants are ammonia, hydrocarbons, R404a or R134a (Sawalha, 2008).

The trans-critical system for CO_2 eliminates the need for a cascade heat exchanger as it uses a common refrigerant for the whole system. A booster configuration is most commonly used, where the refrigerant is cooled on the high pressure side, and then expanded down to both low and medium temperatures (pressure). The temperature levels for freezing and chilling are often referred to as LT and MT. The refrigerant is then evaporated at both temperature levels. The system is has two compressor stages, where the low pressure stage operates sub-critically and the high pressure stage operates either trans-critically or sub-critically, depending on the ambient conditions (Ge and Tassou, 2011). This paper will focus on natural refrigerant systems only, and a R744 booster system is considered state of the art technology for energy efficient and environmentally friendly supermarket refrigeration. Therefore the only system solution described further is the trans-critical CO_2 booster system.

2.1.1 R744 booster refrigeration system

A typical trans-critical CO_2 booster system is shown in figure 2.1. The system consists of four pressure levels, that is low, medium, intermediate and high. The pressure levels can easily be observed in a logarithmic pressure-enthalpy (LogP-H) diagram. A LogP-H diagram for the CO_2 trans-critical booster system is shown in figure 2.2. The numbering of locations indicated on the system schematic in figure 2.1 corresponds to the state points in figure 2.2. The low and medium pressure sections starts at the outlet of their respective thermostatic expansion valves. The low pressure side runs from the outlet of the low pressure expansion valve, through the low pressure evaporator and ends at the inlet of the low pressure compressor. Similarly, the medium pressure part of the system runs from the outlet of the medium pressure expansion valve, through the medium pressure evaporator and to the inlet of the high pressure compressor. The high pressure side starts at the outlet of the high pressure compressor, through the gas cooler and to the inlet of the expansion valve before the receiver tank. The intermediate pressure level starts at the outlet of the high pressure expansion valve, through the receiver and to the inlet of the bypass valve and low/medium pressure expansion valves. Since the outlet flow of the low pressure compressor and the bypass valve is mixed before the high pressure compressor, a suction gas heat exchanger can be included (not shown on Figure 2.1). This heat exchanger is used to ensure no liquid CO_2 is sucked into the high pressure compressor rack.

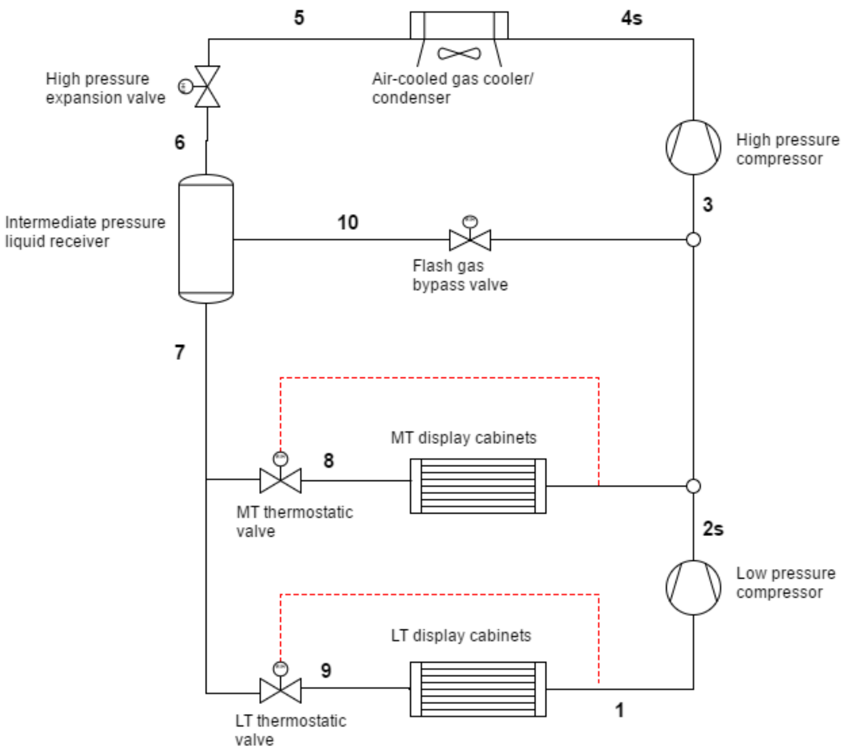

Figure 2.1: A conventional CO_2 booster refrigeration system.

Figure 2.2 below clearly illustrates the trans-critical operation of the high pressure part of the cycle. The figure is based on 90 bar and 10°C outlet temperature from the gas cooler, MT temperature of -5°C, LT temperature of -30°C and a pressure of 40 bar in the intermediate pressure receiver. The cycle also is ideal, meaning no isentropic losses in the compressor or pressure drops in any heat exchangers or pipes.

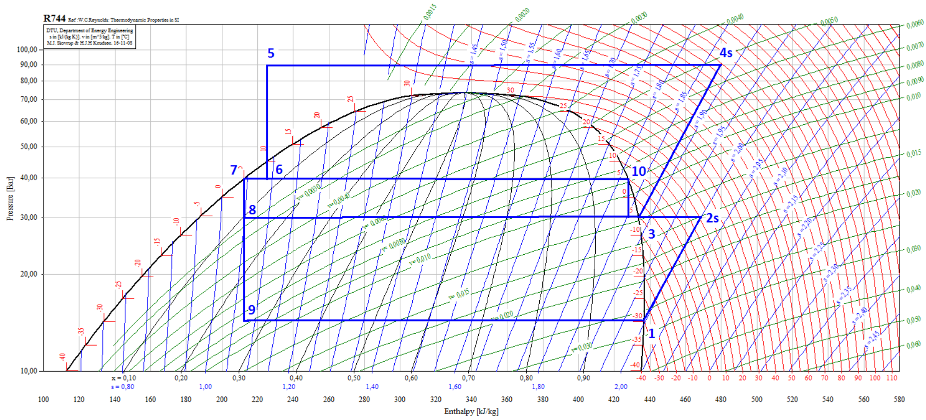

Figure 2.2: Logarithmic pressure-entalphy diagram showing a CO_2 booster system cycle. Figure adapted from Coolpack.

The principle drawing of the system indicates a few important points regarding the cycle efficiency. If one consider state point 5 in figure 2.2, it represents the outlet of the gas cooler. After being cooled from discharge temperature (4s), the CO_2 gas is expanded into the intermediate pressure receiver (6). An important observation here is that the higher the outlet temperature, the higher the vapor fraction after expansion is obtained. This change in the process is associated with moving state point (5) to the right in the LogP-H diagram. The formed vapor is sent through the flash gas by-pass valve and sucked off by the HP compressors. Only the liquid fraction of the refrigerant after the throttling from gas cooler pressure can be fed to the LT and MT evaporators, serving the chilling and freezing duty.This is the reason why a standard CO_2 booster system with an air cooled gas cooler have lower efficiency in warmer climates. Since the ambient temperature is high, the outlet temperature from the gas cooler also will be high. The result is a high vapor fraction after throttling, and a large recirculation rate at the high pressure stage. This again increases compressor power requirement without any increase in evaporator duty, hence lowering the COP of the system.

Different measures has been investigated to reach higher efficiencies for trans-critical CO_2 booster systems also in warmer climates (Hafner et al., 2014; Hafner and Nekså, 2015). Among the different solutions proposed to increase efficiency is mechanical subcooling with a hydrocarbon cycle and ejector-supported parallel compression. These studies shows that energy efficiency of CO_2 systems are comparable to those of standard R404a HFC systems, which represents the majority of supermarket refrigeration systems in Europe. It is shown that the performance of a R744 system applying configurations with ejectors and mechanical subcooling can exceed measured values from a standard R404a system even at ambient temperatures up to 42°C. In cold and mild climates, the R744 system efficiency always outperforms the HFC system (Hafner and Nekså, 2015). For this reason refrigeration systems using CO_2 has become the refrigeration system of choice in Scandinavia when building new supermarkets.

2.1.2 Heat recovery from R744 booster refrigeration system

The purpose of the refrigeration plant is, as described in the previous section, to provide chilling and freezing of the goods in the supermarket. It does so by removing heat at low temperatures and lifting it above the temperature of the heat sink in order to reject the heat. This heat can be recovered in several ways from a trans-critical CO_2 refrigeration system. Heat recovery from refrigeration systems has been subject to several studies in previous years, and a selection is presented below.

The supermarket building itself has a certain demand for heating, varying over the year. Hafner et al. (2012) investigated different control strategies for heat recovery to see what was the most energy efficient option. The study presented two scenarios for heat recovery over 24 hours for a typical Norwegian supermarket. In the first scenario the system was optimized for as high as possible COP for the refrigeration side only. In the second scenario the system was optimized for maximum heat recovery by keeping the gas cooler pressure at 95 bar during hours with high heat demand. In both scenarios an auxiliary heat source was needed to fully cover the demand of the building. The results of the study shows somewhat higher energy efficiency for heating in scenario one when the remaining heat demand was covered by a heat pump (3.7 vs 3.0). The difference was considered negligible between the two scenarios when the remaining demand was covered by electric heaters (1.86 vs 1.87).

Sawalha (2013) reports that heat recovery to a low temperature heating system from a CO_2 booster system is a more energy efficient solution than providing the necessary heat with a separate heat pump unit. The system in this study was employing a de-superheater before the gas cooler. At times with no heating demand the refrigeration system was operated in floating condensing mode, i.e adapting the condensing pressure to the ambient temperature. In heat recovery mode, the system studied was controlled in such a way that the gas cooler pressure was raised to provide sufficient heating energy to the supermarket. The extra operational energy required to lift the pressure in the refrigeration system was lower than typical heat pumps would require to provide the same heating energy. The study concludes that this was the case for nearly all ambient temperatures.

In addition to the presented alternative with a de-superheater, Sawalha (2013) suggests other options that could be implemented for heat recovery from supermarket refrigeration. Figure 2.3 shows four different approaches to heat recovery. In option a, the head pressure is raised to provide the required heat to the HVAC system directly. Option b is a heat pump cascade solution, where the heat from the condenser is upgraded to higher temperature before delivered to the HVAC system. Alternative c is a de-superheater solution, as described in detail above. Option d uses a heat pump as a sub-cooler to recover heat and at the same time lower the inlet temperature to the high pressure expansion valve. The heat from the heat pump evaporator is further upgraded and delivered to the HVAC system of the supermarket. Sawalha (2013) argues that option b, c and d are suitable for R744 systems due to the possibility to operate the system at low condensing pressures.

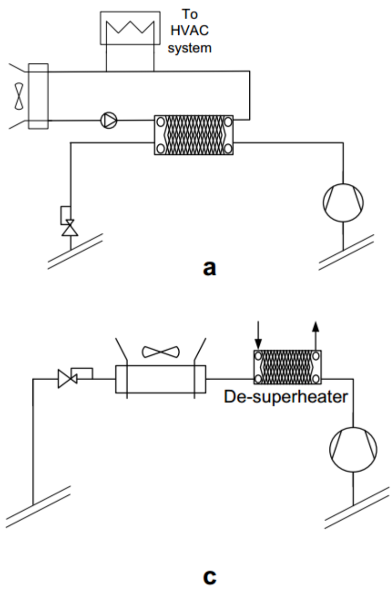

Figure 2.3: Different options for heat recovery from a refrigeration system (Sawalha, 2013).

2.2 Hydrogen refueling station

The transport sector is one of the largest sources of CO_2 emissions today, and is responsible for around one quarter of all greenhouse gas (GHG) emissions in the EU (European Environment Agency EEA, 2014). Hydrogen is regarded by many as the energy carrier of the future alongside with electricity (Rosen and Koohi-Fayegh, 2016). It is argued that energy consumers need a chemical fuel in addition to electricity to satisfy all their needs. Consumers will need a chemical fuel for transportation, as airplanes needs a energy source other than electricity to operate economically (Balat, 2008). As fossil fuel resources are depleting in the world, hydrogen is predicted to become more and more important, and eventually be the dominant chemical fuel. The reason for this is that hydrogen gas can be produced, stored and used when required as opposed to electricity. Battery technology has not come so far today that the available amount of renewable energy produced from wind and solar energy can be stored efficiently.

In order to cut emissions in the transport sector, hydrogen produced from renewable energy sources has received attention as an alternative fuel (Rosen and Koohi-Fayegh, 2016; Balat, 2008). Hydrogen serves as fuel for fuel cell electric vehicles (FCEV), and several automotive manufacturers has developed and are developing FCEV models. Most notable in the automotive industry are the Japanese car manufacturers Toyota, Hyundai and Honda. All three are commercially producing FCEVs today. In table 2.1 below the sales figures for fuel cell vehicles in the US by April 2017 is presented. Similar sales figures are found for the Japanese market, while European sales are lower (Carsalesbase.com, 2017).

Make and model	Sales in US
Toyota Mirai	1523
Honda Clarity FCV	134
Hyundai Tucson/ix-35 FC	116 (2015)

Table 2.1: Cumulative sales of fuel cell vehicles in the US by April 2017 (Carsalesbase.com, 2017).

In order to implement and spread the use of hydrogen as an automotive fuel, it is important that a satisfactory infrastructure exist to serve the increasing fleet of FCEV. A key part of this infrastructure is the network of hydrogen refueling stations to supply the customers with fuel. Refueling stations supplying hydrogen to automotive vehicles can in essence be divided into two categories, namely:

- Stations where the hydrogen produced at a central hydrogen production plant or elsewhere and delivered by truck, train, boat or pipe for storage at the station.
- Stations where the hydrogen is produced on-site, either by a renewable or non-renewable energy source. The produced hydrogen is stored at the station and is ready to serve customers with FCEVs.

Hydrogen fuel cell vehicles has zero tailpipe emissions. When hydrogen gas and oxygen is led to the fuel cell on board the FCEV, the only byproducts are water and heat. However, the indirect emissions of FCEVs are very dependent on how the hydrogen fuel was produced. A study based on the Norwegian energy system reports that for a FCEV to have lower GHG emissions than a electric-gasoline or electric-diesel hybrid cars, the hydrogen must be produced by electricity from renewable energy sources or by fossil fuels employing carbon capture and storage (Svensson et al., 2007). For the hydrogen FCEVs to have a positive effect on the emissions of GHG from the transport sector, it is therefore important to emphasize that the hydrogen must be produced from renewable energy sources. Since the scope of this work is to minimize the import of primary energy to the considered energy system, only the refueling station with on-site production of hydrogen will be considered further. This configuration gives the opportunity for using locally produced electricity from photovoltaic panels to produce some of the hydrogen gas in addition to power from the grid.

2.2.1 Hydrogen refueling station with on-site production

The most common production method for hydrogen from renewable energy sources today is by electrolysis of water. The process yields high purity hydrogen (higher than 99

% (Alazemi and Andrews, 2015)) and no impurities, and can therefore be used directly in fuel cell powered cars. The efficiency and operation of fuel cells are very sensitive to impurities in the fuel gas, and especially from carbon monoxide (CO) (Baschuk and Li, 2001). The presence of CO gas in the fuel can lead to a condition known as CO poisoning of the fuel cell, and thereby lower its efficiency.

In the present energy situation, electrolysis of water has the highest cost per kg hydrogen of the commercially available production methods, including the non-renewable methods (Alazemi and Andrews, 2015). However, it is still the cleanest process to produce hydrogen and the system can be scaled to a variety of sizes, which makes it very flexible. Electrolysis of water requires electrical energy as input and the fuelling station can either be fully connected to the grid, powered by a renewable energy source on-site or a combination of the two. If the refueling station have access to electrical energy from a source on site, it can either be powered by photovoltaic panels or by wind turbines. However, more than 95% of the total production capacity of hydrogen gas world wide is today based on fossil fuels. These production methods include steam reforming of natural gas and gasification of coal (Kalamaras and Efstathiou, 2013). Most of the hydrogen gas produced today is used in ammonia and fertilizer production.

Since the motivation of this thesis is to minimize energy consumption and thereby reduce the emissions of greenhouse gases, only refueling stations supplying hydrogen produced by electrolysis is considered further. This is due to the high share of renewable electricity in the Norwegian power grid. A hydrogen refueling station with on-site production consists of a number of components. The important components and their function required for this type of refueling station is listed in table 2.2 below.

Component	**Function**
Energy source	Provide electric energy for electrolyser
Electrolyser	Produce low pressure H_2 from water and electricty
Compression system	Increase H_2 pressure to high
Storage tanks	Store H_2 at different pressure levels
Cooling system	Keep temperature of H_2 at required level
Fuel dispenser	Deliver high pressure H_2 to fuel cell vehicle

Table 2.2: Description of components in a H_2 refueling staion with on-site production.

A simple illustration of the fundamental components in a hydrogen filling station and their interactions is shown in figure 2.4 below.

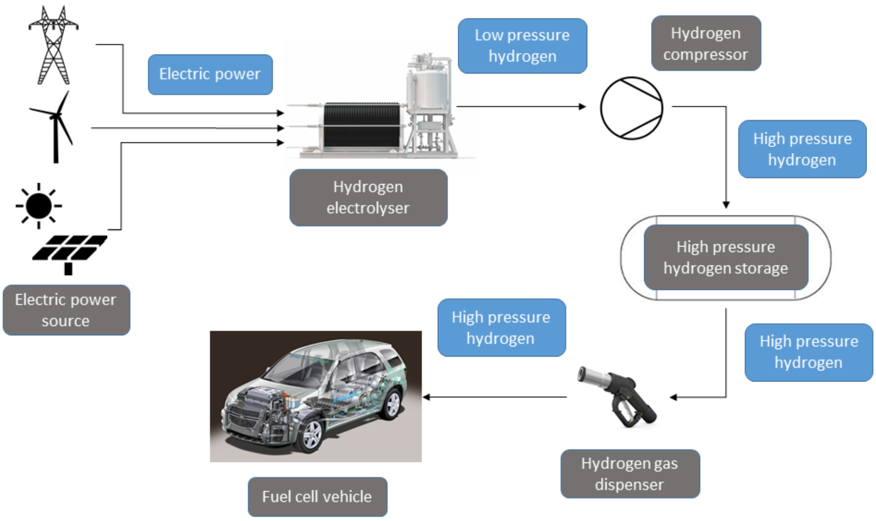

Figure 2.4: Main components of a hydrogen refuelling station with on-site production.

2.2.2 Hydrogen fuelling, heat recovery and cooling demands

To increase the popularity of fuel cell vehicles and the use of hydrogen as an automotive fuel the driving ranges, refueling times and safety must be competitive to the existing technology for gasoline and diesel. Filling a full tank on a gasoline or diesel car takes only a couple of minutes, fossil fuel filling stations are readily available and the infrastructure is well established. Fuelling of hydrogen is a more complicated process due to the properties of hydrogen. Firstly, hydrogen needs to be stored at high pressures. This is due to the very low density of hydrogen at normal conditions, 0.08994 kg/m^3 (United States Department of Energy, 2016) (1 atm, 0°C). For economic reasons and to increase the energy density of the gas, FCEV are equipped with tanks that store hydrogen at either 350 or 700 bar in order to obtain the driving ranges comparable to gasoline or diesel cars. For hydrogen to be stored at liquid conditions, the temperature needs to be kept below -253°C. This is one of the main reasons why hydrogen needs to be stored as a high pressure gas for automotive applications.

To ensure safe fuelling of fuel cell vehicles and within acceptable filling times the Society of Automotive Engineers has written a technical information report, called ”Fueling of light duty gaseous hydrogen vehicles”. The report includes fueling protocols and safety requirements for automotive hydrogen fuelling stations (Society of Automotive Engineers, 2010). The different classifications of protocols are based on the pressure involved (350 or 700 bar) and the level of precooling of the hydrogen gas during dispensing. The temperature at dispensing ranges from ambient temperature (no cooling), 0°C, -20°C down to a precooled temperature of -40°C. Stations built today aim to satisfy the strictest classification A70, meaning 700 bar storage and precooling of -40°C to enable fast filling of the fuel cell vehicle (Rothuizen et al., 2013a). The target density for the vehicle tank is 40.2 kg/m^3 after refueling for a 700 bar system. The classification of hydrogen refueling stations are summarized in table 2.3.

Type	**Pressure [bar]**	**Temperature [°C]**
A70	700	-40
A35	350	-40
B70	700	-20
B35	350	-20
C35	350	0
D35	350	Ambient

Table 2.3: Classification of hydrogen fueling stations. Table adopted from (Rothuizen et al., 2013a).

The precooling of the hydrogen is needed because the hydrogen gas heats up during expansion during the filling process. Heating of the tank on-board the fuel cell vehicle is regarded as a safety issue, and the temperature limitations of the composite tanks are 85°C (Richardson et al., 2015). Filling a vehicle only by using a high pressure tank and a pressure reducing valve to supply hydrogen into the vehicle would take very long time. The mass flow rate of hydrogen had to be very low to avoid exceeding the temperature limitations (Rothuizen et al., 2013b).

Rothuizen et al. (2013b) provides a description of how automotive hydrogen fuelling station complying with SAE J2601 are designed and operated, both for dispensing and storage. A brief summary of this follows.

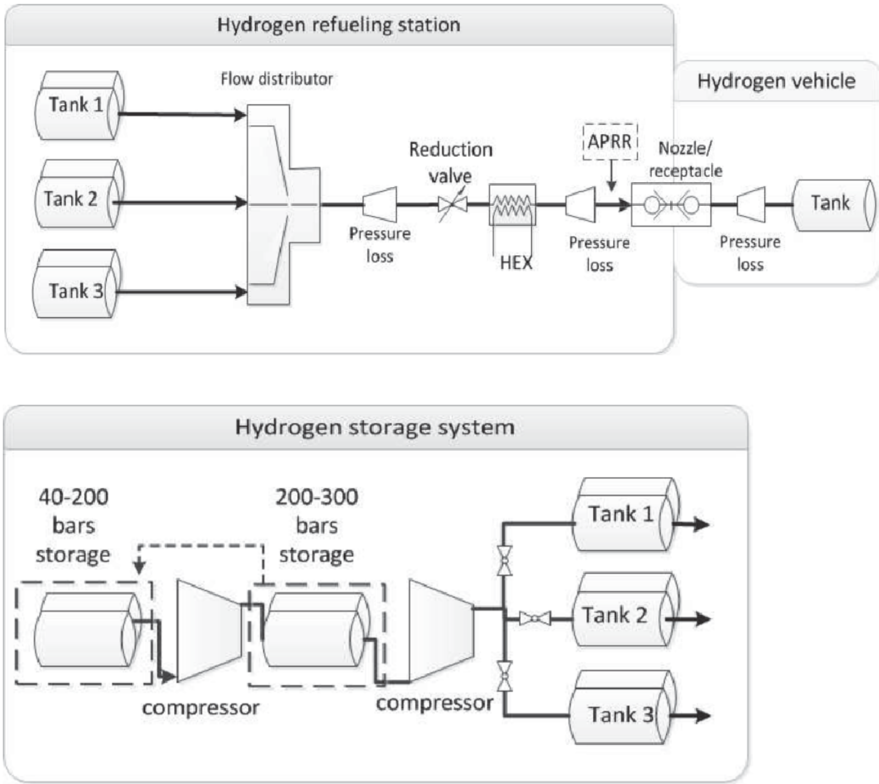

Figure 2.5: Layout and components in a hydrogen refueling station. Figure adapted from (Rothuizen et al., 2013b).

Hydrogen gas is stored in several tanks at different pressure levels at the hydrogen station. Usually the hydrogen is stored at a cascade system, shown as tank 1, 2 and 3 in figure 2.5. The pressures of the tanks vary by manufacturer, but is normally between 400-600 bars for tank 1, 600-800 bars for tank 2 and 900-1000 bars for tank 3. Fuelling of the vehicle is done by levelling the pressure between the tank on board the vehicle and the high pressure tanks at the station. The filling starts with the lowest pressure (tank 1) and is switched to the next tank in the cascade when the pressure difference becomes too low to obtain the required pressure ramp in the vehicle tank. The large overpressure tank 3 is required to ensure that the vehicle reaches 700 bar and sufficient hydrogen according to the filling protocol in SAE J2601. The average pressure ramp is controlled by the pressure reduction valve. The precooling heat exchanger is placed after the reduction valve because the hydrogen heats up when it is throttled, as mentioned above.

The way the hydrogen is stored at the station is somewhat dependent on how the hydrogen gas is supplied. Hydrogen is stored in steel cylinders of about 200-300 bars at stations that are supplied with hydrogen externally by trucks or other means. The tanks can either be prefilled with compressed hydrogen and delivered as a unit, or be stationary at the station and filled by the truck externally. For stations equipped with electrolyser for production of hydrogen on-site, the outlet low pressure hydrogen gas is first stored at 40-200 bars, depending on the working pressure of the electrolyser. Then the pressure is raised to intermediate storage pressure of 200-300 bars. As the pressure drops in tanks 1-3 during filling of vehicles, the hydrogen stored at 200-300 bars is compressed and delivered at the tanks to recharge them to the correct pressure level. After compressing the hydrogen from 200-300 bars, there is also need for cooling before delivering the hydrogen to tanks 1-3.

Other variants of the system solutions for hydrogen stations has been suggested. One system solution is described in (Richardson et al., 2015), where hydrogen is delivered as liquid at -246°C. The whole refueling station is designed to fit inside a standard 40 foot shipping container. The dispensing and storage system in this desing is a combination between mechanical and cryo-compression. The overview of the station and components are shown in figure 2.6 below.

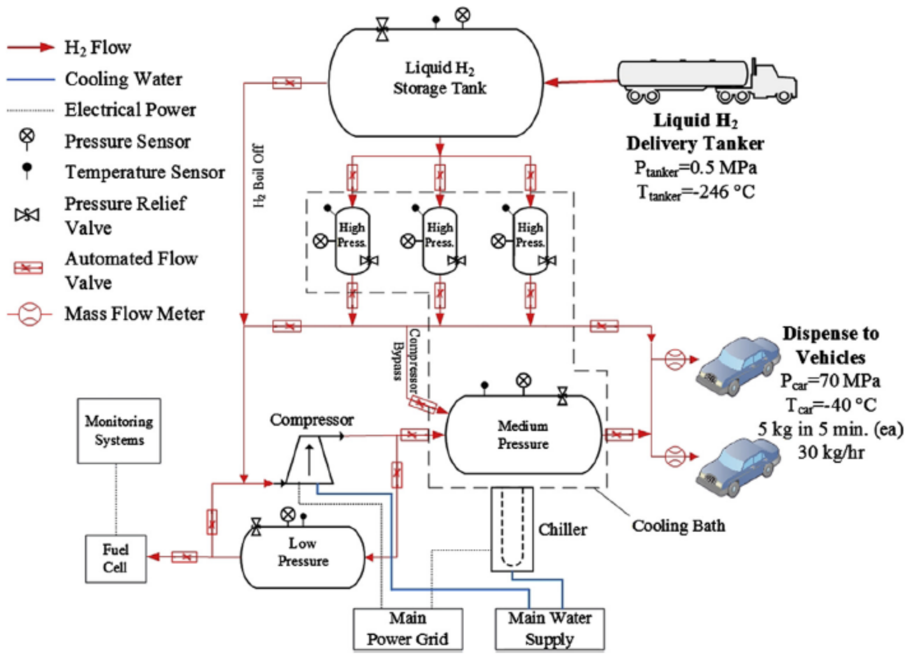

Figure 2.6: Alternative layout of a hydrogen station with liquid hydrogen storage. Figure adapted from (Richardson et al., 2015).

Mechanical compression will supply pressures up to 410 bars (Medium pressure) and the high pressure tanks of 1120 bar will be supplied by cryo-compression. After each fuelling of a fuel cell vehicle the high pressure tank will be at 700 bar. The tank is evacuated and filled with low temperature liquid hydrogen, the closed. The tank is heated up to -40°C and the hydrogen inside the tank boils, creating the required pressure. The key in this design is that both the medium pressure and high pressure tanks are kept in a cooling bath of -40°C. The authors argue that by keeping the high pressure and medium pressure at low temperature both lowers the refueling time and makes the system simpler. The expensive in-line refrigeration system required if the hydrogen is compressed on demand or kept at high pressure at ambient temperature is avoided. The precooling bath is kept at -40°C by an industrial chiller.

An alternative solution is to compress the hydrogen in stages intercooling, featuring a cascade refrigeration system. Compressing the hydrogen to final pressure on demand will also require an in-line heat exchanger to keep the hydrogen gas temperature at -40°C during filling. Instead of releasing the condensing heat from the refrigeration system to the ambient air, the waste heat could be utilized elsewhere. An alternative for waste heat management was described by Meratizaman et al. (2014). The authors carried out an economic and environmental evaluation of a hydrogen refueling station, where the hydrogen gas is compressed on demand in three steps up to 350 bar, and water cooled to ambient temperature after each compression step. The hot water could then be accumulated at 55^{o}C in a water tank and used as a utility, for example for a car wash. The amount of hot water which can be produced from a system like this will depend on throughput of hydrogen gas in the system.

2.3 Energy systems for high performance buildings

Residential and commercial buildings are responsible for a large share of the total energy consumption worldwide. In the EU, buildings represent 40% of the total energy consumption (European Union, 2010). In order to reduce the energy demand of buildings in the Union, a directive on energy performance in buildings obligate all member states to ensure that all new houses built after 2022 is nearly-zero buildings (European Union, 2010). According to the EU nearly zero-energy buildings are buildings with very high energy performance. The zero or low amount of energy required by the building should be covered by energy from renewable sources. The energy should be produced on-site or nearby to the building location. Figure 2.7 shows the Kyoto pyramid for passive energy design of buildings. The pyramid illustrates the approach when designing for low energy and high performance buildings.

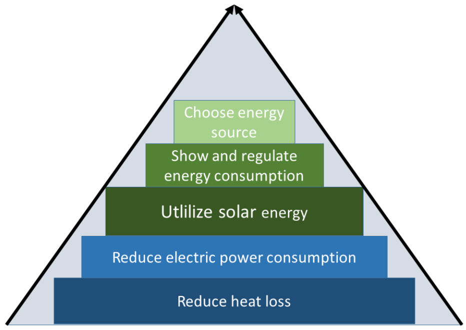

Figure 2.7: Kyoto pyramid for passive energy design of buildings. Adopted and translated from (Dokka and Andresen, 2012).

As the figure above illustrates, starting by reducing the heat loss of the building is the first step towards low energy houses. In Norway two standards has been commissioned that apply for low energy and passive houses, one for residential and one for commercial buildings (Standard Norge, 2013a,b). The standards set strict requirements for heat losses through walls, roofs, windows and doors, among many other requirements.

As requirements for heat losses for both residential and commercial buildings are getting stricter, there is a shift from space heating to domestic hot water (DHW) as the dominating energy demand. For residential buildings following the Norwegian passive house standard, the maximum allowed specific space heating demand is 15 kWh/m^2 for buildings over 250 m^2 in Oslo climate zone (Standard Norge, 2013a). The shift from dominating space heating demand to DHW has been observed over some time as new norms for building quality is introduced. Figure 2.8 shows the development in electricity consumption to different sources for Norwegian buildings following different building standards. While the demand for space heating is decreasing, the energy demand for DHW is fairly stable. As a consequence, DHW now represent the dominating load on the heating system for a Norwegian residential passive house. In the figure below it seems that the energy requirement for DHW is reduced between Low-energy houses and passive houses. The change in the figure is because NS3700 requires minimum 50% of the energy to DHW to come from a renewable energy source.

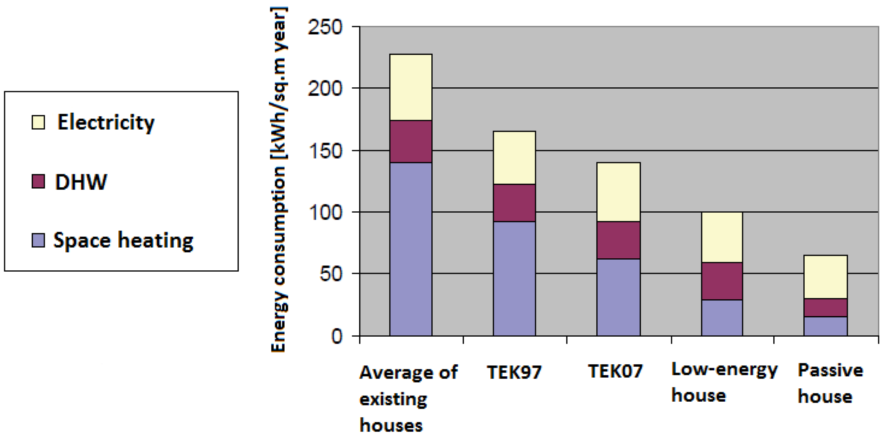

Figure 2.8: Change in energy demand for Norwegian buildings with stricter norms. Translated from (Andresen, 2008).

2.3.1 Heat Pumps

As described in the previous section, high performance buildings have a lower demand for space heating due to a highly insulated building structure. This must be taken into consideration when designing the heating system for the building. As a result, efficient systems for domestic hot water must be taken into consideration for modern buildings, since the need for space heating is decreasing.

Trans-critical CO_2 heat pumps for tap water heating has received attention because high COP and because of the environmental characteristics of CO_2. Nekså et al. (1998) investigated the performance of a CO_2 water heater already in 1998, showing the large potential of the technology in both residential and industrial sector. Among the findings of the study was possibility of supplying hot water at 90°C without operational problems, good temperature fit between cooling CO_2 and heated water, high COP and compact design due to high volumetric heating rate. The study concludes the area of application for CO_2 hot water heat pump is much larger than for traditional heat pumps where the water temperature often is restricted to 55°C (Nekså et al., 1998). Running the heat pump with 0°C evaporation and 60°C outlet water temperature resulted in a seasonal COP of 4, saving 75% of the primary energy consumption compared to electrical or gas fired systems. The optimum gas cooler pressure was reported to change with changing boundary conditions, such as water inlet temperature and hot water exit temperature.

Stene (2005) suggests using a residential CO_2 heat pump in combined operation with three gas coolers in series to supply heat at three different temperature levels. The discharge gas enters the first gas cooler, where tap water is reheated to 70°C. Then the CO_2

enters the middle gas cooler where heat is rejected to a low temperature hydronic space heating system, with a typical temperature drop of 5°C. Finally the CO_2 enters the last gas cooler where tap water is preheated before it is reheated in the first gas cooler, as described. A schematic of the systems shown in figure 2.9.

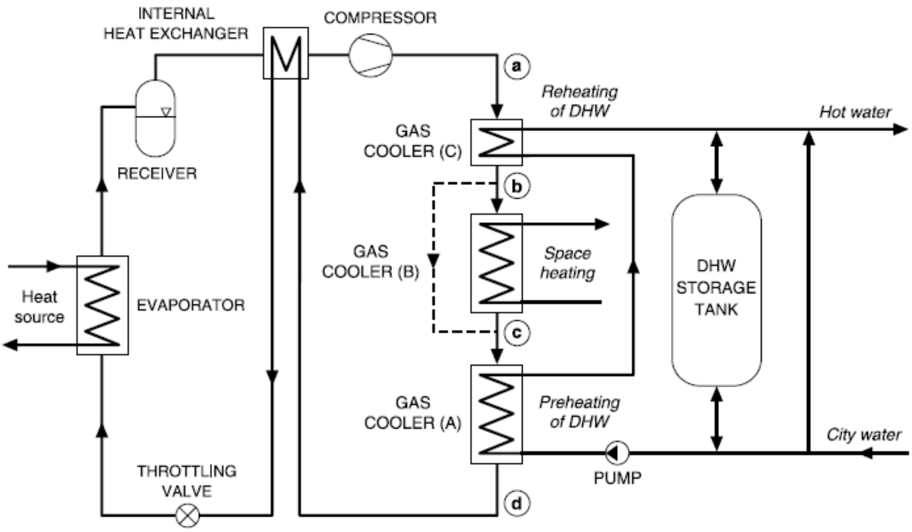

Figure 2.9: Schematic of residential CO_2 with three gas coolers, taken from (Stene, 2005).

The study reports that the prototype CO_2 heat pump achieved the same or higher seasonal performance factor (SPF) than the most efficient brine-to-water heat pumps as long as the fraction of DHW production was higher than 25%, return temperature of space heating system was lower than 30°C and city water temperature lower than 10°C (Stene, 2005). This proves that a CO_2 heat pump fits well in the new generation of highly insulated buildings, due to the dominating demand for DHW. However, it is critical that a low temperature space heating system can be implemented. This will ensure a high COP for the system in combined operation.

For a heat pump unit to reach a high seasonal performance factor, a heat source with stable temperature throughout the year is beneficial. A smaller temperature difference between heat sink and heat source provides good operational conditions for the heat pump compressor, as the required pressure ratio decreases. The heat source used in heat pump systems depends on a number of factors, such as the location of the building, availability of the different sources, investment costs and operational costs (Stene and Havellen, 2016). Typical heat sources for larger heat pump installations are seawater, ground water, ground/soil, bedrock and outside air.

Outside air is readily available, but the temperature is subject to a lot of variation throughout the year. Another aspect of this variation is during the time of the year with highest demand for space heating, the temperature of the air is at its lowest. Normally air-source heat pumps are switched off at ambient temperatures below -10°C, but for heat pumps equipped with compressor cooling this can be extended to -20°C. Bedrock can be utilized by boring deep vertical energy wells, often in the range of 200-300 meters deep. This heat source has a more stable temperature during the year, but with heat imbalance the temperature can change over seasons. Boreholes gives the opportunity to employ "free cooling" by direct cooling of ventilation air with the circulating brine in the boreholes. Energy wells will be treated more in detail under the section for thermal storage. Seawater, lake water and ground water will have similar qualities and characteristics as bedrock, but is of course dependent on the geographical location of the building. Common for the water-based heat sources is a more stable temperature during the year (Stene and Havellen, 2016).

Hydrocarbons has received attention as sustainable refrigerants for both refrigeration purposes and in heat pumps. Hydrocarbons includes isobutane R600a, propane R290 and propene R1270 to mention a few. Granryd (2001) reports that in general, hydrocarbons offers a sustainable, energy efficient and environmentally friendly alternative for both heat pumping and refrigeration purposes. The main concern of the hydrocarbon refrigerants are the flammability, and the author emphazises that this risk must be taken seriously. Although hydrocarbons are flammable, a variety of applications for hydrocarbons is already established. Isobutane as refrigerant for domestic refrigeration is well-known and dominates the market in Europe. Propane has also seen some applications in small heat pumps, and exhaust air heat pumps for single family houses has been a large sale success in Sweden.

Studies show that the performance of hydrocarbons can match or outperform similar HFC systems. Palm (2008) investigates the performance of small hydrocarbon heat pumps and refrigeration systems less that 20 kW, and makes a comparison between HFCs, hydrocarbons and ammonia (R717). The author reports that the expected system efficiency of hydrocarbon heat pumps and refrigeration systems will be close, or higher than R22 and R134a systems. One of the reasons are the higher volumetric refrigeration effect of the hydrocarbons compared to HFCs. The main concern for use of hydrocarbons is also here the flammability of the gases, and the author points out that careful engineering and manufacturing is of great importance to avoid leaks and accidents. Suggested actions to reduce the charge is use indirect systems, compact heat exchangers and in general to use alarms, forced ventilation or placing the unit outdoor.

A thermodynamic performance study of propane as substitute of R22 in refrigeration systems was carried out by Choudhari and Sapali (2017). Required mass flow of refrigerant and discharge temperature where found to be lower for the R290 system compared to R22. These parameters indicate longer lifetime for the compressor. However, volumetric refrigeration capacity and COP was found to be slightly lower for propane. But the author argues that R290 is expected to outperform R22 in a real life system designed according to the properties of propane. In overall it is concluded that R290 is a good substitute for R22 because of its excellent environmental and thermo-physical properties.

2.3.2 Integration of solar energy

Using the energy provided by the sun is considered as step three in passive energy design of buildings, see figure 2.7. The free energy provided by the sun can be used by correct design of the building and by placing windows and facades strategically (Dokka and Andresen, 2012). This is regarded as passive use of the solar energy, but the energy from the sun can also be exploited actively by use of solar thermal collectors and solar photovoltaic panels.

Solar thermal systems in buildings has received attention as a sustainable alternative due to increasing demand for reduction in fossil fuel and electricity consumption. A study for an office building in Montreal, Canada concludes that a combination between a ground-source heat pump and direct solar heating heating by thermal collectors obtains the best overall balance between the use of solar energy, capital costs and energy efficiency (Tamasauskas et al., 2014). The studied system achieved an annual solar fraction of 0.25 and energy saving of 76% compared to the base case of fuel fired boiler and electric driven air condition system for cooling.

2.4 Thermal Storage

In this section different solutions for thermal storage is studied in more detail. Thermal storage is an interesting feature because it makes it possible to recover heat from the energy system when it is available, and then store it for a time when the heat is needed. In that sense, thermal storage offsets any mismatch between heat supply and heat demand. The technologies for thermal storage has different time scales for which the heat can be stored. It varies from a day-night perspective to possibility for seasonal storage. The three technologies of thermal storage which is considered further in this section are water tanks, phase-changing materials (PCM) and energy wells/boreholes.

2.4.1 Phase-changing materials

The use of PCM is an interesting option for thermal storage. PCMs store energy in form of latent heat, compared to sensible heat by storing hot water in tanks. The energy released for such a system is the sum of the energy due to temperature change and phase change. Thus the PCM material has a high energy density. For most PCM, the phase change occurs at a constant temperature, making it suitable for holding its surroundings at a stable condition. The temperature for phase change for different PCMs varies over a large span, and includes almost every temperature. In other words, there is a suitable PCM for almost every temperature level available.

Studies has shown that placing PCM inside the refrigerated cabinets in a supermarket has several benefits. Lu and Tassou (2013) reports that water based PCMs with nucleate agents are good candidates for refrigerated cabinets. One of the benefits of integrating PCMs into the cabinets is that the PCM can help reduce the peak loads for cooling. At periods of low cooling loads the refrigeration system can operate at full load to freeze the PCM material, and at peak cooling demands the PCM can retain the cold for a given amount of time by melting and absorbing heat from the chilled goods, even at times of insufficient refrigeration capacity. In this way the peak cooling loads are shaved off, allowing for a lower capacity refrigeration system to be installed. An illustrative example of a loading process for a PCM thermal storage is shown in figure 2.10. The inner tube can be the evaporator tube in a refrigerated cabinet of a supermarket.

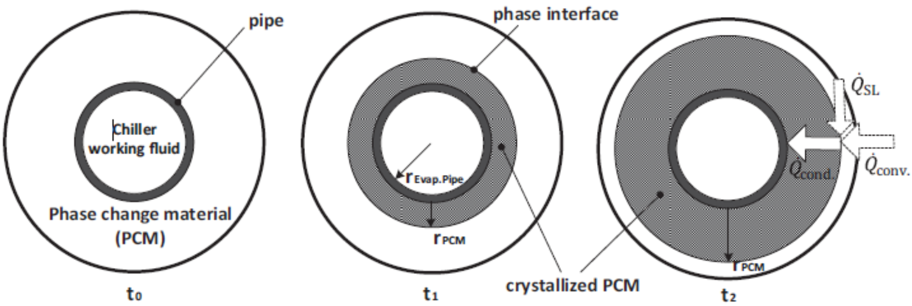

Figure 2.10: Schematic of a loading process of a PCM thermal storage (Beck et al., 2016).

The potential of using cold thermal storage for storing of excess renewable energy has been suggested as an interesting option for supermarkets, which gives the opportunity to use the renewable energy when it is available. The principle has been studied also for domestic purposes, where excess solar energy is stored in a low temperature thermal storage connected to a domestic fridge and a freezer (Beck et al., 2016).

2.4.2 Energy wells

Compared to the two other solutions for thermal storage considered in this section, energy wells has a longer time perspective of storing the heat. An energy well is a collection of several boreholes in the ground ranging from 50-250 meters in depth. In each borehole there is normally two tubes filled with an antifreeze solution that circulates and exchanges heat with the surrounding bedrock. The reason for using deep boreholes in building projects is that the temperature some meters below the surface is very stable over the year. The boreholes are used as a heat source during the winter, and depending on the amount of heat taken from the wells, the temperature of the well will decrease. But during summertime the wells are used as a heat sink, where heat from the refrigeration system or ventilation cooling can be rejected to the boreholes. In this process the energy wells are being heated up again, or in other words, recharged. This temperature swing from season to season is why energy wells are denoted as seasonal thermal storage.

Chapter 3

System Design and Simulation Model

In this chapter the selected design for the system is presented together with the building simulation model. In section 3.1 the building envelope and boundary conditions for the building simulation model is presented. Then the fundamental concept of the energy system is presented followed by an in-depth review of each part of the system. The review of each part of the energy system includes background for choices made as well as the representation of the system in the simulation software Dymola. The selected design of the building and energy system emphasizes the concept of minimizing import of primary energy to the building plot as well as integrating the different parts of the energy subsystems together in order to maximize heat recovery.

3.1 Building simulation model

In order to obtain heating and cooling demands for a full year for the building, a simulation model of the complete building have to be established. To be able to investigate the dynamics of a integrated energy system, an hour by hour demand for heating, cooling and domestic hot water have to be determined. In this way one can see how much of the waste heat that can be utilized to satisfy the heating demands in the building. The key factor is the time scale on which heat is available, and at which time it is needed. This relationship between supply and demand is what is interesting to investigate, and the demand side of the problem can be determined by a building simulation model. The software chosen for the task is a Norwegian building simulation software called SIMIEN. This software is used for calculating energy demands, heating power demand at peak loads, and thermal comfort of buildings. The software also features evaluation against different Norwegian building standards, such as TEK07/TEK10 and the standards for low energy buildings/passive house buildings. The software uses input data for the building geometry and structure, as well as climate data for the location considered, to do an hour-by-hour energy simulation over a whole year. The software gives options for choosing different building materials as well as making evaluation of retrofit solutions to an existing building, including cost estimates and payback times.

A separate model was established in SIMIEN for the apartments and the supermarket. The reason for this is that the default values for the simulation is different for non-residential buildings and residential buildings. Examples include schedules for ventilation and values for internal loads and standard air supply rates. The building concept considered in this thesis is highly motivated by minimizing the import of primary energy to the building plot. One way of achieving this is by having very energy efficient buildings that use as little energy as possible. Passive houses and zero-emission buildings are the most energy efficient buildings existing today and they have strict specification on the energy used for space heating, electronics as well as domestic hot water. For this reason the requirements and specifications from the Norwegian passive house standard for residential buildings NS3700 (Standard Norge, 2013a) and non-residential buildings NS3701 (Standard Norge, 2013b) are chosen as input data for the simulation software. One can argue that the easiest and cheapest way of saving energy is by not using it at all. ”The cheapest kWh is the one that you do not use” is a saying that fits well for this project. Passive houses have a super insulated building envelope to minimize the heat loss through the structure. The input data to be used for the building envelope in SIMIEN is summarized in table 3.1 below.

Characteristic	**Value [W/m^{2}K]**
U-value windows and doors	0.80
U-value roof	0.09
U-value floor	0.08
U-value outer walls	0.12
Normalized thermal bridge value	0.03

Table 3.1: Required values for the construction for residential and non-residential passive houses (Standard Norge, 2013a,b).

Together with the information on U-values required for the building envelope, the dimensions and physical appearance of the building are needed in order for the simulation software to calculate heating and cooling demands. Since the actual building structure is yet to be planned, the building plan is subject to freedom of choice for this project. The chosen layout of the building is based on a plausible design for a combined supermarket and block of flats. It is assumed that the supermarket occupies the whole first floor, then three more floors with apartments on top to make it in total 4 floors. The base of the building is assumed to be 30 meters by 35 meters, making the area of the supermarket 1050 m^2. The design and chosen layout of the building is illustrating an example of how a building of this type can be arranged. Whenever data on the actual building is available, adjustments on the building model in SIMIEN can be made to get more accurate results for that specific case.

The design of the supermarket is based on a typical Norwegian supermarket without a kitchen for preparing fresh fish or meat. The supermarket consists of a sales area for fruit and vegetables, general product shelves, general and cold storage room as well as an office for the employees. There is a loading ramp connected to the storage room for receiving new supplies to the store. The supermarket is modelled as one thermal zone with uniform temperature setpoints, including the office and general storage. However, the cold room and cold display cabinets are excluded from this thermal zone as they are designed to keep the goods at low temperatures. A simple illustration of the supermarket floor plan is shown in figure 3.1 below.

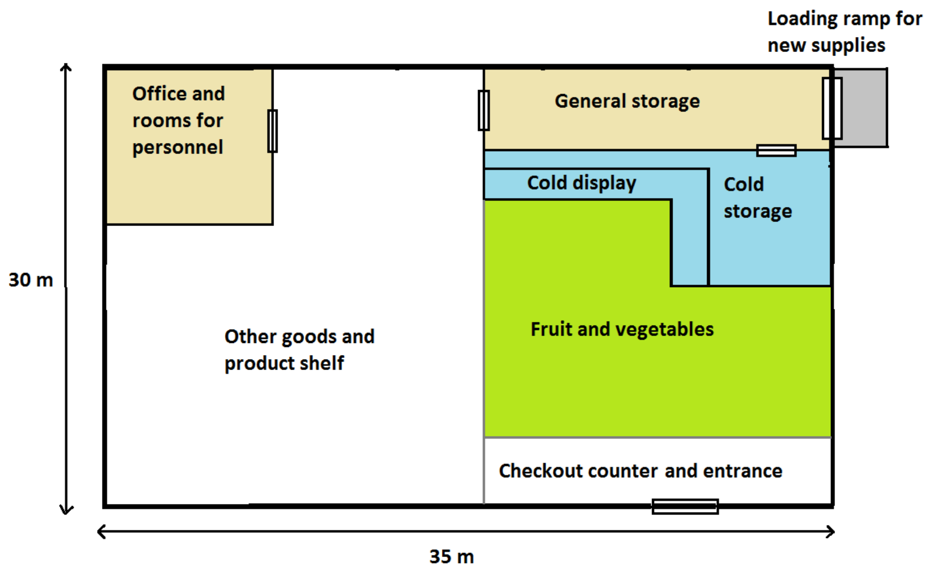

Figure 3.1: Simple illustration of the supermarket layout.

For the apartments on the upper floors the design is chosen such that each of the three upper floors are divided into ten equally large apartments, making it 30 apartments in total. On each floor there is a set of five apartments on the north side of the building and a set of five apartments on the south side of the building, where a hallway in the middle connects the apartments. Each apartment is 100 m^2, leaving approximately $50m^2$ for the hallway. It is assumed that the staircase and/or elevator is on the outside of the building, as the supermarket occupies the whole first floor. Each apartment has two windows on the end of the apartment, so that each floor has ten windows facing north and ten windows facing south. The internal design of the building do not have a large influence on the heating demand, as also the apartments are linked together as one thermal zone, including from one floor to the other. The internal design of the building has influence on the thermal inertia of the building only, which depends on the heaviness of the construction material used in the internal structure. More accurate description of the internals of the building is therefore not considered further in this thesis. A simple illustration of one floor of apartments is shown in figure 3.2 below.

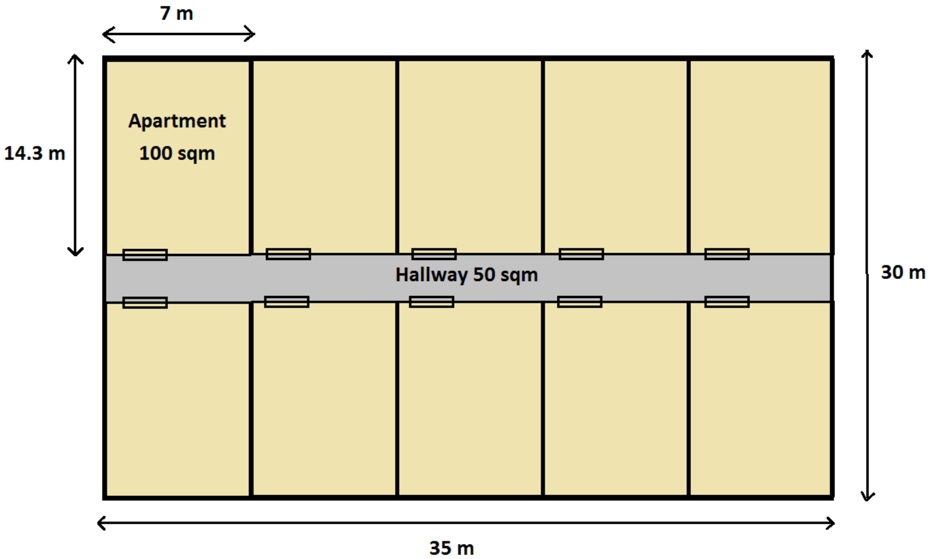

Figure 3.2: Simple illustration of one floor of the apartments.

When relevant information on the actual building is available, more detailed and accurate energy calculations can be made. For this thesis, the assumptions made here will form the basis for the heating and cooling demand of the building. Relevant information on the building is summarized in table 3.2

Characteristic	**Value**	**Unit**
Building base	1050	m^2
Number of floors	4	-
Supermarket area	1050	m^2
Total area apartments	3150	m^2
Number of apartments	30	-
Location of building	Tromsø	-

Table 3.2: Building characteristics for input to SIMIEN.

As previously stated, the internal design of the building will not affect the energy consumption to a large extent, but the size of the base of building will certainly do. The chosen design is based on similar sizes for the same type of supermarkets in Norway. The building design and layout is chosen to investigate the feasibility of heat recovery and integration of the energy system. The chosen design and layout of the building illustrates an example on how much of the energy demand that can be satisfied by heat recovery. Since neither the building nor the size of refrigeration system is determined in detail yet, it must be emphasized that this paper illustrates an example of such a system. However, scaling the size will most likely affect only the value of the energy demand, and the dynamic behavior of the system is predicted to remain unchanged.

3.2 Energy system design and simulation models

In this section the fundamental layout of the energy system is presented. First the principle solution of the energy system will be presented, followed by a more detailed description of each component in the system together with the representation of the component in the simulation software. The background and explanation for the choice of system components will be given for each subsystem.

Multiple options for heat recovery to the building energy system exists on the property. In order to evaluate the potential for energy savings for the different options, three separate models in the simulation software Dymola was developed. The three models represents the level of integration that can be achieved in the building energy system. The first model and study includes heat recovery from the supermarket refrigeration system. Case study two and three investigates how waste heat from the hydrogen refueling station can be recovered in two different ways in addition to heat recovery from the supermarket. How heat recovery and integration is fulfilled in each case is described in the following sections along with boundary conditions for the models. First a short introduction to the simulation software Dymola is given along with some general simplifications and assumptions made for the simulations.

3.2.1 Dymola and the Modelica language

Dymola is a dynamic simulation software (DYnamic MOdelling LAnguage) that today serves as an user-interface for the object-oriented modelling language Modelica. Modelica is used for modelling physical systems, and is able to combine components from many disciplines. For this thesis thermodynamic properties of refrigerants, liquids and gases are needed in order to simulate the models. There are several thermodynamic property libraries available for use with the Dymola software, and the TIL Media package from TLK-Thermo GmbH is chosen. The package contains thermodynamic properties of the most common refrigerants, gases and liquids. In addition, the component library TIL 3.4 from the same company is used for building the models. The library features pre-modelled components for thermal systems including compressors, heat exchangers, valves, sensors, controllers and other. Boundary conditions and operational parameters must be specified into the components, along with the connections between them. Most of the components in the three models are from the TIL library, with some logical functions from the standard Modelica library.

A table of the different colored lines and their representation in the TIL package is given in table 3.3 below.

Characteristic	Color
Refrigerants (phase changing fluids)	Green
Liquids	Blue
Gases	Orange
Signals	Dark blue
Logic signals	Pink

Table 3.3: The different lines in the Dymola model and their representation.

These are the default colors for the components in the TIL library. During the presentation of the models through this section, the colors and their representations are as described in table 3.3.

3.2.2 General assumptions and simplifications in Dymola

Modeling the complete energy system for the building is a challenging and complex task, as the number of components are large. For this reason it was found necessary to make some simplifications to the simulation models in order to reduce the simulation time and complexity. The focus of the work is to investigate the dynamic behaviour of the energy system, and see how waste heat availability and heat demand affect each other and the energy coverage in the building. Since the focus is the influence of supply and demand of heat on each other, the assumptions and simplifications are accepted. Possibly the largest simplification is neglecting pressure drops in pipes, junctions and heat exchangers. Secondly, a constant heat transfer coefficient is assumed for each type of fluid in each type of heat exchanger. Table 3.4 below presents the heat transfer coefficients on each side of the heat exchangers used in the models. Other general simplifications and assumptions are no heat losses from the components in the system and a constant compressor isentropic efficiency of 70%.

Fluid	Plate [W/m^{2}K]	Fin and tube [W/m^{2}K]
Refrigerant side: phase change	2500	2500
Refrigerant side: gas cooling	2000	2000
Liquid side	1000	600
Gas side	-	150

Table 3.4: Heat transfer coefficients for different fluids in the heat exchangers used in the models.

Due to the number of components, varying loads and complexity of the model, the assumptions helps shortening simulation time. Although the demand for space heating and domestic hot water is found for a whole year by building simulation in SIMIEN, a full year simulation of the energy system is not suitable for the models developed in Dymola. It is instead chosen to select four weeks from the building simulation results. The intention is that these four week should represents the whole year, and therefore it is chosen one week in winter, spring, summer and fall. Each of these weeks represent their respective season with their characteristic heating demand. In this way, the performance of the integrated energy system can be investigated with different load and boundary conditions. The results of the full year building simulation as well as heating demands for the four selected weeks are presented in section 4.1.

3.2.3 Energy circuits and principle solution

The energy system of the building is arranged in three different temperature circuits. It has a high temperature (HT) water circuit ranging from 5-70°C, providing the apartments and supermarket with domestic hot water. A low temperature space heating system is completed with a medium temperature (MT) water circuit with supply and return temperature of 33/28°C, respectively. Finally, a low temperature (LT) antifreeze circuit ranging from from 0-15°C is used as cooling circuit for ventilation and other purposes. The antifreeze circuit is connected to boreholes to enable seasonal heat storage and have a stable temperature source for an auxiliary heat pump. The key feature of this three-level arrangement is that the available waste heat at different temperatures can be utilized. The relative low temperature of the space heating system enable use of low grade waste heat (40-60°C) to space heating directly. Another key feature is having storage tanks for water at both 70 °C and 33°C. The HT circuit have storage tanks with hot water at 70°C available, which acts as a buffer for storing the DHW at times with low demand. This enables hot water production at time of available waste heat, and use of this hot water later if there is a mismatch between supply and demand. Similarly, the MT circuit also has storage of water in tanks, but at a temperature of 33°C . The three temperature loops all have their respective ”suppliers” and ”consumers” of energy. Figure 3.3, 3.4 and 3.5 below shows simplified schematics of the three temperature circuits.

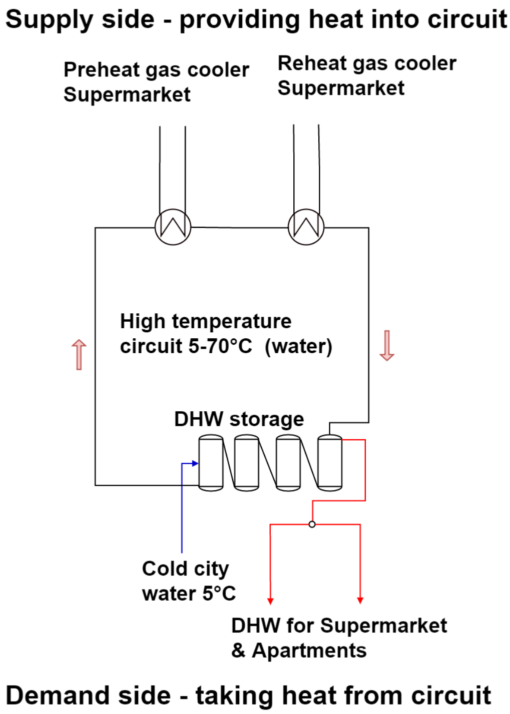

Figure 3.3: Principle design of the high temperature circuit

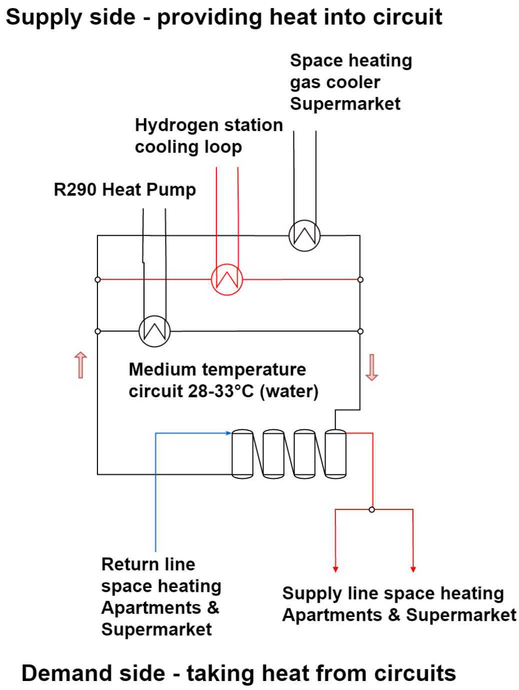

Figure 3.4: Principle design of the medium temperature circuit

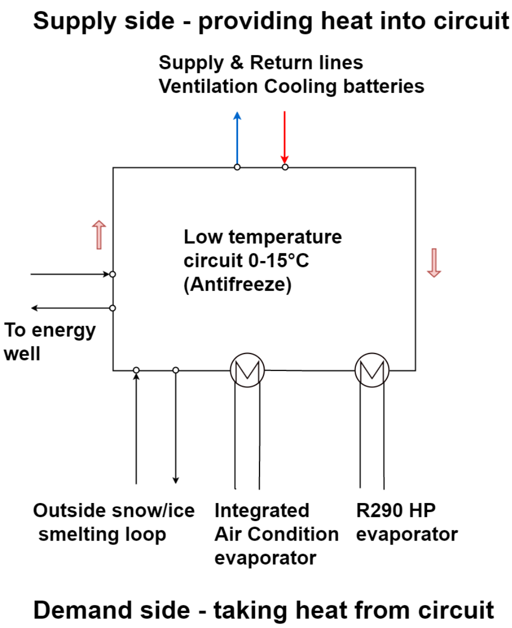

Figure 3.5: Principle design of the low temperature circuit

A more detailed discussion on the different subsystems in the centralized energy system, including suppliers and consumers of heat in each temperature circuit, will follow in the next sections.

3.2.4 Centralized heating and cooling system

The objective of this work is to minimize the import of primary energy to the building plot. Waste heat is available from several sources on the building property, and the best way to avoid importing energy is to utilize as much of the waste heat as possible. By designing the heating and cooling system for the building in such a way that the whole building plot is connected together, more waste heat can be utilized. The building envelope consists of the supermarket and apartments, and they are the consumers of thermal energy in the system. On the other hand, the supermarket refrigeration system and hydrogen refueling station are the suppliers of waste heat to the system. Connecting the demand for heating to the supply of waste heat in a centralized system ensures as much as possible of the waste heat is utilized.

As described in section 2.3, heating demand for domestic hot water is becoming the dominating load on the heating system in residential buildings. Therefore it is important that the heating system for DHW consumes as little energy as possible per unit of hot water produced. As shown in the previous section, the supermarket refrigeration system is designed so that hot water at 70°C is supplied as waste heat when there is refrigeration load in the supermarket. More details on the heat recovery system from the supermarket refrigeration system is presented in section 3.2.5 later in this chapter. To ensure that as much of the produced hot water is utilized, hot water tanks is needed to handle the expected offset between supply and demand of DHW in the building. Storing hot water at high temperature is an advantage as the energy intensity of a given water volume increases with temperature. Another advantage is that growth of the legionella bacteria in the DHW system is avoided when storing the hot water at 70°C instead of storing at consumer temperature around 45°C. When hot water is required by residents in the building, 70°water is taken from the tank and mixed with cold city water to supply the consumer with the correct temperature. See figure 2.9 for illustration of the principle.

The total storage volume of the DHW tanks need to be sized so that the hot water produced during refrigeration load in the supermarket can be stored for use later when hot water is needed. The ultimate goal is to be able to supply the whole demand for domestic hot water by heat recovery from the supermarket, saving the investment cost of a separate water heating system completely. The demand for domestic hot water in the building is calculated according to the technical documentation SN/TS 3031 from 2016, which offers estimated energy demand for hot water per unit area of different building categories (Standard Norge, 2016). The technical documentation also includes energy demand per hour during the day for domestic hot water for the different building categories. This gives an opportunity to calculate a schedule for daily energy demand for domestic hot water in the building in this thesis. Using the technical documentation together with the building specifications in section 3.1, the DHW demand for one day is calculated for the supermarket and the apartments. The demands are added together, making up the total energy requirement on the domestic hot water system for one day. The resulting load curve is presented in figure 3.6 below.

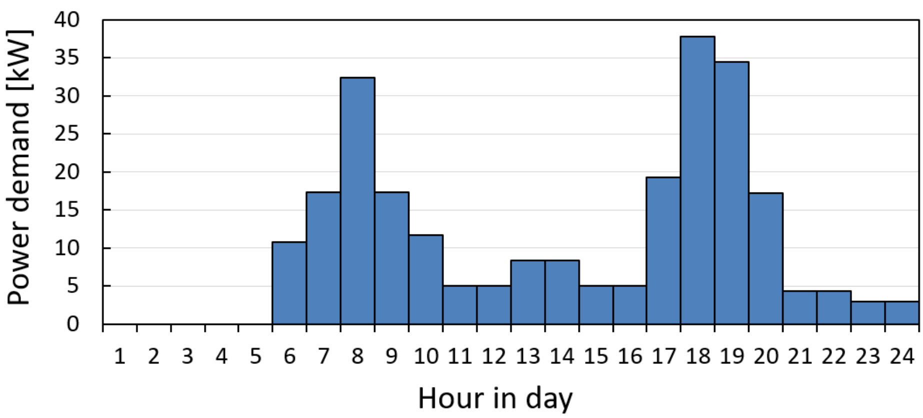

Figure 3.6: Energy requirement for domestic hot water during 24 hours in the building.

The load profile for domestic hot water consumption shows two distinct peaks where most of the hot water is consumed in the building. This is due to the assumption that the occupants of the apartments use much hot water during the morning routine. The occupants shower, cook breakfast and prepare for school and work, typically between six and nine o'clock. The other peak is associated by cooking dinner, cleaning and showering after exercising in the afternoon/evening. The large peaks are mostly due to the load profile of the apartments. The supermarket has a more evenly distributed hot water consumption during the operating hours. Summing up the energy demand for hot water for the entire day, a total of 249.89 kWh is found. When calculating the demand for domestic hot water for buildings, it is normally assumed that the energy demand is equal every day through the year. The same procedure is followed here, and the pattern for hot water demand is assumed to repeat itself each day.

Some simple preliminary calculations were necessary in order to establish the domestic hot water system in Dymola. To exploit the available waste heat, a thermal storage of hot water of at least one days consumption is assumed to be necessary. This allows for offset between supply and demand for hot water. Calculating the required storage volume is carried out according to equation (3.1) and (3.2) below.

$$E_{DHW} = m \times (h_{set} - h_{ref}) \tag{3.1}$$

$$V = m \times \rho \tag{3.2}$$

The properties of water were found by use of an Excel sheet with the add-on RnLib. RnLib is a refrigerant library and contains thermodynamic properties for a large number of refrigerants. The library was developed by NTNU/SINTEF. Assuming the density of water is 1000 kg per m^3 and the reference temperature is the temperature of the cold city water at 5°C, a total volume of approximately 3200 liters is found. This corresponds to the hot water consumption for one day in the building. This is assumed to be sufficient storage for the system. The total storage volume is assumed to be separated into 4 equally sized hot water tanks of 800 liters each, connected in series. The consumer port is set at the first tank, and the hot water volume drawn from the tank is replaced by cold city water at the bottom of the last tank in the series.

A screenshot of the domestic hot water system model in Dymola is shown in figure 3.7 below. For simulation in Dymola, a the whole storage volume is modelled as one single tank. The representation of the tank is carried out by sizing a pipe component to store the correct volume. The pipe is separated into 20 equal liquid cells, where temperature and other properties of each cell can be logged. This enables investigation of the stratification effect in the storage volume as hot water is drawn according to the demand and supplied according to availability of waste heat.

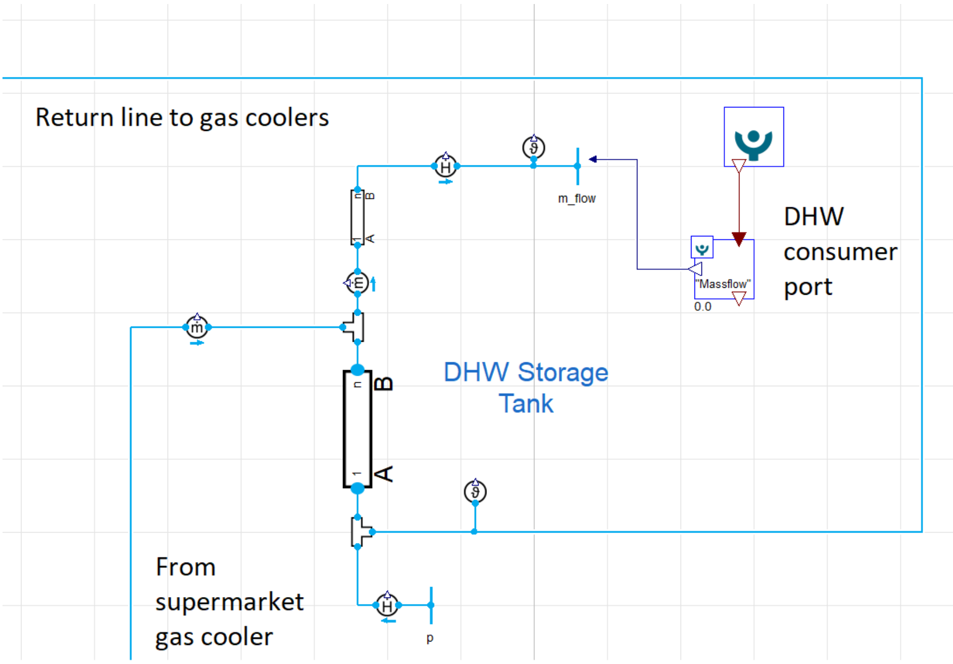

Figure 3.7: Model of the DHW tank and associated supply system. Screenshot from Dymola.

The demand for hot water in the simulation model is represented by a mass flow of water drawn from the tank, denoted as the DHW consumer port in the figure. The required heating per hour from figure 3.6 is calculated to a mass flow using the enthalpy difference between water at city water temperature and setpoint temperature, 5 and 70°C. The mass flow of water corresponding to the heating effect is found by using specific enthalpies from RnLib. The data is then given as a two column matrix, one column for time and one for the corresponding mass flow. The data is read by a block from the TIL Filereader library in Dymola. When water at 70°C is drawn from the top of the tank, the corresponding volume is replaced by 5°water at the bottom of the tank. The return line to the supermarket gas cooler from the bottom the tank closes the circuit of domestic hot water.

The logic and structure behind behind the space heating system is much the same as the for the domestic water system. A centralized solution where both waste heat and heat by the auxiliary heating system is supplied to storage tanks in order to shave off peaks of demand. The heating of the building is fulfilled by hydronic floor heating in each apartment as well as heating of ventilation air. It assumed that the heating system has a supply temperature of 33°C and a return temperature of 28°C both for heating of ventilation air and the room heating. Normally heating batteries in the ventilation has a larger temperature difference between supply and return to reduce the size of the heating batteries. For simulation in this thesis a common supply and return temperature is accepted to reduce the complexity of the model. The heating demand of the building is found by building simulation in SIMIEN, and is presented in section 4.1. The heating demand for space heating is calculated as a corresponding mass flow of water at 33°C, similarly to the procedure for domestic hot water. The difference is that the reference temperature for the space heating system is 28°C, compared to 5°C for the domestic hot water system. Figure 3.8 shows the representation of the space heating system in Dymola.

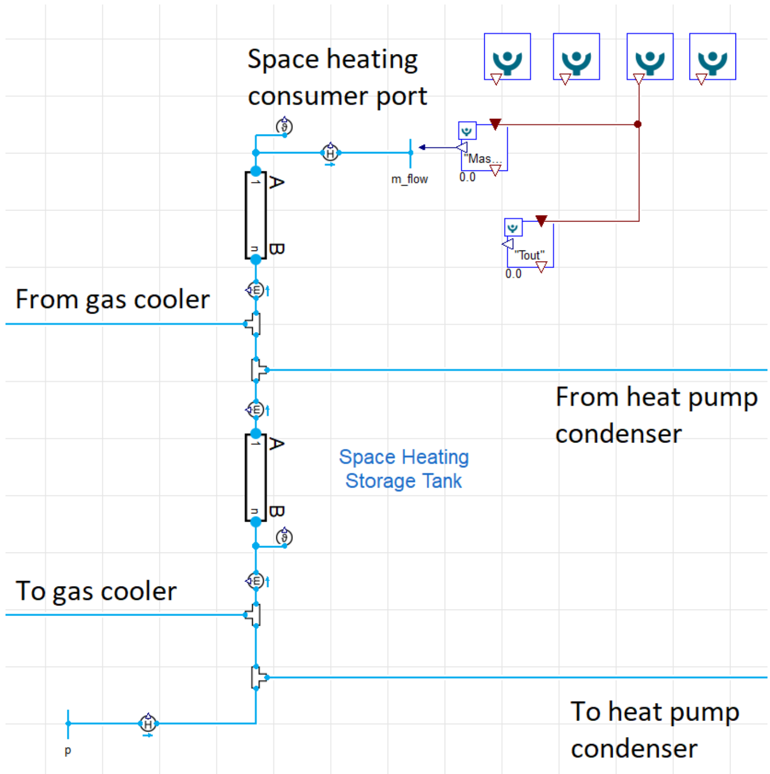

Figure 3.8: Model of the SH tank and associated supply system. Screenshot from Dymola.

The mass flow of water at the consumer port is set according to the demand schedule from the building simulation. Supply of waste heat from the supermarket is stored in the space heating tank, while the heat pump supplies heating when recovering waste heat is not covering the full demand. A more detailed description of the auxiliary heat pump and its operation can be found in section 3.2.6. The tank volume is modeled in the same way as for the domestic hot water tank. Since there is only a 5 degree temperature difference between supply and return in the tank, the energy intensity of a given water volume is very low. A storage of 1 m^3 33°C water corresponds to approximately 6 kWh when the reference is 28°C. For the domestic hot water tanks, where the temperature difference is 65°C, a 1 m^3 storage volume equals approximately 78 kWh of thermal energy. This means that the space heating tank is functioning more as a node for redistribution of the thermal energy than a storage. However, the storage function increases with the storage volume.

But since the energy intensity is low, a moderate tank volume is chosen to balance investment cost and space consideration with the increased storage function. For the simulation model a total tank volume of 5 m^3, corresponding to approximately 30 kWh is chosen.

The return line to the supermarket gas cooler goes by a heat rejection system with dry coolers. As the hot water tanks gets charged, the temperature of the return line to the gas coolers increases. To ensure stable and high performance of the supermarket refrigeration system, the return temperature should be kept as low as possible for the domestic hot water circuit. Depending on the season that is simulated, there may be more heat available from the supermarket than what is needed for heating the building and domestic hot water. For this reason a heat rejection system is needed. Figure 3.9 below shows the representation of the dry coolers in the simulation model designed to give off heat to the ambient when there is surplus of waste heat in the system.

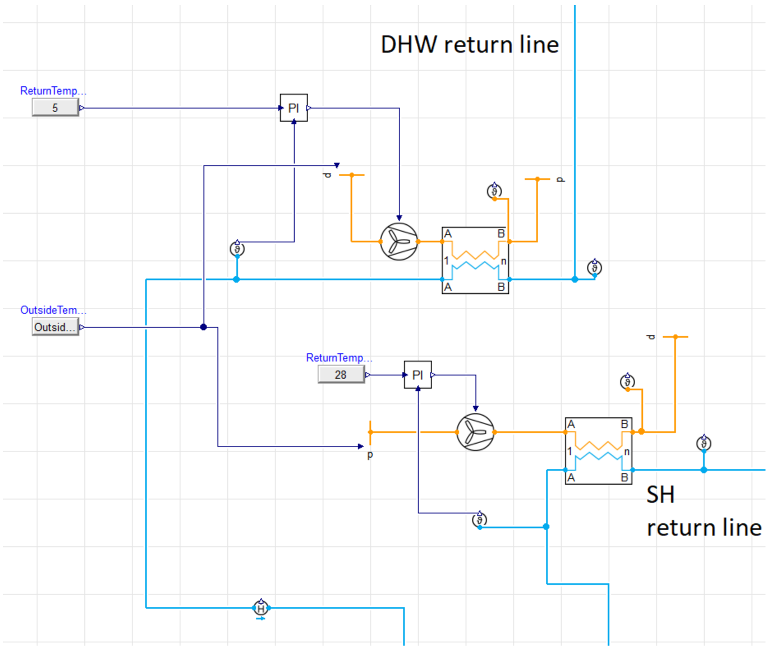

Figure 3.9: Model of the heat rejection system with dry coolers. Screenshot from Dymola.

As the domestic hot water tank is getting fully charged with water at 70°C, the temperature of the water in the return line to the CO_2 gas coolers starts to rise. This is due to the stratification effect in the hot water tank. As reviewed in the literature study, a low return temperature is critical for achieving a high COP for the CO_2 trans-critical refrigeration system. Therefore a dry cooler is placed on the return line to the reheat gas cooler. In the model the water rejects heat in a fin and tube counter-current heat exchanger, giving heat to ambient air. The ambient air is modelled as an infinite heat sink with a temperature equal to the outside temperature for the week that is simulated. A controller for the fan increases the volume flow through the heat exchanger until the water setpoint temperature of 5°C is reached. If the ambient temperature is so high that the setpoint can not be reached, the fan operates at max capacity and lowers the temperature as much as possible.

The heat rejection in the space heating circuit is modelled in the same way as the domestic hot water circuit. The heat is rejected to ambient air, and a controller ensures the return temperature of the water to the space heating gas cooler is 28°C. In the model, the gas cooler pressure is kept at 85 bar irrespective of if there is heating demand or not. The CO_2 refrigeration system have to operate trans-critical even for providing heat to the space heating system, due to the low critical temperature of CO_2 (31.1°C). Because space heating and domestic hot water demand not always occurs at the same time, adapting the gas cooler pressure to the dominating demand would mean a frequent adjusting. Instead, the gas cooler pressure is kept constant at 85 bar and dry cooler duties are logged. The energy rejected in gas coolers are presented in the result to see what the remaining potential for heat recovery is after as much heat as possible has been recovered to the building. A more detailed description of the supermarket refrigeration system and heat recovery system is presented in section 3.2.5

The purpose of the low temperature circuit in the energy system is mainly to provide cooling of the building and function as a heat source for the auxiliary heat pump for the space heating system. A simple sketch of the system layout is presented in figure 3.10 below.

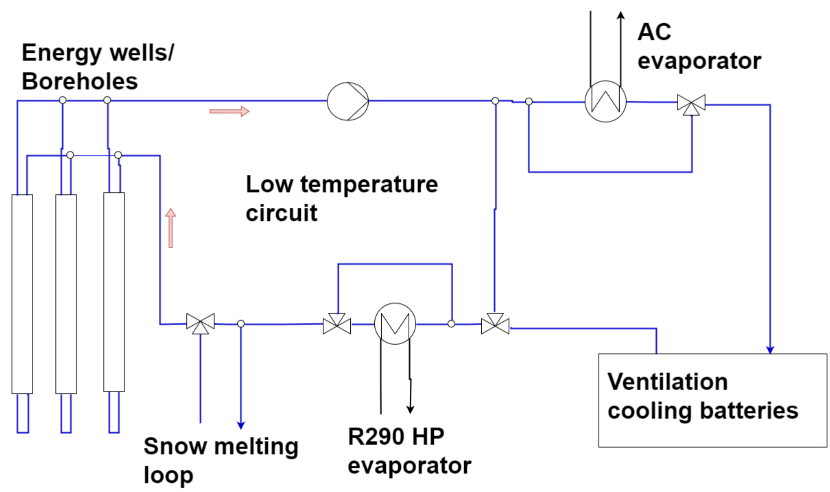

Figure 3.10: Overview of the low temperature circuit with components.

The boreholes is a central component in the low temperature circuit. A antifreeze refrigerant is circulated in the deep holes and either absorbs or rejects heat with the surrounding bedrock, depending on the season. During winter, heat is taken from the boreholes as source for the heating system. During summer, heat is rejected from the cooling system into the boreholes. When there is cooling demand in the building, a supply line of antifreeze is sent to the ventilation cooling batteries in the ventilation channels. A ventilation cooling system has typically a supply temperature of 7°C and return temperature of 12°C. If the temperature of the antifreeze is at supply temperature or lower, so-called ”free cooling” is possible using the antifreeze directly in the ventilation batteries. This type of cooling obtains very high COP, as the only energy needed to deliver the cooling duty is the pump to circulate the cooling water/antifreeze. If the temperature of the antifreeze is higher than the set point temperature, the antifreeze is cooled by the AC evaporator connected to the supermarket refrigeration system. Liquid CO_2 is sent to the AC heat exchanger at lower temperature than the antifreeze, and is evaporated as it exchanges heat with the antifreeze. This lowers the supply temperature of the antifreeze to the setpoint of 7°C.

The ground has a stable temperature through the year in comparison to outside air. Winter is the time of the year with highest demand for space heating and lowest outside temperature. This is a challenge for air-source heat pumps, but is avoided by using a ground source heat pump using bedrock or horizontal collectors as heat source. The evaporator of the ground source heat pump for this energy system is placed on the return line from the ventilation batteries, assuring the evaporation temperature is as high as possible. A lower pressure lift for the heat pump compressor is beneficial for the COP of the system. The system is designed so that it can be used also when there is combined cooling and heating demand. At times with heating demand, the ventilation circuit is shut off and antifreeze is circulated through the heat pump evaporator only. With only cooling demand, the heat pump is bypassed. During combined operation, both circuits are open and the antifreeze is sent to the ventilation batteries and the return line is mixed with supply line to the heat pump evaporator. There is also a possibility to use some remaining heat for snow/ice melting outside of the supermarket. Because of the building for this case is located in Tromsø, the cooling demand is predicted to be fairly low. Therefore the low temperature circuit is not included into the simulation models developed in Dymola, and more focus is put on the heating side of the energy system. The design of the low temperature circuit is still included in the description of the system, so that the building concept is suitable also for warmer climate conditions than north of Norway.

3.2.5 Supermarket refrigeration system and heat recovery

One of the most important subsystems in the energy system of the building is the refrigeration plant. For the supermarket model in this thesis, a CO_2 booster system with parallel compression is chosen. It is regarded as state of the art technology for supermarkets in Scandinavia. The refrigeration plant provides freezing and chilling of the goods in the supermarket. A detailed explanation of commercial CO_2 booster system was given in section 2.1.1 of this paper. Before going into the details on the different components and their function in the simulation model, some boundary conditions and overall operational parameters for the system is given in table 3.5 below.

Characteristic	**Value**	**Unit**
Chilling capacity	60	kW
Freezing capacity	15	kW
Refrigerant	R744	-
LT cabinets setpoint	-20	°C
MT cabinets setpoint	2	°C
Pressure in intermediate pressure receiver	40	bar
Gas cooler pressure	85	bar
Compressor isentropic efficiency	0.7	-

Table 3.5: Specifications of the refrigeration system in the supermarket.

The capacity of the supermarket is based on installed capacity of similar supermarket sizes. The refrigeration load on the CO_2 booster system is set according to typical values during the day, where the base load for most of the day is approximately 20% of maximum installed capacity, with some peak during the opening hours. A behaviour typically observed from similar systems is that load increases during loading of fresh goods into the freezing and chilling cabinets, which occurs primarily before noon. The other distinct peak during the afternoon is associated with the hours with most customers in the shop. This occurs typically after working hours, approximately between 15.00 and 18.00 hours. Frequent opening of the doors and lids of the refrigerated cabinets increases the load during these hours. The assumed refrigeration load curve on the supermarket refrigeration system during a normal working day is shown in the figure 3.11 below. For simplicity, the refrigeration load pattern is assumed to be equal every day for the whole week simulated in Dymola.

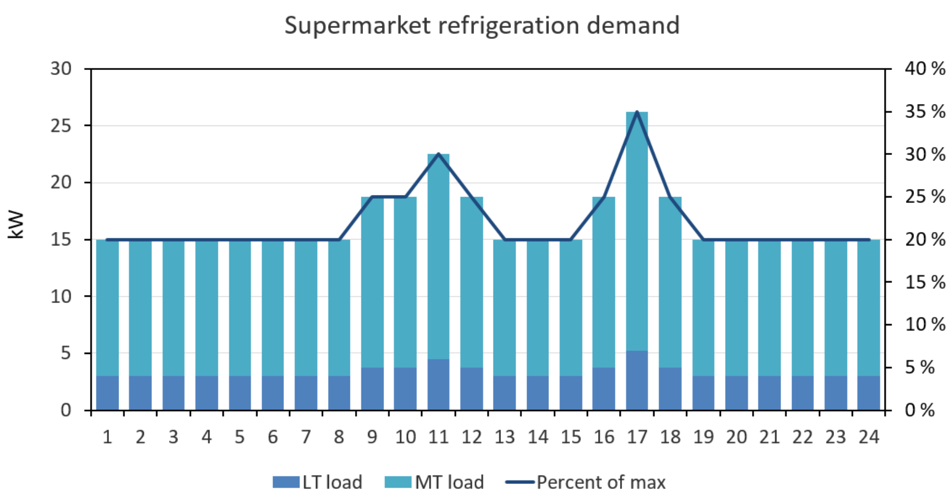

Figure 3.11: Load on the supermarket refrigeration system during a day.

A brief description of the components in the supermarket simulation model including control system follows in the next paragraphs.

The chilling and freezing cabinets in the supermarket are modelled as a closed air volume, where the refrigeration load is simulated as a heat boundary connected to the air volume. Figure 3.12 below shows a screenshot of the LT cabinet model in Dymola.

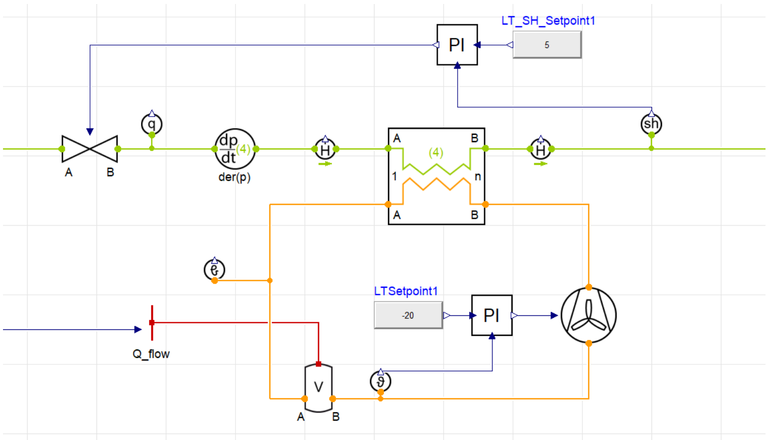

Figure 3.12: Model of the LT cabinets in Dymola.

The closed air volume represent the refrigerated volume inside the freezing cabinets in the supermarket. Although most supermarket has several small freezing cabinets standing in the shop, they are added into one larger volume is here. The freezing load is also summarized for the whole supermarket and set as a fixed heating effect in watts to the air volume via the heat boundary, noted as Q flow in the figure. The temperature of the freezing cabinets are controlled as follows. A sensor measures the temperature out of the cabinet, and a controller controls the volume flow through the fan. The volume flow increases to reach the setpoint temperature of -20°C for the cabinet. On the refrigerant side, liquid CO_2 at 40 bar from the intermediate pressure receiver is sent to the thermostatic expansion valve before the LT evaporator. The valve throttles the refrigerant and lets through just enough refrigerant to reach five degrees superheat at the outlet of the evaporator.

The pressure in the LT evaporator is maintained at 15 bar by the LT compressor rack. The pressure is maintained at 15 bar during all simulations irrelevant of the temperature in the LT cabinet. The MT chilling cabinets in the supermarket are modelled in the same manner as the LT cabinets. The only difference is the evaporation pressure of 30 bar and temperature setpoint of 2°C. The refrigeration load of the cold storage room is included in the load put on the MT cabinets.

Figure 3.13 below show a screenshot of the LT compressor in the supermarket model.

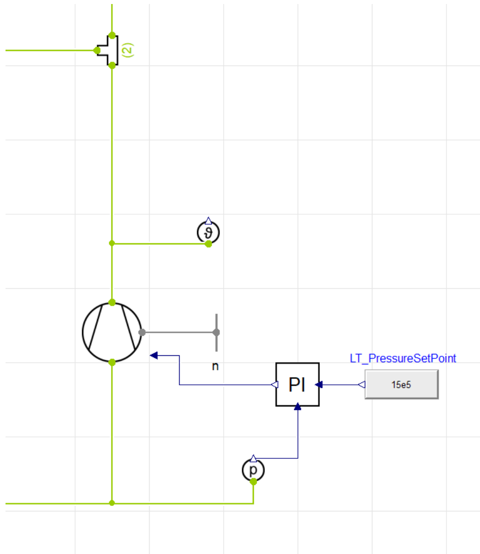

Figure 3.13: Model of the LT compressor in Dymola.

The LT compressor rack is modelled in Dymola as a single compressor with varying relative displacement. The gray line and the mechanical boundary n represent the the mechanical shaft and electric motor driving the compressor. When the compressor is on, the frequency is set to 50 Hz. The controller ensures the suction pressure kept at 15 bar. As the refrigeration load increases, more refrigerant is evaporated in the LT evaporator, increasing the pressure on the suction side of the compressor. The pressure sensor sends a signal to the controller, which increases the relative displacement of the compressor. This represents more compressors being activated in the compressor rack, by increasing the total volumetric capacity. When refrigeration load decreases again, the process is reversed to ensure that the suction pressure does not drop below 15 bar. The refrigeration load is never zero due to infiltration losses and other heat load that is present 24 hours a day. This means the mechanical port to the compressor is active all day, but the compressor relative displacement is lowered outside working hours of the supermarket.The discharge gas of the LT compressor is sent to the suction line of the MT compressor, which is modelled in the same way as the LT compressor. Table 3.6 below gives information on the compressors in the system.

Variable	**MT Compressor**	**LT Compressor**
Suction pressure	30 bar	15 bar
Discharge pressure	85 bar	30 bar
Displacement	1.5e-5 m^3	0.7e-5 m^3
Relative displacement variation	1 to 4	1 to 4

Table 3.6: Operational data for the compressors in the supermarket

The high pressure side of the supermarket refrigeration system consists of the three gas coolers in series. At the outlet of the preheat gas cooler the cold CO_2 gas is throttled by the high pressure control valve, lowering the pressure from 85 bar to the intermediate pressure level at 40 bar. The high pressure control valve is controlling the gas cooler pressure to be held at 85 bar. The high pressure side of the refrigeration system is shown in figure 3.14 below.

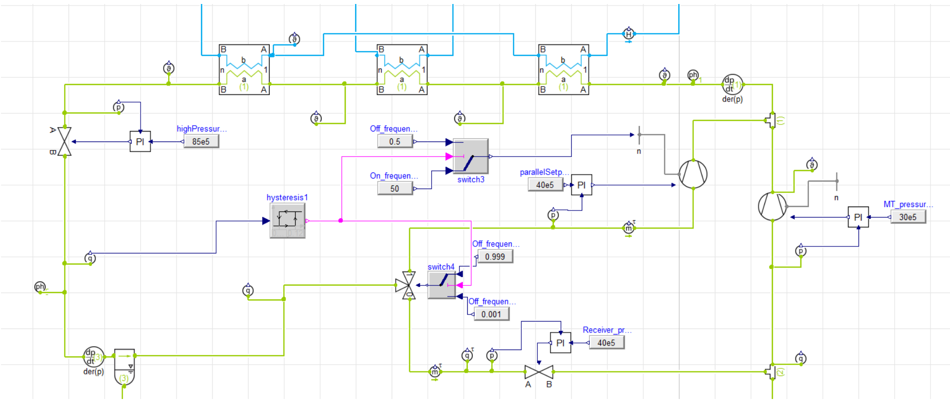

Figure 3.14: Supermarket refrigeration system high pressure side. Screenshot from model in Dymola.

After throttling to the intermediate pressure tank, the liquid refrigerant is supplied to the evaporators by the pressure difference between the tank and the evaporators. It is assumed that a 10 bar difference is enough to supply the required amount of refrigerant to the MT evaporators which operates at 30 bar. Depending on the temperature after the gas coolers, there will be some vapor formed after throttling. The pressure in the intermediate pressure tank is controlled in two ways, dependent on the vapor formation after throttling. If the vapor quality is lower than 0.20, a flash gas bypass valve throttles the formed flash gas to MT evaporation pressure level of 30 bar. The controller connected to the valve controls the intermediate tank pressure by opening and closing the flow area in the valve. If the vapor quality is larger than 0.20, the three-way valve closes the line towards the by-pass valve and opens towards the parallel compressor.

The purpose of the parallel compressor arrangement is to reduce the load on the high pressure compressor rack. As explained in the literature study, the operation of the refrigeration system in high ambient temperatures results in a high vapour fraction formation after throttling. The purpose of the parallel compressor is to suck off the gas from the receiver and compress it directly up to gas cooler pressure, instead of throttling down to MT evaporation pressure through the flash gas by-pass valve. By direct compression to gas cooler pressure the parallel compressor works with a lower pressure ratio than the high pressure compressors, which is normally associated with higher isentropic efficiency. This results in lower power consumption compared to first throttling down to the MT evaporator pressure before compressing to gas cooler pressure.

Heat recovery from supermarkets has been analyzed and reviewed in section 2.1.2.

The supermarket refrigeration system is in continuous operation throughout the year, and reliability of these systems are top priority for the supermarket shop owners. The chilled and frozen goods in storage represent a significant value, and keeping the temperatures at the required levels is critical for quality. Based on this and the findings in the literature, heat recovery from the supermarket refrigeration system is considered a reliable and good solution.

The fact that the refrigeration system is using CO_2 as refrigerant opens up for a possibility to recover heat a high temperature. By running the refrigeration system in transcritical mode, water can be heated up to temperatures required for tap water. CO_2 heat pumps for tap water heating has been explained and reviewed in section 2.3.1. Since a large share of the building plot consists of apartments, there is a much larger need for domestic hot water compared to the supermarket alone. This gives an interesting opportunity to utilize the large potential that lies in heat recovery from a CO_2 refrigeration system. For this reason, a tripartite gas cooler on the high pressure side of the CO_2 refrigeration system is chosen for rejecting the heat. By implementing this solution, heat recovery to both domestic hot water and space heating can be done simultaneously. The solution is inspired by the suggested system design for a combined DHW and space heating heat pump using CO_2 as refrigerant, explained in more detail in 2.3.1. The design of the high pressure side in the supermarket will replicate the corresponding high pressure side in a CO_2 heat pump. A principle sketch of the gas cooler layout in the supermarket is presented in figure 3.15 below.

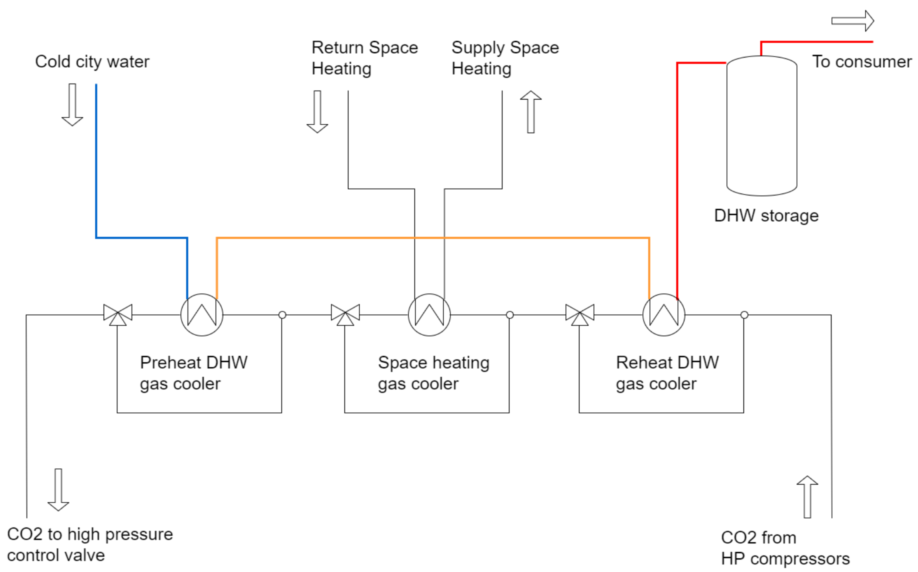

Figure 3.15: Principle sketch of the tripartite gas cooler solution for the supermarket.

Following the flow direction of the refrigerant in the figure above, the process is as follows. The discharge gas from the high pressure compressor rack has the highest temperature and gives off heat the water in order to reach the desired temperature for hot water storage. The outlet temperature of the water from the reheat gas cooler is controlled by a variable speed drive pump, which controls the mass flow, and thereby controls the outlet temperature. The CO_2 is sent to the next gas cooler, which is connected to the space heating system in the building. In the space heating gas cooler, CO_2 is cooled further down and exchanges heat with the circulating water in the circuit. At design conditions the refrigerant exits the second gas cooler at approximately 31°C. In order to cool the CO_2 down to the final temperature, the refrigerant is sent through the third gas cooler. This heat exchanger acts as a preheater for domestic hot water, where cold city water is sent through on the cold side of the gas cooler and exits at approximately 28°C. On the refrigerant side, the CO_2 is cooled down to about 8°C at design conditions, implying a temperature difference of 3 degrees at the cold end. The process is illustrated in a temperature-enthalpy diagram in figure 3.16 for a gas cooler pressure of 85 bar, and no pressure drop.

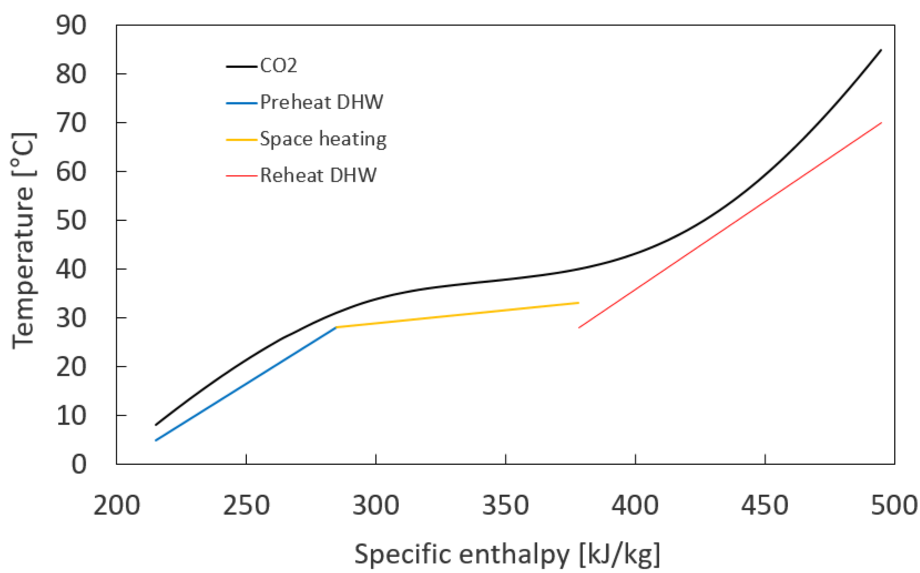

Figure 3.16: Temperature-enthalpy diagram of a trans-critical CO2 cooling process. Gas cooler pressure of 85 bar.

The black CO_2 line represents the isobar (constant pressure line) of 85 bar, and the temperature glide during the heat rejection process is evident. The cooling of the CO_2 is happening from right to left in the diagram, while the heating of the water at different stages are represented from left to right. The specific enthalpy on the x-axis is related to the refrigerant only, while the temperature on the y-axis is related to both refrigerant and water. Table 3.7 below summarizes the water temperatures at the inlet and outlet of the three gas coolers in the heat exchange process.

Gas cooler	Flow direction	Water temperature
Preheat DHW	in	5°C
	out	28°C
Space Heating	in	28°C
	out	33°C
Reheat DHW	in	28°C
	out	70°C

Table 3.7: Inlet and outlet water temperatures of the three gas coolers.

By arranging the the high pressure side in this manner, heat can be recovered to satisfy some of the heating demand for both space heating and domestic hot water. Since the supermarket refrigeration system is in operation all day, no other separate system is included for domestic hot water production. The remaining demand for DHW that can not be satisfied by the heat recovery process is assumed to be covered by electric heating in the buffer tanks.

There is also an opportunity to dedicate the full heating duty of hot CO_2 gas to hot water production only. If this is the case, the space heating gas cooler in the middle is bypassed, and all heating duty is taken by the preheat and reheat gas coolers. However, the gas cooler pressure often need to be altered if only heating of tap water is done. This was described by (Nekså et al., 1998), increasing the water setpoint from 60°C to 70°C resulted in an increase in optimum gas cooler pressure from 90 bar to 100 bar. With lower gas cooler pressures may cause a pinch in the gas cooler, see figure 3.17 below.

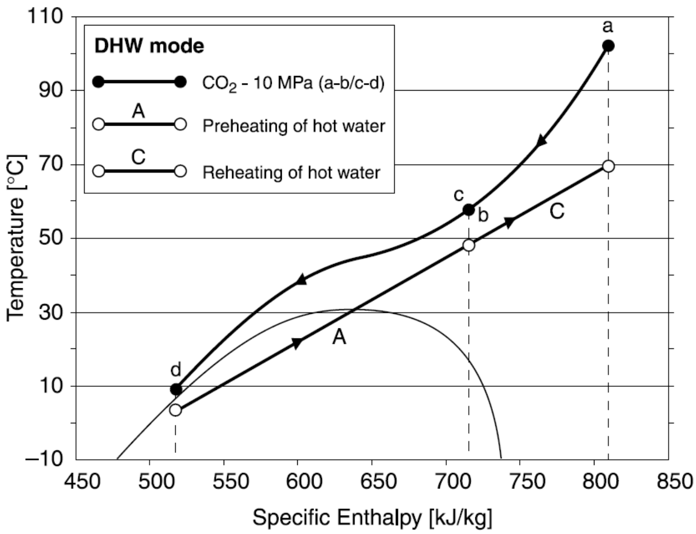

Figure 3.17: Temperature-enthalpy diagram for a water heating process, taken from (Stene, 2005).

The figure shows the process for a gas cooler pressure of 100 bar. Lowering the pressure can make a pinch at the point marked c/b in the gas cooler due to the sway-backed shape of the CO_2 cooling curve. The swing of the isobars for CO_2 in overcritical pressure gets more and more distinct the closer to critical pressure they are, eventually causing a pinch in the gas cooler. For space heating an additional system for satisfying the heating demand is required, because most of the heat recovered in from the refrigeration system is to domestic hot water.

There can be some hours of operation where the DHW tanks are fully charged with 70°C water, but there is still a certain need for space heating. For operational conditions like this, the reheat gas cooler can be bypassed and all heat from cooling the CO_2 gas down from discharge temperature to approximately 31°C can be made available to the space heating circuit. After exiting the middle gas cooler, the CO_2 gas is either throttled down to intermediate pressure level or, preferably sent to the a subcooler heat exchanger connected to the boreholes via the antifreeze circuits. This allows the temperature before throttling to be lowered. As described in the literature study, a low temperature after the gas cooler is important to avoid large vapor fractions after throttling through the high pressure control valve. The cooling curve for CO_2 and heating of water for operation in space heating mode is presented below in figure 3.18 for a gas cooler pressure of 85 bar as before.

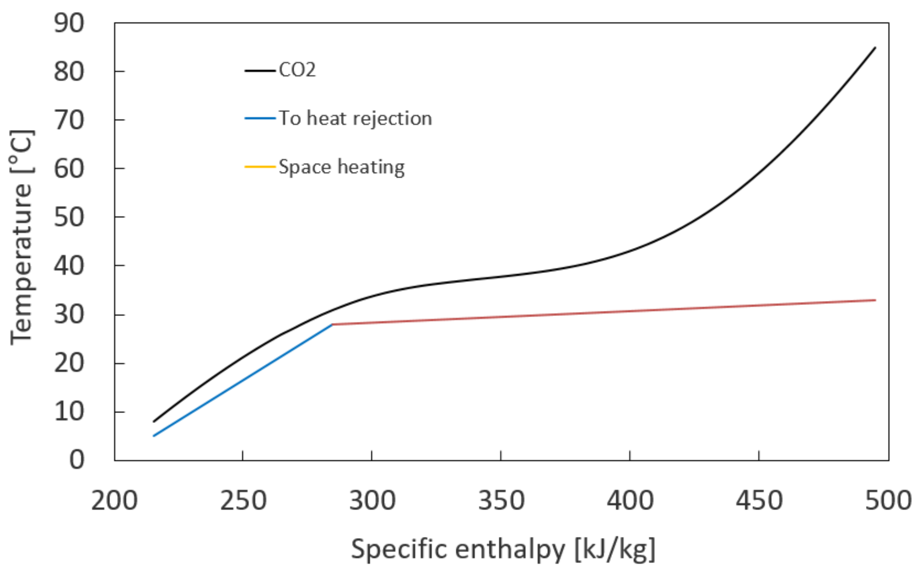

Figure 3.18: Temperature-enthalpy diagram of a trans-critical CO2 cooling process, space heating mode. Gas cooler pressure of 85 bar.

As can be seen from the figure, operating the gas coolers in the supermarket in space heating mode involves a large temperature difference between the water and refrigerant.

Since the setpoint temperature of the space heating system is higher than the critical temperature of CO_2, the system needs to operate in trans-critical mode. As a result, the large temperature difference occurs. A condensing medium will have a much better temperature fit to a space heating system with a five degree temperature fall between supply and return. Hence, it is chosen to operate the supermarket gas coolers in combined mode for all simulations, and another source of heating is chosen to cover the remaining space heating demand in the building.

The simulation model of the supermarket is built to represent as a real life supermarket as closely as possible. For developing the simulation model a set of boundary conditions for the operation is needed. Table 3.8 below summarizes the operational conditions for the model of the supermarket.

Characteristic	**Value**	**Unit**
Refrigerant	CO_2	-
Gas cooler pressure	85	bar
MT (chilling) pressure	30	bar
LT (freezing) pressure	15	bar
Intermediate pressure level	40	bar
Superheat MT/LT evaporators	5	kelvin
Compressors isentropic efficiency	0.7	-

Table 3.8: Characteristics and operational boundaries for the supermarket CO_2 refrigeration system.

For all cases where heat recovery is calculated, the value is always the net heat recovery. The general equation (3.3) is showed below.

$$Q_{HR,net} = Q_{HR} - Q_{drycooler} \tag{3.3}$$

As an example, the DHW dry cooler duty is subtracted from the supermarket reheat and preheat gas coolers duty so that only the heat which is used in the building is taken into account. When heat is recovered in the gas coolers, but the tanks are fully charged, the heat is rejected by the dry coolers.

3.2.6 Auxiliary space heating system

As the supermarket CO_2 refrigeration system has a limited capacity to deliver heat to the low temperature space heating system, it is expected that a part of the demand for space heating have to be supplied by an additional system. The main objective of this thesis is to minimize the import of primary energy to the building property, which involves as much the heating demand should be covered by heat recovery. However, it is also of great importance that the heating system that provides the remaining heat for the building is as efficient as possible. The low temperature space heating system in the building along with the availability of a stable temperature heat source makes a heat pump a very interesting option. The boreholes on the building plot offers a stable, high temperature heat source (0-10°C), and with the relative low temperature level in the hydronic space heating system, this gives a low temperature lift for a heat pump. Since the heating system is designed to have a five degree difference between supply and return lines, a condensing heat pump cycle gives a good temperature fit between the circulating water and the refrigerant.

The design of a heat pump cycle starts by establishing the operational boundaries of the cycle. This includes condensing temperature, evaporating temperature, minimum temperature difference and heating capacity. A minimum temperature difference for the condenser and the evaporator of 5 kelvin is assumed to give a reasonable balance between investment cost and operational cost. As the setpoint temperature of the heating system is 33°C, the condensing temperature of the refrigerant is assumed to be 38°C. The antifreeze from the boreholes is assumed to have a temperature of 5°C at the evaporator inlet, and dropping 5 kelvin to the outlet to 0°C at the cold end. Assuming a superheat of the refrigerant at the hot end of the evaporator of 5 kelvin, the evaporation temperature is set to -5°C. The focus of the thesis is to reduce the environmental impact by smart design of the energy systems, so only natural refrigerants are considered candidates for the heat pump in this thesis. Table 3.9 summarizes the operational characteristics for the heat pump cycle used as a basis for refrigerant evaluation. A maximum heating capacity of 30 kW is assumed to be able to calculate volumetric flow rates and required compressor capacity.

Characteristic	**Value**	**Unit**
Condensing temperature	38	°C
Evaporating temperature	-5	°C
Isentropic efficiency compressor	1	-
Superheat evaporator outlet	5	Kelvin
Subcooling condenser	0	Kelvin
Heating capacity	25	kW

Table 3.9: Characteristics and operational conditions for the auxiliary heat pump.

With the operational conditions set, a simple cycle evaluation of different refrigerants is carried out in Excel with RnLib. Table 3.10 shows performance characteristics for different natural working fluids. Natural refrigerants considered as candidates for the heat pump are ammonia (R717), propane (R290), isobutane (R600a) and propylene (R1270).

Characteristic	**Ammonia**	**Propane**	**Isobutane**	**Propylene**
Pressure ratio	4.14	3.23	4.15	3.15
Compressor capacity [m^3/h]	29.2	37.9	135.6	31.0
COP theoretical	6.23	5.87	6.18	5.87

Table 3.10: Theoretical performance evaluation of refrigerants using RnLib.

From the table the expected performance for different working fluids can be seen. The lowest pressure ratio for the refrigerants is offered by propane and propylen, remarkably lower than the other two refrigerants. A low pressure ratio is usually associated with higher isentropic efficiency for the compressor, as isentropic efficiency falls with higher pressure lift. However, since the isentropic efficiency is set to 1 for this evaluation, the highest theoretical coefficient of performance (COP) is expected by ammonia and isobutane. COP is showing how many units of thermal energy the cycle produces per unit of compressor electrical energy consumed. The pressure level for which the different refrigerants operate under the boundary conditions in the cycle along determines the required compressor volumetric flow rate to deliver the heating capacity needed. Ammonia has the lowest required compressor capacity, propane and propylene has slightly higher, while isobutane has multiple times larger requirement. This is due to the very low absolute pressure level for isobutane under these boundary conditions. At evaporating conditions, the saturation pressure of isobutane is less than atmospheric, giving a large volume per kg of refrigerant. Overall, propane is expected to give high real life performance and efficiency in this heat pump cycle, providing a low pressure ratio and moderate swept volume of the compressor. Propylene offers similar performance, but since propane offers better availability of components and more mature technology, it is chosen as working fluid in the auxiliary heat pump. Off-the-shelf propane heat pumps and chillers can be bought today.

The temperature conditions from table 3.9 are used as a basis for establishing the model in Dymola, where the heat pump condenser is connected to the centralized space heating system in the building. Figure 3.19 below shows a screenshot of the propane heat pump in Dymola.

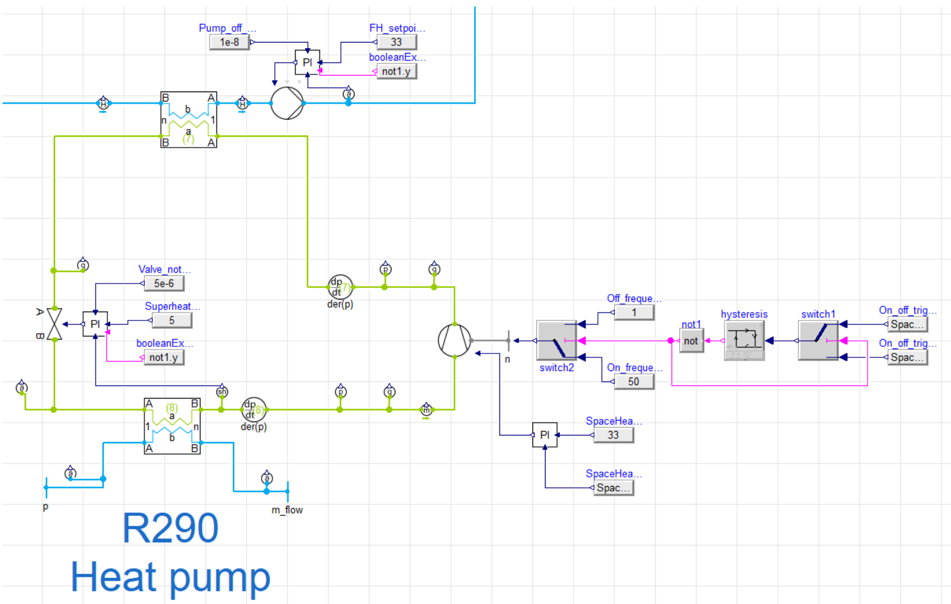

Figure 3.19: Dymola model of the propane (R290) heat pump for additional space heating.

The control of the heat pump is done as follows. A sensor on the bottom of the space heating tank measures the temperature of the bottom layer of water. When the heat pump is off and the sensor temperature reading drops to the return temperature of 28°C, the heat pump compressor is set to 50 Hz and starts circulating the propane refrigerant in the circuit. Heat is then delivered to the space heating tank until the bottom layer reaches the supply temperature of 33°C, and the compressor is shut off. A controller controls the relative displacement of the compressor in the same way as the compressors in the supermarket refrigeration system are controlled. A sensor on the top layer of water in the space heating tank measures the temperature there. If the temperature starts to drop below 33°C, the controller increases the relative displacement to increase the capacity of the heat pump. This ensures that the delivered temperature from the space heating tank is always 33°C. Both the condenser and the evaporator in the heat pump circuits are counter-flow plate heat exchangers. A thermostatic expansion valve ensures five degree superheat out of the evaporator before the gas is sucked by the compressor. When the heat pump is active and delivering heat to the water, a variable speed drive pump ensures the heated water has the correct outlet temperature of 33°C. The controller for the pump in the model is set to control mass flow of the water continuously, thereby reaching the desired water temperature. The heat source for the heat pump is antifreeze from the boreholes, and is set to constant 5°C and with a fixed mass flow rate, providing sufficient heat for the circulated refrigerant to be evaporated.

3.2.7 Case study 2: Hydrogen refueling station indirect heat recovery

Hydrogen refueling stations and related technology has been reviewed in section 2.2. A hydrogen refueling station with local production of hydrogen is planned to be built the building property. For both variants of waste heat recovery from the hydrogen station, it is assumed the station is equipped with an electrolyser. The electrolyser on the station is assumed to have large enough capacity to produce the entire demand for hydrogen fuel in the area, i.e no delivery of hydrogen by truck or similar. The electrolyser at the station produces low pressure hydrogen into a large storage volume. The large tank acts as a buffer for the fueling unit which raises the pressure up to filling pressure in stages. The dispenser unit itself is served with high pressure hydrogen from the filling storage.

The hydrogen station needs cooling at various locations, and this results in some waste heat production. The first proposed solution to investigate is called the *indirect* heat recovery method. This involves that a secondary medium transports away the heat produced by the refrigeration system located at the refueling station. The waste heat from the station comes from cooling of the compressors and other equipment, and then this heat is delivered by a secondary medium to the centralized energy system of the building. The principle is shown in the simplified sketch in figure 3.20 below.

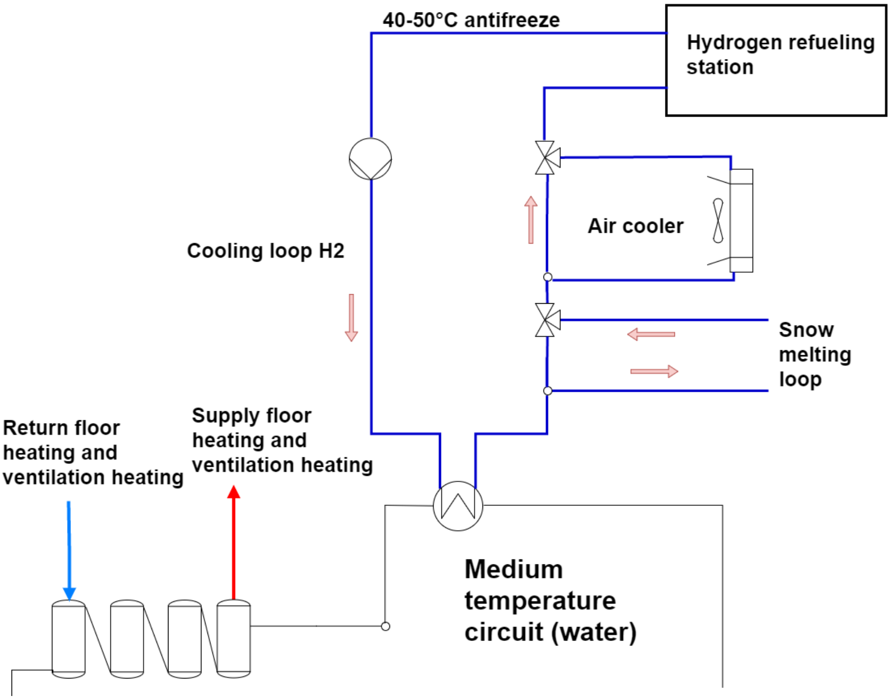

Figure 3.20: Principle solution of indirect heat recovery from a hydrogen refueling station.

As the figure indicates the station in the indirect heat recovery case considered a box with a supply and a return line of antifreeze/glycol. The cooling circuit of the hydrogen station is connected to the space heating circuit of the building energy system by a heat exchanger. The hot antifreeze cools down as it exchanges heat with the return line of the space heating system, increasing the water temperature from 28°C to the setpoint of 33°C. The lower the temperature of the space heating system, more of the waste heat from the hydrogen station can be used. The limiting factor for how much waste heat in the stream that can be recovered is the minimum temperature difference at the cold end of the heat exchanger. During winter time, the remaining low temperature heat in the antifreeze circuit could be used for snow and ice melting outside the building. Snow melting is often required outside of the supermarket during wintertime, for instance on pavement, entrance area, cargo port and loading ramp. This will ensure a higher utilization of the waste heat from the hydrogen station during wintertime, even though the heat is not used for space heating. In many supermarkets, ice melting on the loading ramp is often done by the use of electric heating elements for avoiding ice and snow.

Investigation of heat recovery in both case 2 and 3 requires estimation of how much heat that can be recovered for use in the building. For simulation purposes it is favorable to link the available waste heat to the amount of hydrogen produced at the station. Personal correspondence with the operator of hydrogen filling stations in Norway, UNO-X Hydrogen, gives some boundary conditions and approximate operational values for a typical hydrogen station they install. Relevant information is presented in table 3.11 below.

Characteristic	Value	Unit
Maximum production capacity	200	kg/day
Waste heat available	20-30 %	of supplied energy
Waste heat water temperature	40 - 50	°C
Installed power capacity	700	kW

Table 3.11: Characteristics and specifications for the hydrogen station including electrolyser.

The supplier of the electrolyser to the hydrogen filling station is NEL ASA, one of the largest electrolyser companies worldwide. From the website of the company information on a variety of electrolyser models is offered. Indications from the station operator suggests an electrolyser model delivered to operate inside a standard shipping container for outdoor use is appropriate. Therefore, the model used as a basis for calculations and simulations in this thesis is the electrolyser C-150. The choice is made on indications of the intended maximum capacity from the station operator and available range of products from NEL. The operational parameters for this electrolyser is open and available on the producers website (nelhydrogen.com, 2017). Operational parameters for the NEL C-150 are summarized in table 3.12 below.

Characteristic	Value	Unit
Capacity range, V_{flow}	50-150	Nm^3 H_2/h
Production dynamic capacity range	15-100	%
Maximum daily production capacity	320	kg
Electrolyser specific power consumption, e	3.8 - 4.4	kWh/Nm^3 H_2
Hydrogen outlet pressure	200	bar g
Operating temperature	80	°C
Power consumption at max production, P_{max}	750	kW

Table 3.12: Characteristics and specifications for the hydrogen electrolyser NEL C-150.

The data from UNO-X Hydrogen includes power and operation for the electrolyser, station module, fuelling storage and dispenser. This explains the higher power consumption even at only 200 kg/day production capacity, compared to a power consumption of 750 kW for 320 kg/day from the C-150 electrolyser. To compare the information from the whole station with the electrolyser only, some simple calculations were made.

The maximum production rate of the NEL C-150 electrolyser is 150 Nm^3 H_2/h. Assuming 24 hour continuous production, this equals 3600 Nm^3 of H_2 a day. At normal conditions (0°C, 1 atm), a cubic meter of hydrogen equals 0.08994 kg, giving a daily production of approximately 324 kg, which is in accordance with the capacity listed by NEL. Using the maximum production rate and the specific power consumption of the electrolyser from NEL, the power consumption of the electrolyser in kW can be found according to (3.4).

$$P_{electrolyser} = V_{flow,max} \times e \tag{3.4}$$

NEL gives a range of the electrolyser specific power consumption, and the power consumption of the electrolyser will vary accordingly. Using the range of specific power consumption, the electrolyser power consumption is found to be in the range of 570 kW to 660 kW during full load. Comparing the electrolyser power consumption with the total power consumption, a difference varying from 90 kW to 180 kW is found, corresponding to a 12 to 24% difference. This difference is assumed to be power to auxiliary equipment and heat taken away to maintain the operational conditions for the electrolyser.

For comparison of the operational data from NEL, data from another manufacturer is gathered. Hydrogenics is a large company providing systems for electrolysis, power to gas and hydrogen filling stations. The largest electrolyser in their outdoor portfolio with available data online is a HySTAT 60 Nm^3 H_2/h capacity electrolyser. The data is available at the company website (hydrogenics.com, 2017). The company reports a specific energy requirement of 5.2 kWh/Nm^3 H_2 with all systems included. Assuming the electrolyser itself has similar performance as the NEL electrolyser of 4.4 kWh/Nm^3 H_2, the difference between electrolyser and total energy consumption is 15.4%. This is in the same range as for the NEL electrolyser. It is assumed that this energy difference can be taken taken away as waste heat. This is an obvious simplification, since some of the supplied energy is lost through heat dissipation and radiation from the container where the electrolyser is placed. Furthermore, the heat needs to be collected by the liquid cooling circuit and transported to the space heating system of the building in order for it to be utilized.

Hydrogenics includes some information on the cooling system of their electrolyser HySTAT 60 in the product specification. In order to cool down the system a dry cooler and a chiller is supplied. The purpose of the chiller is to provide cooling water for gas cooler heat exchangers, increasing the gas purification process. The dry cooler provides cooling water for keeping the appropriate temperature of the electrolyte in the electrolyser system. Following this design, it is possible to take away a substantial amount of the additional power supplied to the electrolyser system which is not consumed by the electrolyser itself.

The information provided by the station operator, UNO-X Hydrogen, gives an approximate share of waste heat to be in the range 20 to 30% of the supplied energy to the station. The higher share given by UNO-X is due to the waste heat originating from the dispensing and fuelling part of the station also is included. As reviewed in section 2.2, a considerable share of the refrigeration demand in a hydrogen filling station is related to cooling of compressors and precooling of the hydrogen before dispensing. The system described in section 2.2.2 by Meratizaman et al. (2014) shows how the waste heat from a hydrogen filling station can be taken away as hot water at 55°C. This waste heat originates from raising the pressure of the hydrogen to dispensing pressure at 350 bar.

A record from the US Department of Energy DOE (2009) included in the Hydrogen and Fuel Cells Program contains some approximate data on cooling demands for precooling and compressor energy to alter the pressure of hydrogen to 700 bar. A calculation model from DOE estimates a need for 0.33 kWh/kg H_2 for cooling from ambient 15°C to -40°C and 0.44 kWh/kg from 30°C to -40°C. The characteristics of the precooling of hydrogen is that the duration is very short, and the required refrigeration effect is high. Rothuizen et al. (2013a) reports that the peak demand for cooling is between 60 and 80 kW, but only for a minute or so. The whole cooling process takes about to to three minutes, where the demand typical peak occurs halfway through the process.

To make a conservative choice regarding the waste heat available for heat recovery to the centralized energy system in the building, it is chosen to use the data from NEL and include waste heat from the electrolyser only. Although additional waste heat may be available from the fuelling and dispensing part of the hydrogen station, the waste heat originating from these sources are directly linked to the situation where hydrogen vehicles pull up to the station to refill. Because hydrogen gas is compressed to high pressure from the low pressure buffer storage tanks on demand, the cooling demand of the fuelling part of the station is linked directly to the actual filling process of fuel cell vehicles. With no good data available for times of the day with most customers, it is found best to link the waste heat availability with the electrolyser operation. If relevant data of customer frequency and more details on the refrigeration system in the dispenser unit is provided, this can be included in a later study.

The data from NEL is from a 320 kg/day capacity electrolyser with 750 kW power consumption at maximum production capacity of 150 Nm^3 H_2/hr. It is assumed that waste heat share of supplied energy to the electrolyser is 15%. This is in the low end of the data from NEL, and close to the data from Hydrogenics. As mentioned earlier, it is expected that some of the energy difference is not recoverable due to heat dissipation and radiation. This is compensated for by not including waste heat from the fueling and dispensing part of the system. To relate the daily production of hydrogen at the station to waste heat available for heat recovery, a linear relationship between hydrogen production and waste heat is is assumed. This involves that 750 kW is supplied to the station at full production of 320 kg H_2/day, where 112.5 kW (15%) can be taken away as waste heat. Maximum capacity per day requires continuous operation of the electrolyser, i.e 24 hours a day. It is assumed that for lower production volumes of hydrogen gas, the electrolyser still operates 24 hours a day, but with lower production rate to reach the desired quantity. Since the station operator UNO-X has a typical installed capacity of 200 kg/day, this is considered maximum production in the Dymola simulation model. Calculated data for available waste heat at different production volumes is provided in table 3.13 below.

Production [kg/day]	No of cars filled	Supplied power [kW]	Waste heat [kW]
200	40	468.8	70.31
150	30	351.6	52.73
100	20	234.4	35.16
50	10	117.2	17.58
20	4	46.9	7.03

Table 3.13: Waste heat calculations for the hydrogen station based on NEL C-150 electrolyser. Energy share to waste heat is 15%.

Now that the operational data and boundary conditions for waste heat is set, the representation of indirect heat recovery solution in the simulation model can be presented. As shown by the principle drawing in figure 3.20, the waste heat is collected by the antifreeze circuit and transported to the space heating system of the building. Figure 3.21 below shows how the cooling loop fron the hydrogen station is represented in the Dymola model.

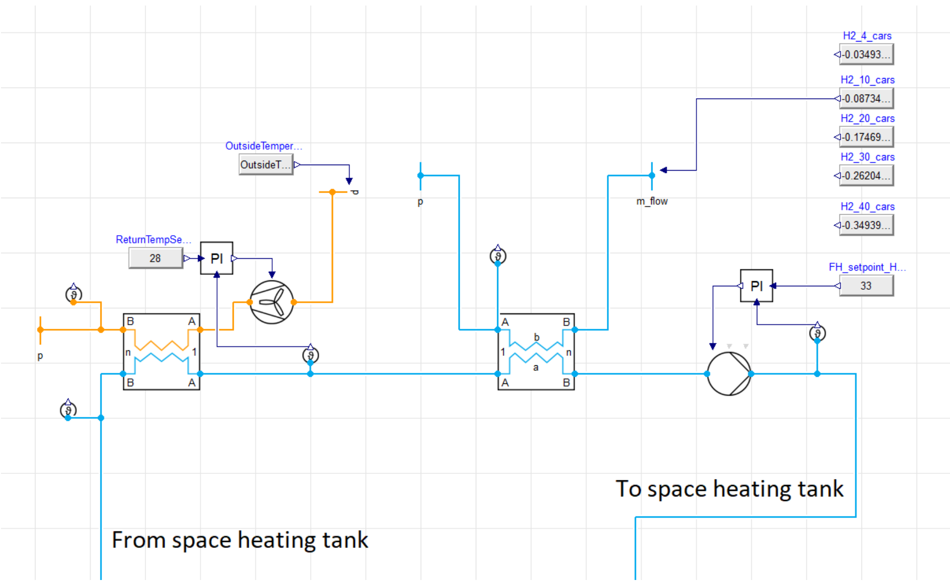

Figure 3.21: Representation of the indirect heat integration solution for the hydrogen station. Screenshot from Dymola.

In the simulation model the cooling load from the hydrogen station is represented by a mass flow of antifreeze transporting heat to a liquid-liquid plate heat exchanger in the space heating circuit. The water is heated from 28°C to the setpoint of 33°, and the output temperature is controlled by a variable speed drive pump. The PI controller regulates the volumetric flow rate to the correct set point temperature at different load, similar to the pumps in the heat recovery system in the supermarket. The supply temperature of antifreeze from the station is predicted to be in the range of 40-50°C by UNO-X, and a temperature of 45°C is chosen for this model. The waste heat energy load from the hydrogen station is calculated in watt according to the amount of hydrogen produced every day at the station. As explained in section 3.2.4, the heat flow is converted to a mass flow rate by using the enthalpy difference between supply and return temperature of the cooling circuit from the hydrogen station according to equation (3.5) below. The return temperature to the hydrogen station is assumed to be 5°C.

$$m_{flow} = \frac{Q_{Antifreeze}}{(h_{in} - h_{out})} \tag{3.5}$$

The enthalpies are found using the refrigerant library RnLib. Since the return temperature to the hydrogen station is lower than the supply temperature of the space heating system, some of the heat can not be recovered by this solution. An illustration of this is shown in figure 3.22.

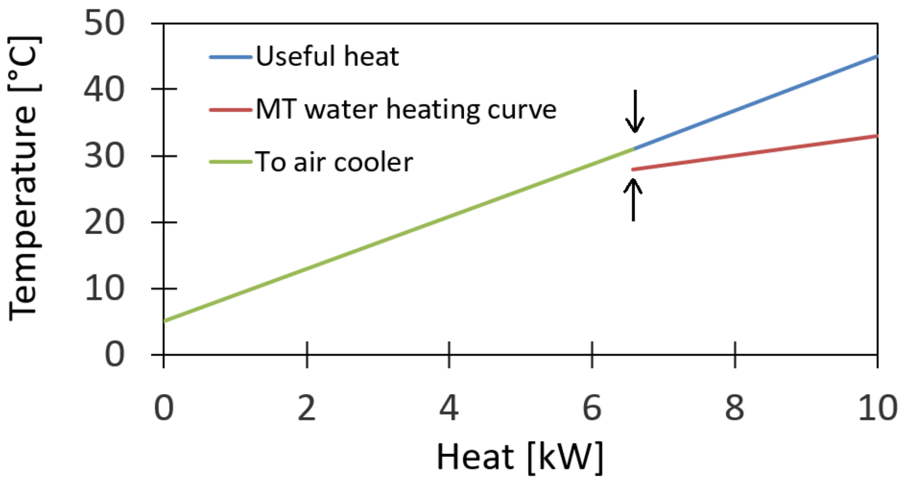

Figure 3.22: Heat recovery process by indirect integration of the hydrogen refueling station. Example of 10 kW waste heat in cooling stream.

The figure illustrates how much of the waste heat that can be recovered to the space heating system. Assuming a 3 Kelvin minimum temperature difference in the heat exchanger, the antifreeze is cooled from 45 to 31°C in the heat recovery heat exchanger. The pinch point, meaning the location of minimum driving forces for heat transfer, occurs at the cold end of heat exchanger. To cool the antifreeze down to 5°C, the rest of the heat must be given off by a dry cooler on the station, or used for other low temperature purposes. The energy that can be recovered to space heating accounts for about 35% of the total energy in the cooling circuit with the boundary conditions in this model. Snow and ice melting during winter has been mentioned as a possible application for the remaining low temperature heat in the hydrogen stream.

In the Dymola model a dry cooler is placed on the return line from the space heating system. This dry cooler ensures the return temperature from the space heating system is 28°C at all times, so that a five degree temperature increase is assured. When the space heating tank is fully charged and there is still more heat available for heat recovery, the return temperature from the space heating tank increases. When calculating the net heat recovery from the hydrogen station, the dry cooler duty on the water circuit is subtracted. The reason for this arrangement is to have a more stable operation for the simulation software, and to log the dry cooler duty to investigate the amount of energy rejected to ambient air. As a base case study, a daily hydrogen production of 50 kg is assumed, sufficient to supply 10 fuel cell cars equivalent to the Toyota Mirai with hydrogen fuel (5 kg tank capacity). The amount of hydrogen produced is assumed to be equal every day of the week, resulting in a steady state operation of the electrolyser and waste heat availability for seven consecutive days.

3.2.8 Case study 3: Hydrogen refueling station direct heat recovery

The main goal of this thesis is to integrate the different energy systems together to enable interaction between them. In section 2.2, the function and components in a hydrogen refueling stations was reviewed. The need for cooling of the hydrogen gas after compression and high pressure storage is evident. A refrigeration systems is therefore needed to satisfy the demands. The temperature levels involved varies from design to design, but a low temperature level of -40°C is required in all designs to satisfy standards for refueling times required by SAE-J2601. One can argue that the solution presented for case study 2 in section 3.2.7 is not fully integrated, where only the waste heat from the hydrogen station is integrated with the rest of the energy system in the building. A solution that is worth investigating is to couple the hydrogen refueling station with the supermarket refrigeration system. Since it already exists a refrigeration system on the building property in the form of a CO_2 system at the supermarket, it makes sense to couple the refrigeration demand at the refueling station with this system. The supermarket trans-critical CO_2 system can be modified so that cooling demands at different temperature levels at the refueling station can be satisfied. This requires some redesign of the hydrogen station. The principle of the combined hydrogen station - R744 Supermarket refrigeration system solution is presented in figure 3.23 below.

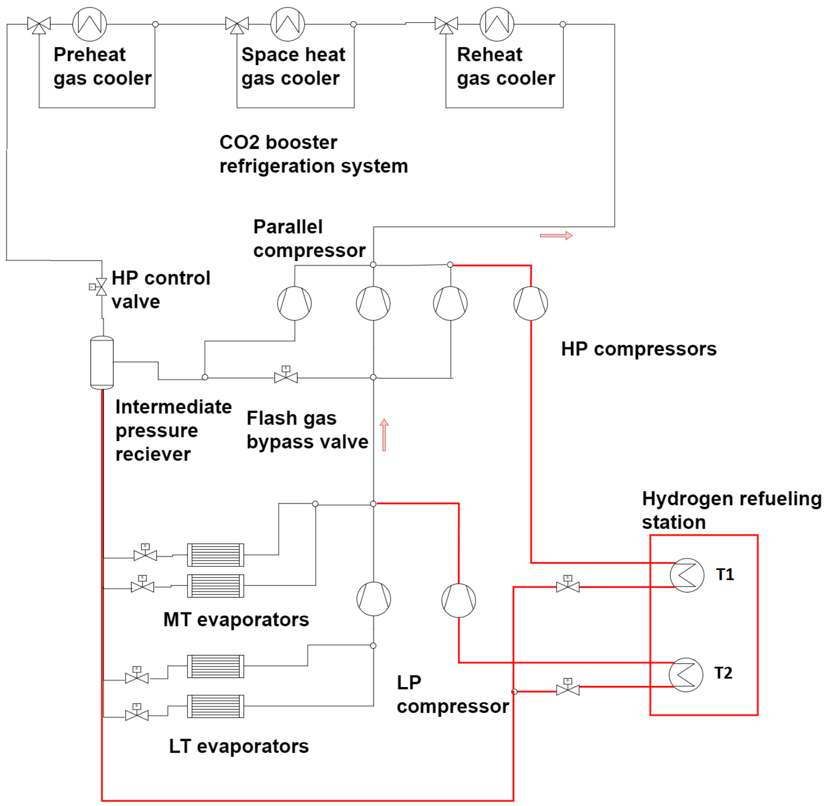

Figure 3.23: Principle solution for integrating a hydrogen refueling station to a CO_2 refrigeration system.

Instead of having a separate refrigeration system both at in the supermarket and at the hydrogen station, the station is redesigned to have connection to the CO_2 booster system. As illustrated by the simple system sketch in the figure above, liquid CO_2 at 40 bar is taken from the intermediate pressure receiver in the machine room of the supermarket and transported to the hydrogen refueling station on the outside of the building. The common liquid supply line is then separated into as many branches as is required, determined by the number of different temperature levels needed for refrigeration in the station. In figure 3.23, the principle is illustrated by having two temperature levels at the hydrogen station. The common liquid supply line is divided into two branches and throttled to different evaporation pressures. In the figure, temperature level T2 typically satisfies the cooling demand at -40°C, and temperature level T1 is a higher temperature level. When the liquid CO_2 refrigerant is evaporated at the the different pressure levels, the gas is sucked off by the compressor rack in the supermarket. The lowest temperature level can be compressed and delivered in the suction line of the high pressure compressor rack, while the gas from higher temperature levels in at the station can be compressed directly by the high pressure rack. The figure represents an example of a configuration that can be implemented, but the general idea is the same irrespective if the number of required temperature levels is one or four. The only design change is the number of return lines of evaporated CO_2 refrigerant back to the supermarket machine room.

As the principle solution has been explained from the supermarket refrigeration system side, the design is now also described from the perspective of the hydrogen refueling station. As described in the previous paragraph, the interface between the supermarket refrigeration system and the hydrogen fueling station is one liquid refrigerant line and a number of return lines with refrigerant vapor back to the refrigeration system. In this way the station is connected on the evaporator side of the CO_2 system, and acts as a load similarly to the MT and LT cabinets in the supermarket sales area. Based on the findings in the literature study, a design that can be implemented for integrating the fueling station and the rest of the energy system in the building is proposed. Figure 3.24 below shows the suggested design.

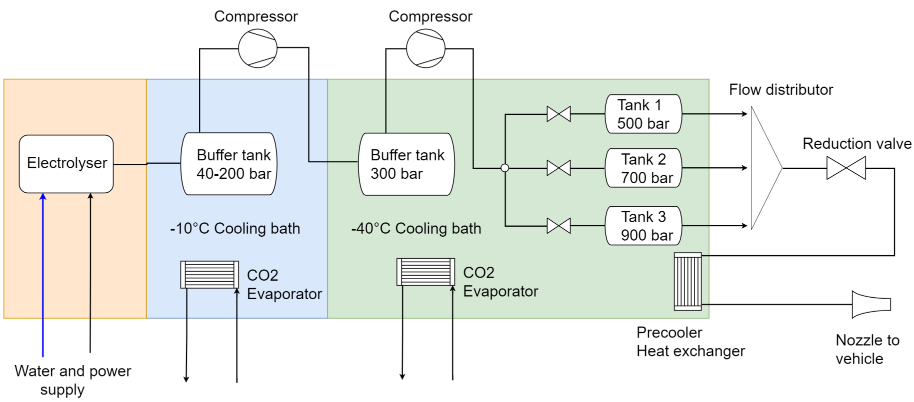

Figure 3.24: Design of the hydrogen station with CO_2 refrigeration system integrated.

The design of the refueling station inspired by the solution suggested by Richardson et al. (2015), where the high and medium pressure tanks for hydrogen are stored at -40°C instead of storing them at ambient temperature. The tanks are stored in a cooling bath of antifreeze, and liquid CO_2 from the supermarket refrigeration system is evaporated as it exchanges heat with the antifreeze from the cooling bath. This ensures the temperature of the cooling bath is kept at -40°C. In addition to the low temperature cooling bath, an equivalent level at -10°C is introduced. The electrolyser produces low pressure hydrogen, varying from 40 to 200 bar depending on the manufacturer. For this case study, it is assumed the outlet pressure of hydrogen produced at the electrolyser is 200 bar, and then compressed to 300 bar for intermediate storage in the buffer tank in figure 3.24. According to the the operational data for the electrolyser NEL C-150, the operational temperature is 80°C, meaning the hydrogen need cooling before and after compression to storage in the buffer tank at 300 bar. The cooling of hydrogen after the electrolyser and after compression to 300 bar is done at the -10°C level, which functions as an intermediate temperature level between the ambient temperature and the -40°C level.

Fuelling of the fuel cell vehicle is carried out as described by Rothuizen et al. (2013b) in section 2.2.2. The high pressure tanks 1, 2 and 3 are charged with hydrogen at different pressures. As the fueling process is initialized, hydrogen from tank 1 is throttled and let into the empty vehicle fuel tank with a mass flow rate to achieve the pressure ramp according to fueling protocol in standard SAE-J2601. As the pressure difference between the vehicle tank and tank 1 decreases, tank 2 is taking over supply of hydrogen. This cascade continuous to the filling is complete. A slight variation to this procedure is that after the pressure reduction valve, the hydrogen line goes through the cooling bath, ensuring the required precooled temperature is reached at outlet. It is assumed the high pressure tanks are at -40°C when fueling starts, and this reduces usual peak of precooling which observed when the high pressure tanks are stored at ambient temperature.

Now that the design of the station is described from the both sides of the energy system, the representation of the fully integrated hydrogen station solution in Dymola software follows. Figure 3.25 below shows a screenshot of the cooling system of the hydrogen fueling station in the simulation software.

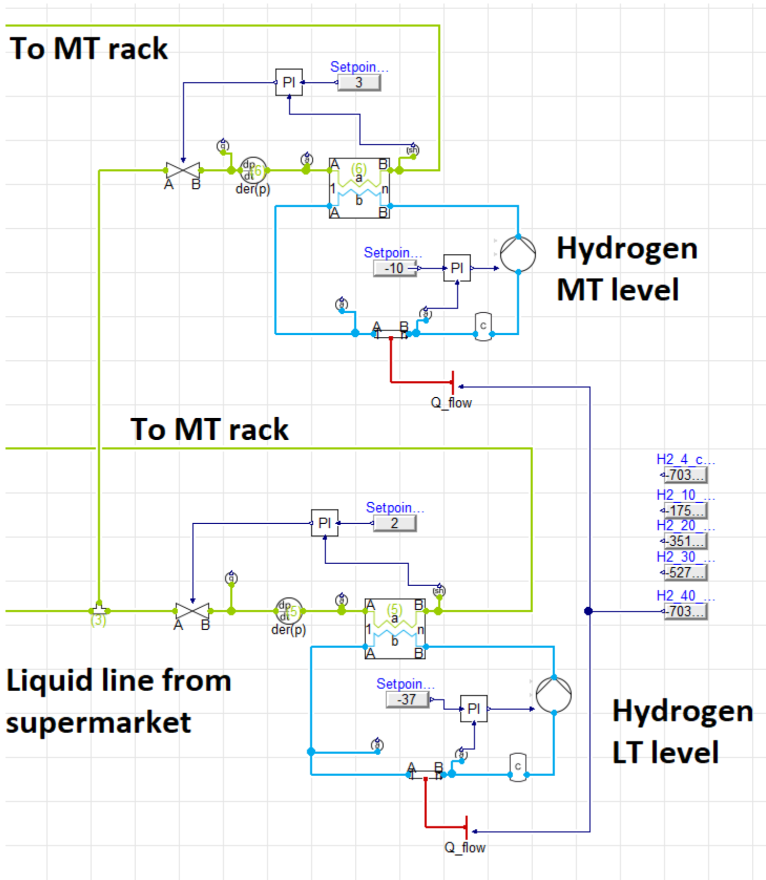

Figure 3.25: Representation of the hydrogen fueling station in the simulation model. Screenshot from Dymola.

As can be seen from the figure, two temperature levels are chosen for investigating the feasibility of integrating the hydrogen station with the supermarket refrigeration system. In the model, each temperature level consists of a liquid-refrigerant plate heat exchanger, thermostatic expansion valve, liquid pump and a liquid volume. The structure of the temperature levels in the hydrogen station is modelled very similarly to the MT and LT cabinets in the supermarket. A liquid tube is representing the volume of antifreeze in the cooling bath. A controller regulates the flow of antifreeze to the heat exchanger, and thereby controls the temperature in the cooling bath. The cooling load on each level is added through a heat port in the tube representing the cooling bath. A liquid line from the intermediate pressure receiver in the supermarket is delivered to the station and separated into two branches as described earlier. The liquid CO_2 is throttled to the required evaporation pressure by a thermal expansion valve, and evaporated. The valve controls the superheat at the heat exchanger outlet. The refrigerant gas leaves the station in two separate pipelines to the supermarket machine room by help of the suction created by the compressor rack. A deviation from the design is that the lowest temperature level is set to -37°C, since the refrigerant properties for antifreeze used in the model is only included down to -40°. During the initialization of the simulation model, the temperature has some fluctuations before reaching steady state. If the temperature of the antifreeze drops below -40°C, the thermodynamic properties of the model is no longer valid and the simulation may terminate. To avoid this from happening, the setpoint of the low temperature level is -37°C. In this way some small changes in the temperature of the antifreeze can be handled without entering the invalid region of the fluid library.

The compressor configuration for the model is shown i figure 3.26 below.

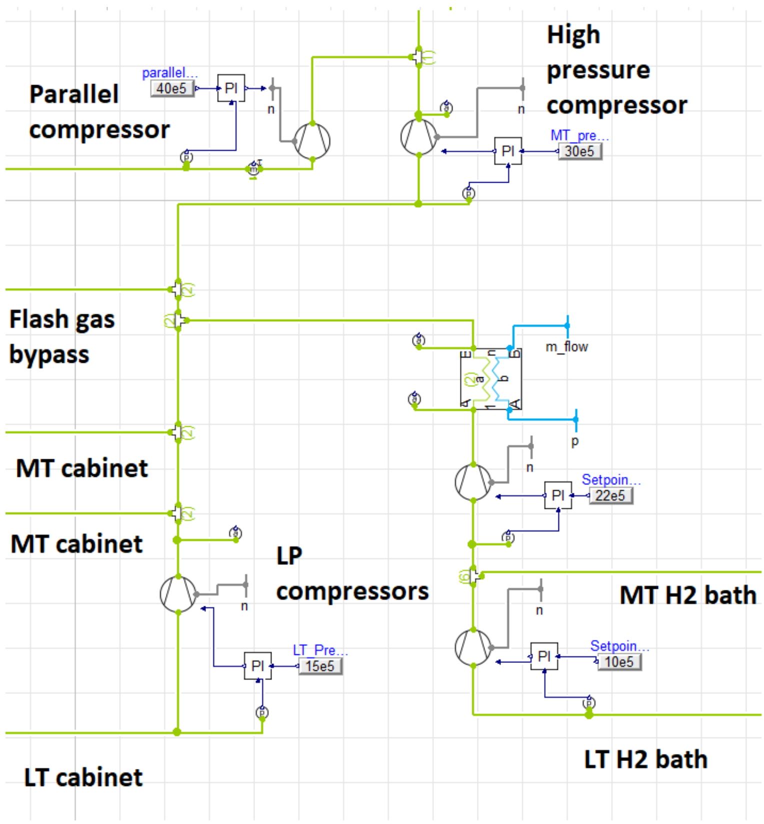

Figure 3.26: Representation of the compressor configuration for case 3 in the simulation model. Screenshot from Dymola.

The LT and MT suction lines from the hydrogen station is visible on the bottom right of the figure. The -40°C level has a suction pressure of 10 bar, while the -10°C level has a suction pressure of 22 bar. The pressure is raised to the evaporation pressure of the supermarket MT cabinets at 30 bar, and delivered in the suction line of the high pressure compressor rack. A desuperheater with cooling water at 5°C is used lower the temperature of the refrigerant before sending it to the suction line of the HP compressor rack. This avoids having a large superheat in the suction line of HP pressure rack, resulting in a lower discharge temperature. It is important to avoid excessive discharge temperature to ensure safe operation and avoid degradation of lubrication oil and seals in the compressor.

There is some uncertainty of the actual operational conditions of the hydrogen fueling station regarding the cooling loads, electrolyser operation and frequency of customers, as explained for case 2 in section 3.2.7. To get a reasonable comparison between case 2 and 3, the value for available waste heat calculated in table 3.13 is used also here. The waste heat available according to the table is in case 3 set as refrigeration load on the cooling baths at the station. Without specific data available, the refrigeration load is set to 50% at -10°C level and 50% at -40°C level. Although the waste heat calculations is based on the electrolyser operation only, typically associated with the -10°C level, refrigeration load is put on the low temperature level. One of the goals of the thesis is to show the feasibility of integrating the hydrogen station to the building energy system, and the cooling at -40°C is unavoidable if the fueling process is to reach fueling times required by SAE-J2601. Therefore the low temperature level is included in the model, so that the principle of the integration can be demonstrated. The operational conditions are assumed to be the same for case 3 as for case 2. This involves a base case study of 50 kg daily production of hydrogen, and continuous operation of the electrolyser as described in section 3.2.7.

Chapter 4

Results

In this chapter the results from the building simulation in SIMIEN and simulation models in Dymola are presented in their respective sections.

4.1 Building simulation

In this section the result from the building simulation is presented. The description and presentation of the building model developed in the software SIMIEN was done in section 3.1. Simulation for the building was run for a full year, resulting in a summary report for the year and an hour-by-hour log file where demand for heating, cooling, indoor and outdoor temperatures, solar irradiation and heating through the year is included among other parameters. This log file is a powerful tool to be used for investigating the dynamic behaviour of the integrated energy system. The log file can be utilized to set the requirement for heat delivery from the space heating system.

The first run of the building simulation showed very low demand for heating in the apartments. This lead to an investigation of the cause, resulting in a parametric study of the ventilation air flow rates in the building. According to the Norwegian Standard for passive and low-energy houses (Standard Norge, 2013a), the minimum required air flow rate is 1.2 m^3/m^2 per hour. This corresponds to an air exchange of 0.5 times per hour of the heated air volume in the building. This value was used in the first run, yielding very low heating demands. According to a report from SINTEF Byggforsk (Wigenstad et al., 2012), the minimum specifications given in NS3700 are too low compared to actual demands, and suggest several methods to calculate more realistic air exchange rates. The report suggests higher values based on more accurate calculations according to functional demands and layout in the building. For apartments of a 100 m^2 the authors recommend an air exchange of 1.5 times/hour (method 1), three times as large as the minimum specifications from NS3700. This results in a very large value for heating demands, even exceeding the maximum limit for heating for passive houses according to NS3700. The second method uses recommended required air flow rates for kitchen, bathroom and bedroom to obtain an average value somewhat higher than NS3700. The results of the parametric study is summarized in table 4.1 below. The values given in this table is for the apartments only, as the supermarket is simulated in a separate model.

Source	**Air flow rate [m^3/(m^2*hr)]**	**Heating demand [kWh/(m^2*yr)]**
NS3700	1.2	6.3
SINTEF Byggforsk (1)	3.6	23.4
SINTEF Byggforsk (2)	1.51	8.2
Arbitrary choice	2.5	14.9

Table 4.1: Effect of increasing ventilation air flow rates on heating demand in the apartments.

If the result is plotted as air flow rate versus heating demand, a linear dependency can be observed.

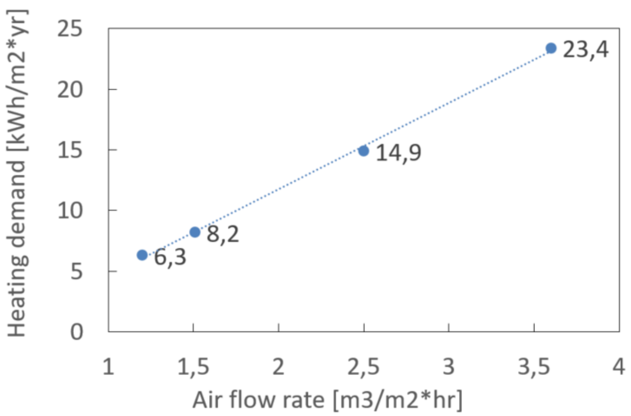

Figure 4.1: Effect on heating demand in the apartments by increasing air flow rates in ventilation.

For further use as input to the Dymola simulation models, the second method suggested by SINTEF Byggforsk is chosen to get a heating demand closer to a realistic value. It is important to point out that the specifications given in NS3700 are the *minimum* air flow rate to avoid moisture and fungus growth, and real life air supply rates are higher.

As mentioned in section 3.1, the simulation was carried out separately for the apartments and supermarket. The output files from the simulation software gives hour-by-hour requirements of energy in watt. The required energy per hour is given separately for heating of ventilation air and room heating. The output data was imported into a spreadsheet in Excel and the two categories was added together to one value, denoted as space heating. From the result file a power-duration curve for the whole year was constructed and is shown for the apartments in figure 4.2 and in figure 4.3 for the supermarket. In this plot the heating demand per hour is sorted from highest to lowest, and then plotted from left to right in the figure. The plot shows the heating characteristics of the building, and for how many hours per year the heating demand exceeds a specific duty.

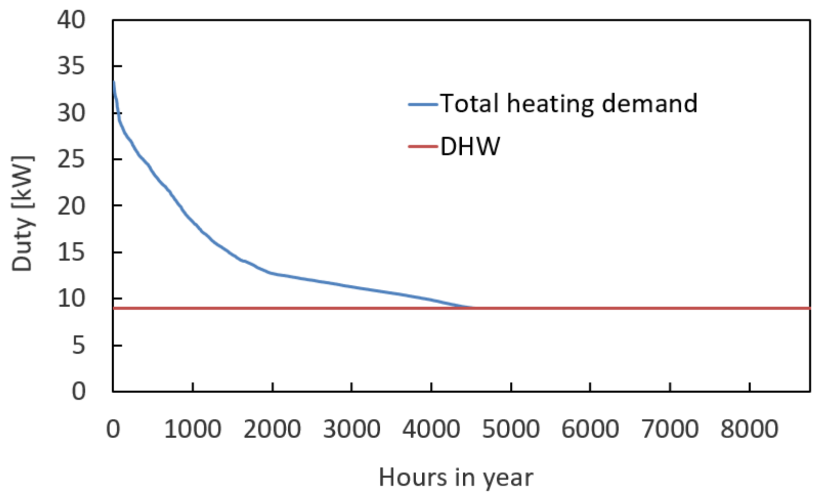

Figure 4.2: Power-duration curve for the apartments.

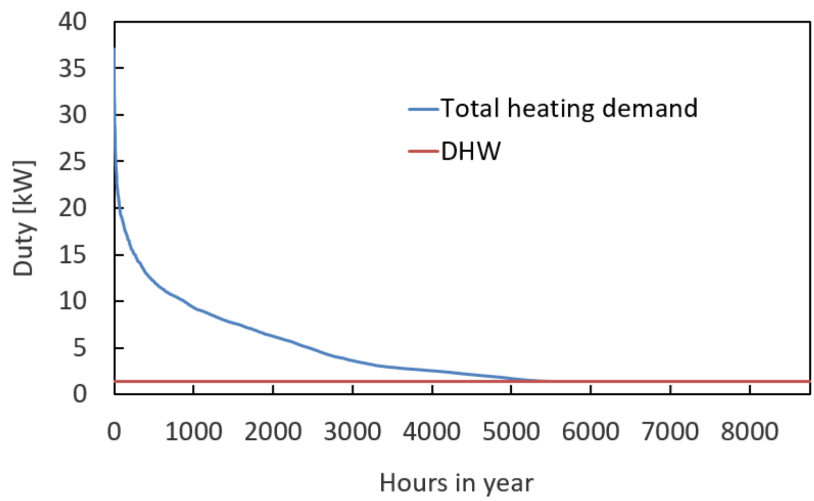

Figure 4.3: Power-duration curve for the supermarket.

A summary of the energy demands for space heating, domestic hot water and cooling is presented in table 4.2 below.

Demand	Apartments	Supermarket
Space heating	25 729 kWh	24 239 kWh
Domestic hot water	78 930 kWh	12 279 kWh
Share DHW	75.4%	33.6%
Cooling	18 kWh	44 kWh

Table 4.2: Summary of energy demands from building simulation in SIMIEN

The demand for space heating is added together for the supermarket and apartments and presented hour by hour in figure 4.4. The heating demand is in this figure organized in a consecutive manner, 1st of January on the left side of the figure and 31st of December on the right.

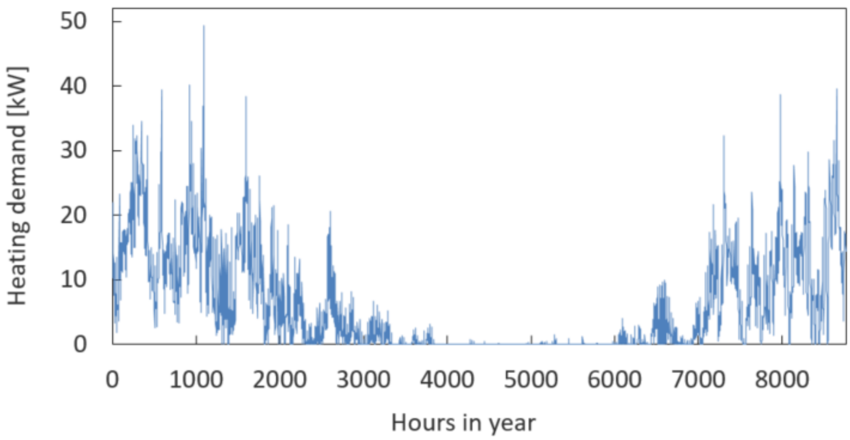

Figure 4.4: Result of hour-by-hour simulation in SIMIEN for the building.

As described section 3.1, four weeks is selected as a representation of the characteristic heating demands for each season in the year. Table 4.3 below shows the selected weeks representing winter, spring, summer and fall.

Season	Start of week	End of week
Winter	13th of January	19th of January
Spring	31st of March	6th of April
Summer	1st of July	7th of July
Fall	23rd of October	29th of October

Table 4.3: Selected weeks for simulation representing the full year.

The heating demands and corresponding outdoor temperature for the selected weeks are shown in figure 4.5, 4.6, 4.7 and 4.8 below.

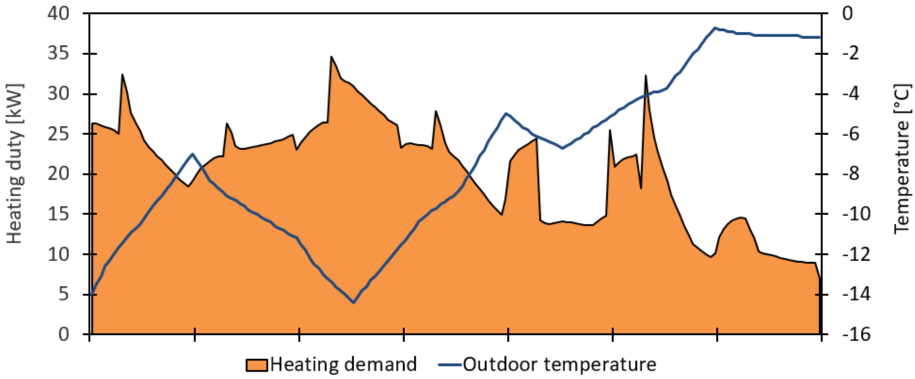

Figure 4.5: Heating demand and outdoor temperature during a winter week (13.01 to 19.01)

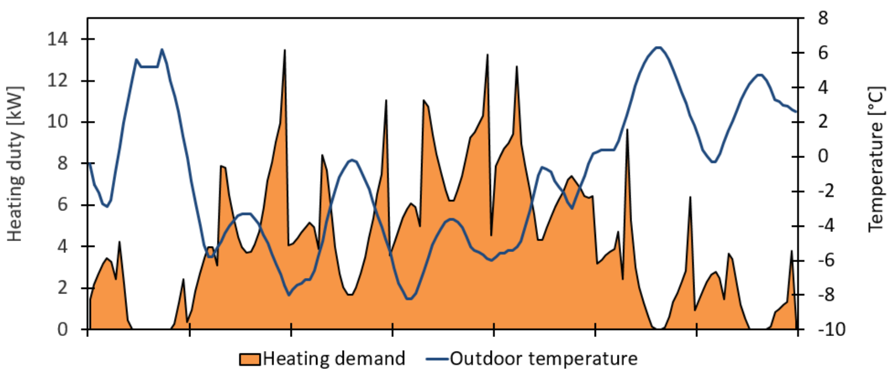

Figure 4.6: Heating demand and outdoor temperature during a spring week (31.03 to 06.04)

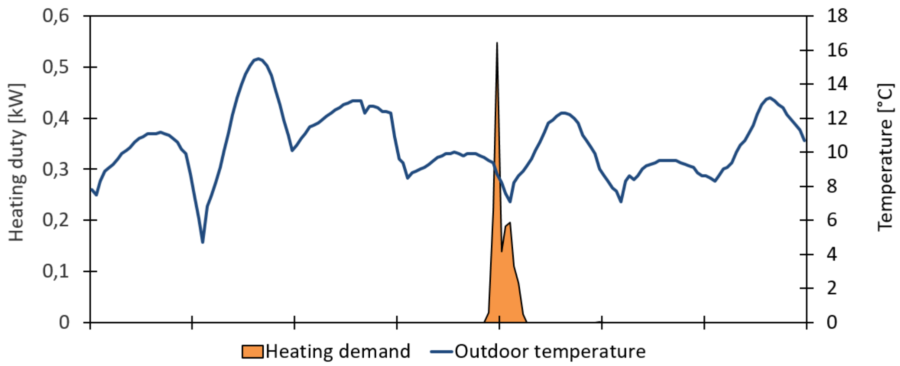

Figure 4.7: Heating demand and outdoor temperature during a summer week (01.07- 07.07)

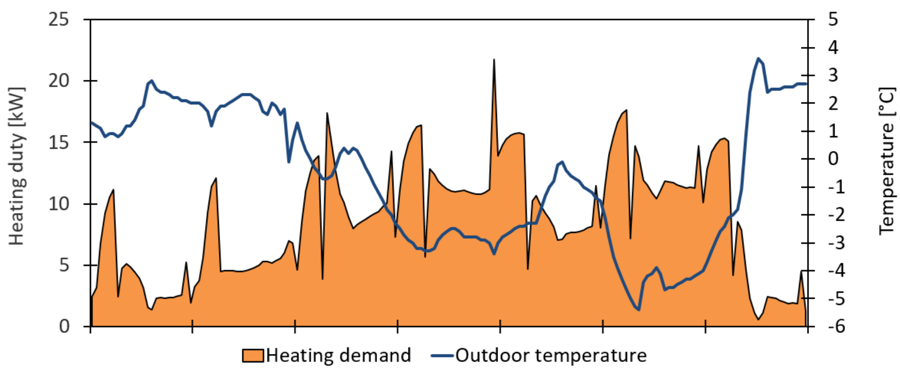

Figure 4.8: Heating demand and outdoor temperature during a fall week (23.10 to 29.10)

The presented heating demands and outdoor temperatures will serve as input to the dynamic simulation model of the integrated energy system in Dymola.

4.2 Dymola Simulations

In this section the results from the simulation in Dymola is presented. As described in chapter 3, three separate simulation models were developed in the software to investigate and evaluate the different options for heat recovery. With the results from the building simulation model as input for heating demands, simulations was carried out for the four weeks representing the four seasons through the year. This section aims to show the potential and differences for heat recovery by arranging the energy system in three different ways. The result file from Dymola simulation is large and contains state points for every in and outlet port of every component contained in the model. Heat exchanger duties, compressor shaft power, volume and mass flow through components are also included. For this thesis, mainly results concerning the heat exchanger duties are of interest in addition to observing that controlled states reaches their setpoint. The result section will mainly focus on duties concerning heat recovery and the associated heat exchangers.

4.2.1 Space heating and domestic hot water heat recovery

Four weeks of simulation where carried out for all three models of the system, representing the performance of the systems in winter, spring, summer and fall. Figure 4.9 below shows how much of the heating demand that is satisfied by heat recovery in each simulation scenario divided by seasons. In this first comparison between the cases a daily production of 50 kg of hydrogen is assumed, enough to refill 10 cars equivalent to the Toyota Mirai (5 kg tank capacity).

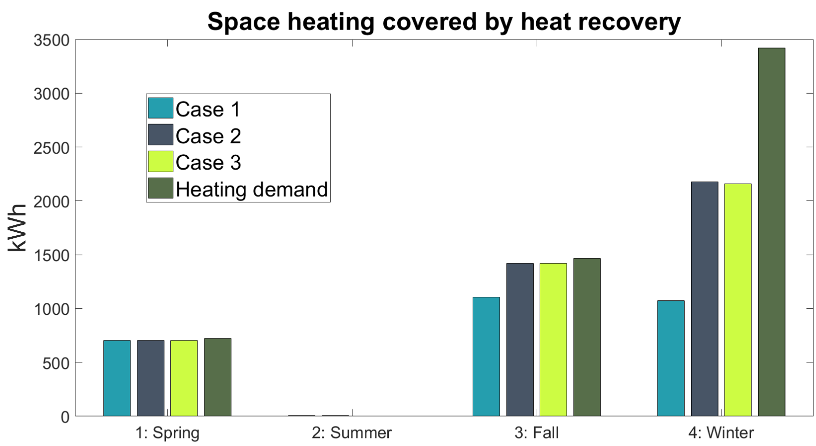

Figure 4.9: Comparison of net heat recovery between case 1, 2 and 3 for all seasons.

Figure 4.10 illustrates the same results, but in percent of heating demand covered by heat recovery for each season.

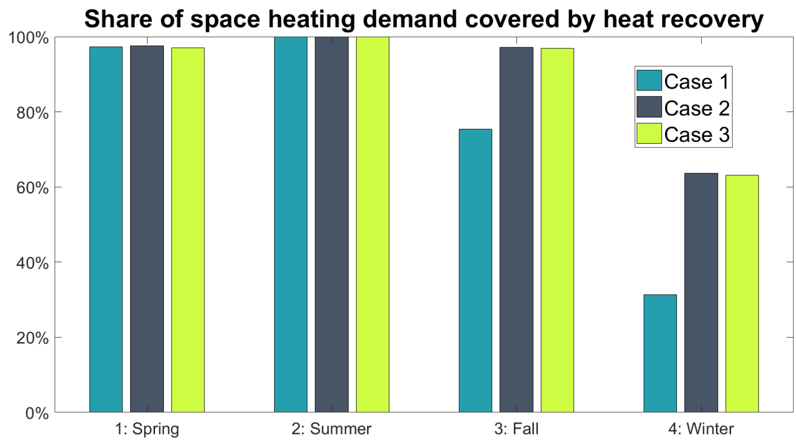

Figure 4.10: Share of net heating demand satisfied by heat recovery in case 1, 2 and 3 for all seasons.

Figure 4.11 below shows what heat source is covering the energy demand in each of the simulation cases for the four weeks. In case 3 the heat recovered due to refrigeration load at the hydrogen station and supermarket is commonly referred to as supermarket gas coolers, because the heat recovery is measured there. And since the refrigeration load at the hydrogen station is taken care of by the supermarket CO_2 booster system, the heat is delivered at the supermarket gas coolers.

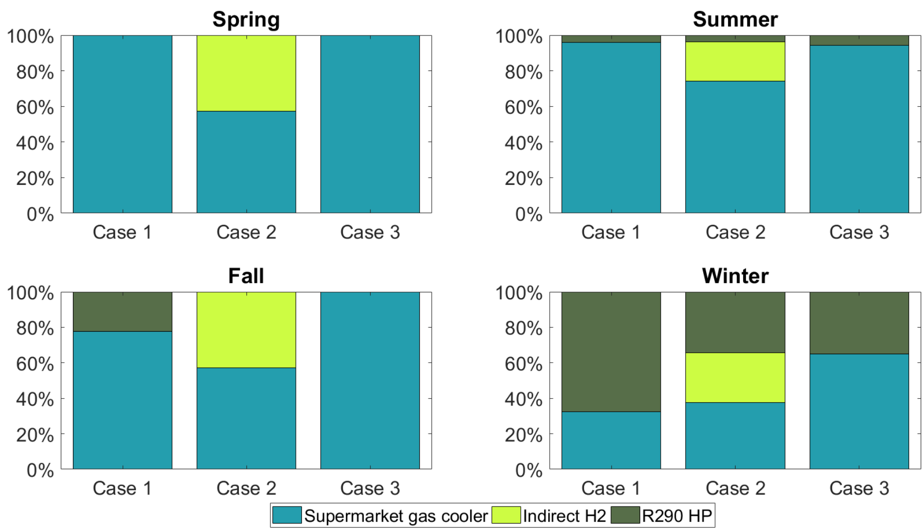

Figure 4.11: Comparison of heat source distribution for simulation case 1, 2 and 3 divided by season.

The heat recovered to domestic hot water from the supermarket reheat and preheat gas coolers is shown for each season in figure 4.12 below.

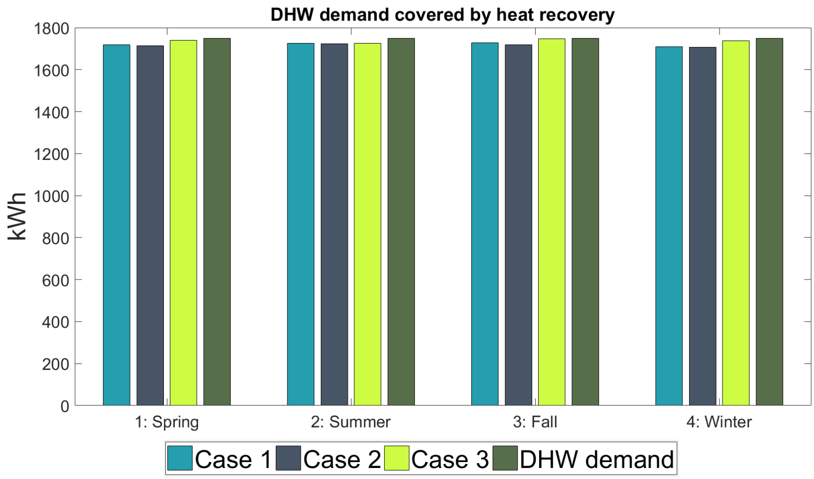

Figure 4.12: Comparison of heat recovery to DHW for simulation case 1, 2 and 3 divided by season.

The share of domestic hot water covered by heat recovery from the supermarket gas coolers are shown in figure 4.13 below.

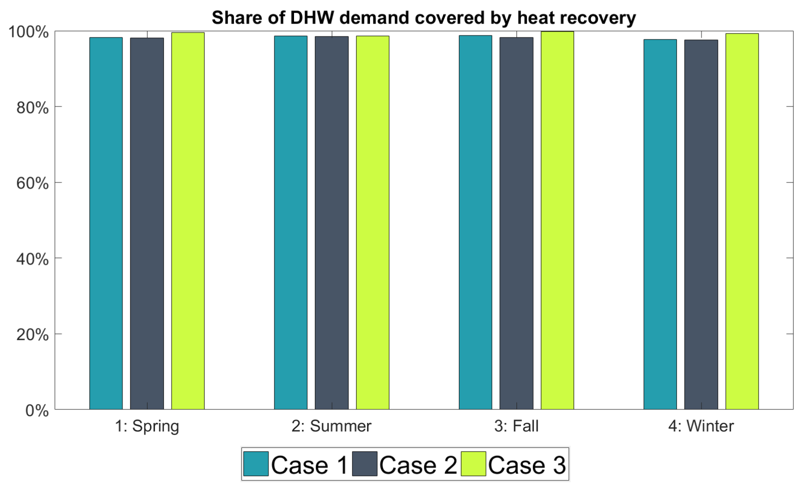

Figure 4.13: Share of domestic hot water demand covered by heat recovery.

4.2.2 Hydrogen production study

There is uncertainty of the demand for hydrogen fuel at the location of the building. Therefore the production level of hydrogen per day is investigated further. Since winter week is the week of largest energy demand, the investigations of hydrogen production is done only with the winter week as boundary conditions. Table 3.13 gives the waste heat available for scenarios with production enough to supply 4, 10, 20, 30 and 40 with hydrogen fuel daily. This corresponds to 20, 50, 100, 150 and 200 kg of hydrogen production per day. The case of no production of hydrogen corresponds to the results from Case 1 where only the supermarket included in the heat recovery system. Figure 4.14 shows the effect of increasing the hydrogen production on heat recovery to space heating for case 2.

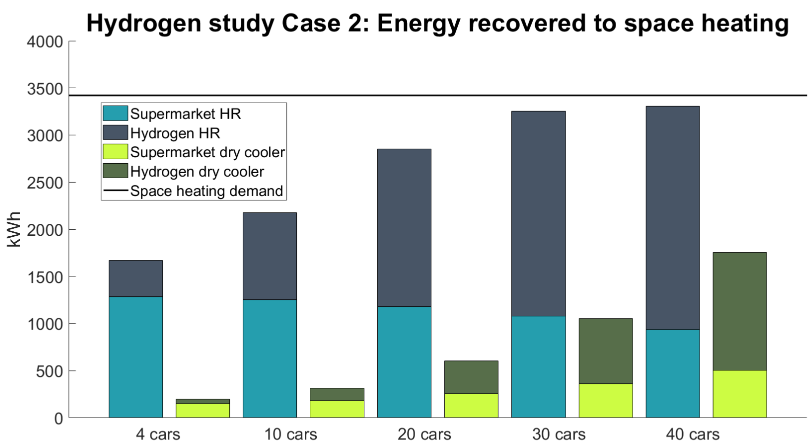

Figure 4.14: Effect of increasing the daily production of hydrogen on heat recovery to space heating in Case 2.

Since the integration of the hydrogen station in case 2 only has impact on heat recovery to the space heating system, the heat recovered to domestic hot water is the same as for case 1.

In case 3 the hydrogen station is fully integrated to the supermarket refrigeration system and increases the heating duty of all three gas coolers, meaning increasing the daily production of hydrogen will influence the heat recovery to both domestic hot water and space heating. Figure 4.15 below shows the effect on space heating recovery for Case 3.

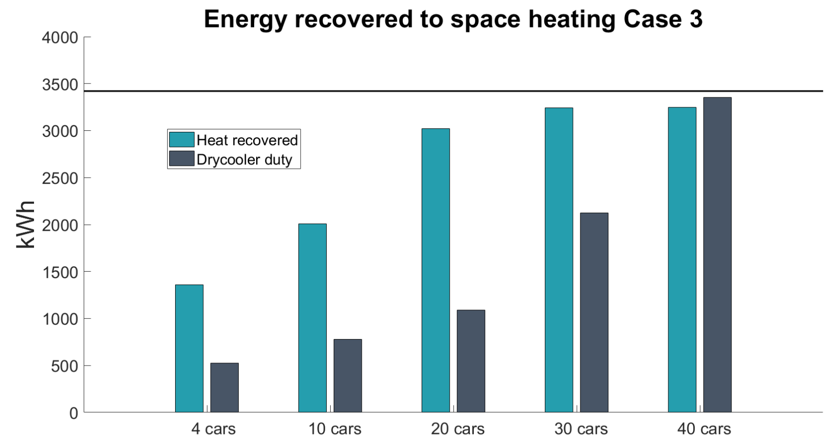

Figure 4.15: Effect of increasing the daily production of hydrogen on heat recovery to space heating in Case 3.

Figure 4.16 below shows the effect on domestic hot water heat recovery for Case 3.

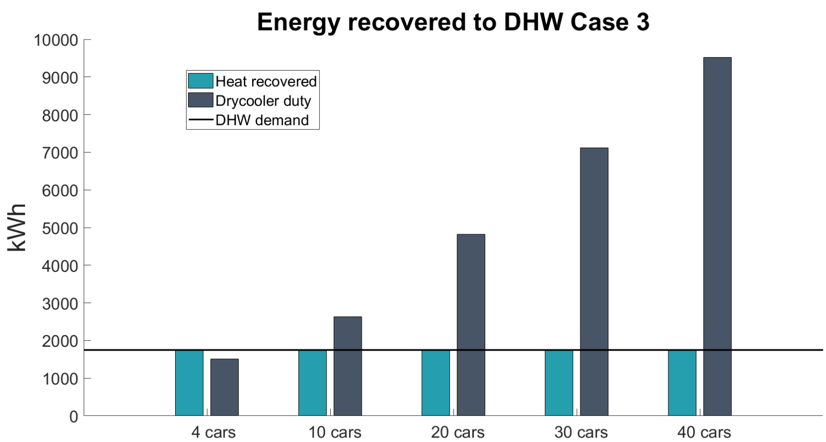

Figure 4.16: Effect of increasing the daily production of hydrogen on heat recovery to domestic hot water in Case 3.

Chapter 5

Discussion

5.1 Building simulation

From the power-duration curves in figure 4.2 and 4.3, the difference between residential and non-residential buildings in distribution of heating demand is clearly observed. The power curves for the supermarket and apartments has similar value for required heat at the highest peak. However, the power demand for the supermarket decays faster than for the apartments. For the apartments the power demand is on average higher due to more hours with higher requirements. The requirement for domestic hot water is responsible for a substantial share of the load on the heating system in the apartment building model. This is in accordance with the findings in section 2.3, where the share of DHW to the total energy demand heating is increasing when the building norm followed is stricter. For the block of flats considered in this thesis, domestic hot water represent approximately 75.4% of the total energy demand through the year. This share of energy for DHW is typically lower based on measurements, because passive houses tend to use more energy for space heating than is estimated by building simulations and calculations (Rohdin et al., 2014). User control over temperature set point, non-air tight building envelope and user behaviour has been pointed out as causes for higher measured energy consumption than predicted by simulations. For this thesis, the result for space heating demand from SIMIEN is considered to be a acceptable foundation for further analysis and input to the other simulation models in Dyomla.

Because of the location of the building there is not much need for cooling, not even during summertime. For this reason, the cooling demands were not considered further for the modeling in Dymola. For the supermarket, which falls under the category of commercial buildings, the share for domestic hot water is much lower and only represents approximately 33.6% of the total energy demand. This value is somewhat higher than expected for a supermarket of this type. The supermarket considered in this thesis is without a counter for preparing fresh fish, meat and dishes. When the supermarket contains a kitchen preparing the goods mentioned, the demand for domestic hot water increases, and is assumed to be in the range which is predicted by the building simulation. Even though the supermarket is missing the kitchen for preparing fresh goods, the results from the building simulation is assumed to be acceptable as further input to the simulation models in Dymola. The heating season is somewhat longer for the supermarket compared to the apartments, which can be explained by a higher required ventilation air flow rate. After the outdoor air has been heated by the ventilation heat recovery unit, the air needs to be heated further in ventilation batteries. The higher required air flow rate of commercial buildings compared to residential buildings extends the heating season. As for the cooling demand, it is also small for the supermarket and not considered further.

Since the scope of the project is investigate the possibility for a centralized energy system, the demand for space heating for the supermarket and apartments was added together and represent a collective energy demand at a common temperature level. The same procedure is done for domestic hot water, and both used as input for energy demand to the integrated system model in Dymola. Having only one input for space heating and one input for domestic hot water simplifies the simulation models in Dymola. The total demand for space heating in the building through the year in figure 4.4 shows that large peaks occurs for some few hours during wintertime. During summer the demand for space heating drops significantly, and for some hours of the year the internal gains (people, light and equipment) of the building and solar irradiation is enough to cover the heating demand. This was one of the fundamental ideas of the passive house when the concept was introduced.

The selected weeks that represent each of the seasons in the year show different heating characteristics. This will test the simulation models for different behaviour of the heating demand curve, which is of interest. The winter week shows high and fairly stable heating demand, which decays towards the end of the week due to higher outdoor temperature. The spring and fall week show similar behaviour, with the highest heating demand during the fall week. The demand for space heating fluctuates in both weeks, ranging from near zero to peaks of 12 kW during the spring week. This puts the heating system in the simulation model in a more dynamic behaviour than the steady state-like behaviour of the winter week. During the summer week, there is almost no demand for heating, just during some few hours of the night between day four and five. The selected weeks are assumed to give a sufficient representation of the behaviour of the system during the year due to the differences in the heating characteristics observed from the simulation in SIMIEN.

5.2 Dymola Simulations

5.2.1 Space heating and domestic hot water heat recovery

Analyzing the results from the simulation starts with looking into the base scenario, where all three simulation models were tested for all four weeks. For simulation model 2 and 3, where the hydrogen refueling station was integrated in two different ways, the daily production of hydrogen was set to 50 kg. This corresponds to filling the tanks of ten fuel cell vehicles equivalent to the Toyota Mirai every day during the week. From figure 4.9 it can be seen that the demand for space heating is very different between the four weeks. As a result, there is large variation in the net heat recovery for each week. Figure 4.10 shows the same results, but in percentage of heating demand satisfied by heat recovery. Starting with the results for the summer week, there is very few kWh required for heating. As expected, the heating demand can be satisfied by heat recovery for case 1, 2 and 3. Since the heating demand can be covered, it is more interesting to analyze the results obtained for the other three weeks of simulation.

The first option of heat integration of the system is represented by case 1. From figure 4.9 and 4.10 it can be observed that whole heating demand can be satisfied by supermarket heat recovery alone during the spring and summer week. This means that integrating the hydrogen station in either of the two alternatives will not increase the net heat recovery to space heating further. This is because during these during these weeks, the demand is already satisfied by the heat recovery from the supermarket alone. For the fall and winter weeks the heating demand in the building is significantly larger, and as a result the heat recovery from the supermarket is not sufficient to cover the full demand for space heating. The heat recovery in case 1 is covering about 75.4% and 31.4% of the energy demand for fall and winter week, respectively. This shows that carrying out heat recovery for the combined supermarket-apartment building is a good solution for covering the base demand for space heating. For the fall and winter week the effect of integrating the hydrogen station in case 2 and 3 becomes evident. For the week simulated during fall, both the indirect and direct integration of the hydrogen station delivers sufficient heat to the space heating system to fully cover the demand. From the figure it can be seen that the percentage values are slightly under 100%, and the reason was found to be the initialization process of the simulation. During this transient conditions there are some small temperature fluctuations in the space heating system, and as a result the measure delivered duty is slightly lower than the required demand given by the SIMIEN simulations.

The simulation results for the winter week is probably the most interesting, as the large space heating demand makes the differences between the three cases more distinctive. Heat recovery from the supermarket can cover about 31.4% of the total space heating demand in the building. In case 2, where heat is delivered indirectly by a antifreeze circuit from the station, a total share of 61% of the space heating demand can be covered by heat recovery. Direct integration of the hydrogen refueling station to the supermarket refrigeration system in Case 3 achieves a total of 58%. The reason for the slightly lower coverage in case 3 is as follows. In case 3 the cooling load at the refueling station taken care of by the trans-critical CO_2 system, and thereby rejecting the heat at all three gas coolers. Approximately 34% of the total gas cooler duty is given off to the space heating circuit. The additional load on the refrigeration system due to the integration of the hydrogen station is delivered as heating both to space heating and domestic hot water. Meanwhile in case 2, the indirect heat recovery is done to the space heating circuit only. However, only heat from cooling the antifreeze from 45°C to approximately 31°C can be recovered. This accounts for about 37% of the total heat available in the stream, since the total temperature glide of the antifreeze is from 45°C to 5°C.

Taking a closer look on what heat source that is providing the required heating in the building gives more in-depth information regarding the heat recovery source and energy flows. From figure 4.11 the distribution of heat from energy sources can be observed. The heat sources displayed in the figure is the space heating gas cooler in the supermarket, the hydrogen cooling circuit heat exchanger in case 2 and the condenser for the R290 propane heat pump. For case 3, the heat delivered at the supermarket gas cooler is the sum resulting from the refrigeration load both at the supermarket and the hydrogen station, since the heat is delivered at a common gas cooler. For all cases, the heat that is not covered by heat recovery is supplied by the R290 heat pump. For case 1, the share of the heat pump is increasing with the heating demand in the building from summer to spring, fall and winter. As mentioned before, the heat recovery covers about one third of the heating demand. From the simulation results it found that the space heating gas cooler provides approximately 8 kW heating on base refrigeration load during night and peaks of 10-12 kW during high load hours during the day. This is beneficial when sizing the R290 heat pump, and a capacity of 25 kW was found sufficient to cover the remaining heating demand and maintaining the supply temperature of 33°C. A rule of thumb for dimensioning heat pumps for space heating is setting maximum capacity to 50-60% of power requirement for heating during the coldest day of the year. For residential buildings this will typically correspond to a energy coverage of 90-95% through the year. Having covered a base capacity by heat recovery from the supermarket gas cooler, the maximum capacity of the heat pump can be reduced to 25 kW for this thesis. The highest peak in the winter week simulated in this thesis is about 35 kW. From figure 4.4, some higher peaks than 35 kW is observed, and it is assumed that these peaks can be covered by electric boiler backup in one of the space heating tanks.

In case 2 the source of heat can be separated further as the heat exchanger taking heat from the hydrogen station cooling circuit is introduced. An interesting observation is that even though heat recovery from supermarket alone can cover the heating demand for spring and summer, some of the heating demand is coming from the hydrogen loop heat exchanger in case 2. This is because there is no hierarchy where the heat is recovered first and what dry cooler is shut off first in the model. Both systems deliver heat and rejects heat at their respective dry cooler in parallel, hence the net heat from the hydrogen cooling loop is visible also for spring and summer weeks. For the fall week, the hydrogen cooling loop provides a large share of the heating demand with 42.8%. In the winter week the indirect hydrogen heat exchanger provides about 27.8% of the total demand.

The demand of domestic hot water represent a significant share of the heating demand of the building since a large part of the building consists of apartments. Therefore it is of great interest to investigate if the heat recovery system to domestic hot water can cover parts of the energy requirement of this demand. Figure 4.12 shows the energy recovered to domestic hot water for case 1, 2 and 3 for the four weeks of simulation. Figure 4.13 shows the same results in percentage of heating demand to DHW covered by heat recovery. The two figures show that the full energy demand for domestic hot water can be covered in case 1 for all seasons. As a consequence, the demand can also be satisfied in case 2 and 3, since the supermarket reheat and preheat gas coolers is also included in case 2 and 3. The result is a consequence of the chosen operational conditions for the supermarket and assumption for domestic hot water consumption. The refrigeration load profile in the supermarket is assumed to be equal every day of the year, irrespective of the outside temperature. The consumption of domestic hot water is assumed to follow the values and schedule given in the technical document SN/TS 3031 from Norsk Standard. This involves having the same demand for domestic hot water every day of the year, resulting in the same results for all four weeks of simulation. The small variations from case to case are found to be due to the initialization process of the simulation, and resulting in small temperature fluctuations. Since all the energy required for supplying domestic hot water to the building can be covered by heat recovery from the CO_2 refrigeration system, the energy saving are substantial. By keeping the system in trans-critical operation all year, heat recovery from the supermarket can save about 91200 kWh in energy costs yearly compared to supplying the DHW with an electric boiler. In addition, heat recovery from the supermarket can also satisfy some of the demand for space heating. Carrying out heat recovery from the CO_2 refrigeration system is strongly advised, as large energy savings can be achieved for the cost of two extra gas coolers and additional energy to run the system in trans-critical mode.

5.2.2 Hydrogen production study

The base case scenario of simulations where carried out with a daily production of 50 kg of hydrogen at the hydrogen station, equivalent to supplying 10 fuel cell vehicles with full tank. For a week this corresponds to 350 kg or 70 vehicle tanks of hydrogen fuel. The uncertainty in the daily production of hydrogen is high, and is very dependent on the popularity and sales of hydrogen fuel cell vehicles in the area of the station. One factor that can secure a stable demand for hydrogen fuel is replacing some of the fossil fuelled buses in the public transport network with fuel cell driven buses. EU funded projects has been carried out in several European cities, with good results (Fuel Cell and Hydrogen Joint Undertaking, 2016). Experiences from these projects show low noise level and cleaner city air as two of the advantages of using fuel cell buses. This type of arrangement would secure stable operation and production level for a hydrogen station like the one considered in this thesis. However, this is not set as a operational condition for the station in this thesis. It it assumed that the station delivers hydrogen fuel to private fuel cell cars in the area. To investigate the effect of varying the demand for hydrogen fuel from the station on the heat recovery in the building, five scenarios was simulated for case 2 and 3. For this hydrogen production study, the production was set to 20, 50, 100, 150 and 200 kg of hydrogen daily for a week. The details of the numbers are found in table 3.13. The winter week is used as boundary condition for the hydrogen production study, since the heating for space heating is highest for this week.

Starting with Case 2, figure 4.14 shows the the heat recovered to space heating and the duty of the dry coolers for each production level of hydrogen. The figure also show where the heat is recovered and is rejected. As expected, the heat recovery from the hydrogen cooling loop increases as the production level increases. An interesting observation is that the there is a small dry cooler duty also for the cases where not all heat is supplied by heat recovery. The reason for this is partly the demand curve for the winter week and partly the control of the system. The heating demand is higher during the first half of the week and falls towards the end of the week, as can be seen in figure 4.5. The heat recovery may not cover all heating demand during the first part of the week, but as heating demand falls there may be excess waste heat available in the system. This offset between supply and demand of heat results in some heat rejection despite having a storage of 5 m^3 of water. The other factor that contributes to dry cooler duty is the on-off control of the R290 heat pump. The heat pump is turned on when the temperature at the bottom of the space heating tank drops below 28°C, and stops when the bottom temperature reaches the set point temperature of 33°C. When the heat pump is on and charges the tank, the return temperature to the supermarket space heating gas cooler and hydrogen cooling circuit heat exchanger rises. Since the dry coolers connected to these two systems is set to always secure a inlet temperature of 28°to the heat exchangers, some heat is rejected during times when the heat pump is on. A better control system turning off the dry cooler connected to the hydrogen loop when the heat pump is on is advised for future investigations. For the supermarket, the heat needs to be rejected anyway, to secure stable operation of the refrigeration system.

As the production level of hydrogen increases the percentage of the heating demand covered by heat recovery increases. According to figure 4.14, a production level of hydrogen to serve 30 and 40 cars daily is sufficient to cover 95% and 97% of the heating demand during the week. This corresponds to a total production of 1050 and 1400 kg of hydrogen during the week. This will most likely not be very realistic in the first time after implementing this station during the phase-in period of fuel cell vehicles, unless a fuel cell bus program is established in connection to the establishing the hydrogen station. Since the integration in case 2 is only supplying heat to the space heating system, it has no effect on the heat recovery to domestic hot water in the energy system and is therefore not investigated further for case 2.

For case 3 the same behaviour is observed as for case 2. The share of heating covered by heat recovery is increasing with the daily production of hydrogen. A difference from case 2 is that the dry cooler duty is higher. This is most likely because the refrigeration load in case 3 is set equal to the waste heat available in the antifreeze stream from case 2. In case 3 the heat from compressor work is added to the refrigerant, as the compressors are assumed to be without heat loss. When bringing the heat from 10 bar evaporation pressure on the lowest level to 85 bar rejection pressure, heat is added on all compressor levels. This is true even there is a desuperheater heat exchanger after the MT level compressor on the hydrogen station suction line. Since the desuperheating process is done with cooling water of 5°C, the CO_2 is cooled down to approximate 8°C. Saturation temperature at 30 bar is -5.5°C, corresponding to a remaining superheat of 13.5 K after the desuperheater. This extra heat is carried through the refrigeration system and eventually rejected at the gas coolers, resulting in a increased dry cooler duty in case 3 compared to case 2. As for case 2, once the production of hydrogen can supply 30 cars or more, most of the heating demand in the building can be covered by heat recovery.

Since the refrigeration load from the hydrogen station is delivered to both the domestic hot water and space heating circuit in case 3, the result also affects the domestic hot water heat recovery. Analyzing the results for the hydrogen production study in figure 4.16

shows interesting values. From the base case study it was found that heat recovery from the supermarket refrigeration load alone could cover the need for DHW in the building. The heat rejection in the gas coolers runs with static heat rejection pressure of 85 bar, even if the heating demand is satisfied. This enables investigating the unused potential of heat available to domestic hot water. This shown in 4.16, where the dry cooler duty increases as more load is put on the refrigeration system by increasing the production level at the hydrogen station. With a daily production of 20 kg/4 cars, the duty of the dry cooler in the DHW circuit is approximately 86% of the net heat recovery. Already with the lowest production volume considered in this study, there is a large unused potential for DHW production. The dry cooler duty increases with the production level of hydrogen, and becomes very large for production levels close to maximum production of 200 kg/day. Table 5.1 below show the details from the study.

Production [kg/day]	**No of cars/day**	**DHW dry cooler energy [kWh]**
20	4	1507.8
50	10	2634.3
100	20	4822.3
150	30	7121.8
200	40	9520.8

Table 5.1: Energy rejected in DHW dry cooler for different production levels of hydrogen.

The supermarket and apartments have a weekly energy demand of approximately 1750 kWh for production of DHW, where the apartments is responsible for the greatest share. If the real life production levels of hydrogen is seen in the high end of the range presented here, there is a definitive potential for hot water production. With the current data, producing constantly on maximum capacity of 200 kg/day, can supply DHW to more than five copies of the same building as considered in the thesis. If there is a reliable demand for hydrogen fuel in the area where the station is located, it could be worth investigating the possibility to supply DHW to some of the neighbouring buildings. The potential is large for supplying residential buildings with hot water and making use of it, since the demand for DHW in residential buildings are fairly constant through the year.

Since there is a fuelling station on the property, it would have been a good idea to build and integrate a car wash with the rest of the energy system. Meratizaman et al. (2014) suggested this solution for a solar cell driven refueling station on a car park. Car washes use a lot of hot water, and supplying the hot water with heat rejection from the CO_2 refrigeration system ensures good waste heat utilization and reduced operational costs for the car wash. Including this feature in addition to having a supermarket and hydrogen refueling station, will make the site more attractive to customers. Domestic hot water can also be supplied to other users, depending on what is located close to the building property. Schools, ball parks, swimming halls or sports facilities all have a substantial demand for DHW and are all candidates for delivering hot water to. As stated, this depends on what type of buildings that are close to the building plot considered in this thesis.

5.2.3 System configuration evaluation

The chosen design for the three variations of the system has been simulated for four weeks, including a hydrogen production study for case 2 and 3. A discussion on system configurations is in order to evaluate the choices made during the design process based on the results of the simulation.

The aim of the boreholes is to function as a seasonal storage of heat. The wells serves as a heat source during wintertime and heat is taken from them. When the building has need for cooling during summer, heat is rejected to the energy wells from the ventilation cooling batteries. This shift between heat release and uptake in the wells is what defines the seasonal storage principle. Heat is stored in the wells during summertime, and used during wintertime. As a consequence, the temperature of the antifreeze in the boreholes will change from heating season to cooling season. The building simulation results yielded very small cooling demands for the building at the climatic conditions considered for the thesis. Since the cooling demand is so small, the heat rejected to the energy wells during summer is limited. The R290 heat pump uses the energy wells as a heat source during the heating season, and depending on the share of heat demand covered by heat recovery, a substantial amount of energy is extracted from the wells. This equals into a large thermal imbalance over the year, and as a result the temperature of the energy wells will drop more and more for each year. This needs to be avoided, and one of the effects is that the evaporation pressure for the heat pump needs to be lowered as the antifreeze temperature drops. The performance of the heat pump will decrease as the pressure lift increases.

One solution to this problem is to install solar thermal collectors on the roof of the building. Solar collectors uses the energy from the sunlight to heat up liquid inside tubes that are circulated in the component. They are commonly used to heat water in southern Europe to supply domestic hot water in residential buildings. When there is sunlight available, some of the heating demand for space heating can be satisfied by the collectors during heating season in late spring and early fall. When there is no heating demand, the collectors can reject the heat to the energy wells to charge them for the winter season. This ensures a thermal balance, and avoiding decreasing temperature from year to year. Another solution that can be implemented is to install a gas cooler/condenser in the supermarket CO_2 booster system connected to boreholes via the low temperature circuit. This can function as a heat rejection system for the refrigeration system for times with no heat recovery. The stable low temperature of the boreholes enables the supermarket to operate in condensing mode. This can be very beneficial during summer when there is no need for heat recovery, but the rejecting to outside air would require trans-critical operation because of higher ambient temperature. For case 2, where the indirect integrating of the hydrogen station is carried out, an additional solution is possible to avoid thermal imbalance. The excess waste heat in the hydrogen cooling after heat recovery can be rejected to the energy wells. When heating season ends, all the waste heat in the stream can be stored in the wells to be used when heating season begins.

In case 3 there is a desuperheater heat exchanger cooling the discharge gas from the hydrogen MT level compressor before entering the high pressure compressor suction line.

As explained in the previous section, in the simulation model the discharge gas is cooled by cooling water at 5°C inlet temperature. The heat given to the water is not included in the heat recovery calculations in the model. Investigating the temperature rise of the cooling water in the desuperheater for a production level of 200 kg/day reveals the outlet water temperature reaches approximate 28°C with a heat flow of 9 kW. This temperature is too low to be utilized for space heating purposes, but this temperature lift is corresponding to the temperature rise in the preheating gas cooler. This might be a application area to utilize this waste heat. Alternatively, the cooling of the discharge gas can be cooled by antifreeze from the low temperature circuit and sent to the energy wells. This may contribute to equal out the thermal imbalance mentioned in the previous paragraph. For high production levels of hydrogen, the available heat from the desuperheater is so high that utilization should be investigated further.

All evaporators in the supermarket refrigeration system are modelled as direct expansion with typically 5 kelvin superheat. Superheating the gas in the evaporators use a substantial share of the heat exchanger area as the heat transfer coefficient for heating gas is much lower than for evaporating refrigerant. Flooded evaporators could be implemented in the system to increase performance, especially for the LT cabinets and for the -40°C level in the hydrogen station. Having a superheat of five degrees when the pressure levels are so low is costing additional compressor power. Having flooded evaporators enables a closer temperature fit between the evaporating refrigerant and the cooled medium through the whole heat exchanger, and the pressure level of the refrigerant can be raised. This is of course a case of trade off between driving forces (temperature difference) and heat exchanger area, and needs to be optimized for each case in order to reach a conclusion.

In the research for supermarkets today there is a trend for investigating solutions where more and more of the required utilities for the supermarket is integrated into the refrigeration system. These utilities include for example, as shown in this thesis, air condition and heating. For case 3 in this thesis, additional evaporation pressure levels are introduced to meet the cooling demands at the hydrogen station. Table 5.2 below show the different pressure levels for case 3.

Pressure level	**Pressure [bar]**
Gas coolers	85
Supermarket MT cabinet	30
Supermarket LT cabinet	15
Parallel/AC	40
Hydrogen station MT	22
Hydrogen station LT	10

Table 5.2: Pressure levels in the refrigeration system for case 3.

The current compressor configuration is arranged so that the hydrogen LT compressor is connected in series with the hydrogen MT compressor, and then delivered into the suction line of the supermarket MT compressor together with the supermarket LT compressor (see figure 3.26 for details). With this arrangement some compressors are working with small pressure ratio. The HP compressors in the supermarket are fixed working between supermarket MT evaporation pressure and gas cooler pressure, but for the other compressors there can be made a choice regarding the working pressures. It would be beneficial to distribute the pressure ratio as evenly as possible to ensure a high overall performance. Especially the hydrogen MT compressor operates with a low pressure lift from 22 bar to 30 bar, corresponding to a pressure ratio of 1.36. A alternative configuration is to send the discharge from this compressor to the suction line of the parallel/AC compressor, with a suction pressure of 40 bar. This increases the pressure ratio to 1.88, and increase the utilization of the parallel compressors. The hydrogen LT compressor can also be set to work directly up to suction pressure of the supermarket HP compressors of 30 bar instead to the suction of the hydrogen MT compressors of 22 bar. This flexibility is a good feature in the case of a compressor failure, as other compressor can be set in operation. Because of all the different pressure levels in the refrigeration system, there is a possibility for having flexible switching between the different units in the compressor racks. There is usually compressors with different capacity in each compressor rack, and by having flexible choice of pressure level the best compressor best matching the load can be set in operation. This ensures the compressors work close to full capacity when in operation, and poor part-load performance can be avoided.

The chosen design for the cooling system in the hydrogen refueling station in case 3 is using cooling baths of antifreeze to keep the temperature of the hydrogen tank at the required level. An alternative solution to this design would be to replace the antifreeze in the cooling bath by oil-free liquid CO_2. As the compressed hydrogen gas enters the tanks in the bath, the liquid CO_2 evaporates. A heat exchanger in the ceiling of the cooling bath condensates the evaporated CO_2 again, keeping the pressure at the required level in the cooling bath. The heat exchanger has colder liquid CO_2 from the supermarket refrigeration system that evaporates on one side, and condensing CO_2 from the cooling bath on the other side. Liquid CO_2 is cheaper to purchase than antifreeze liquid, and are not associated with the same environmental concern if there was a leakage on the system. In case of a failure of the supermarket refrigeration system, and following pressure rise in the cooling bath, a valve for releasing the gas can easily be implemented. The fact that the CO_2 is evaporated in the cooling bath enables the system to operate for some time even if there is a failure of the refrigeration system. CO_2 will evaporate and gas released to ambient until the bath is empty.

Another alternative that avoids the cooling bath entirely is to have coils on the outside of the storage tanks for hydrogen. Liquid CO_2 from the supermarket refrigeration system is sent through the coil that is on the outside of the tank and evaporates as it exchanges heat with the hydrogen tank. If proper design and sizing of such a system is done, it can make the cooling and storage system of the hydrogen station more compact than the cooling bath solution proposed in the current design.

Chapter 6

Conclusion

Design of a building concept including a supermarket, apartments on the upper floors and a hydrogen refueling station with an electrolyser for local production of fuel was carried out. For the building it was chosen a layout of 30 apartments distributed on three floors over the supermarket. An integrated energy system was for the whole building plot was designed with the purpose of utilizing waste heat energy from different sources and reducing the import of primary energy. The energy system was organized in three circuits with different temperature levels. That includes a high temperature water circuit for domestic hot water at 70°C, a medium temperature water circuit for space heating and a low temperature antifreeze circuit connected to boreholes for seasonal heat storage, as heat source for a ground source heat pump and for cooling of ventilation air. Space heating is covered by a low temperature 28/33°C return/supply floor heating system in the apartments. In addition ventilation air is heated by the same circuit.

The building envelope was designed fulfilling the criteria for passive houses given by the Norwegian passive house standard NS3700/NS3701. This involves low demand for space heating due to a super-insulated building envelope and high requirements to prevent heat transfer through windows and doors. Building simulation for a whole year was carried out in the simulation software SIMIEN. The results showed that domestic hot water is the dominating energy load for residential passive houses and low-energy houses. The demand for DHW accounted for 64.6% of the total energy demand for the building through the year, where the apartments were responsible for most of the hot water demand. This result supports the importance of considering energy efficient systems for DHW when designing the energy system for the new generation of buildings. Simulation results show hour-by-hour demand for space heating for the whole year, of which four weeks were selected to give as input to the energy system simulation models in Dymola. The four weeks were picked from each season of the year to investigate the performance of the system for different levels of heating demands and boundary conditions.

Three models was developed in the object-oriented simulation tool Dymola, representing three variants of the energy system for the building. All models includes a CO_2 supermarket refrigeration system, an auxiliary R290 heat pump for space heating and a centralized hot water storage system both for DHW and space heating. The three models, called case 1, 2 and 3, represents different solutions to utilize waste heat that is available on the property. The objective is to investigate how much of the energy demand in the building that can be satisfied by heat recovery. Case 1 involves integrating the supermarket refrigeration system with the centralized energy system for the building. Heat recovery from the refrigeration system was implemented by a tripartite gas cooler solution, recovering the heat from the CO_2 booster system to domestic hot water and to the low temperature space heating system. By assuming a daily profile for refrigeration load in supermarket with base load of 20%, simulations show that heat recovery could cover the full demand of domestic hot water in the building for all weeks. In addition, the full space heating demand could be covered by heat recovery for the summer and spring week, and reaching a share of 75.4% and 31.4% for the fall and winter week. Integrating the supermarket refrigeration system with the rest of the energy system in the building is therefore strongly advised. Based only on the supply of DHW, the potential of energy savings are 91200 kWh/year compared to supplying DHW with an electric boiler. Large energy savings can be reached with moderate modifications of a standard CO_2 refrigeration system.

Case 2 and 3 investigate heat recovery from the hydrogen refueling station in two ways. In case 2, an antifreeze cooling circuit from the hydrogen station brings waste heat at 45°C to the space heating system where the energy is recovered to space heating. In case 3, the hydrogen station is integrated with the CO_2 booster system. The refrigeration load at the station due to the operation of electrolyser, compression and precooling of hydrogen is served by the booster system. The available waste heat/refrigeration load in case 2/3 was linked to the daily production level of hydrogen fuel at the station. The production was assumed constant for the whole week, giving a constant waste heat availability. The first comparison between case 1, 2 and 3 was based on a daily production of 50 kg, 25% of maximum installed capacity. Results showed that both case 2 and 3 displayed similar performance for space heating heat recovery, where both could satisfy the space heating demand for a fall week, and about 60% during a winter week. Due to high uncertainty in the demand for hydrogen fuel in the area, a parameter study of the hydrogen production level was carried out. The production level was varied from 10% to 100% and simulated for the winter week. For both cases a production level of 75% (150 kg/day) and higher could cover all space heating demand. For case 3, the heat due to refrigeration load from the hydrogen station is delivered to both space heating and domestic hot water in the supermarket gas coolers. The available heat that could be delivered to heating of domestic hot water proved to be large for case 3. Since the demand for domestic hot water for the building already could be satisfied by recovering the heat from the supermarket refrigeration load alone, the potential of heat recovery was logged as dry cooler duty. Result showed the dry cooler duty at 25% production capacity was 150% of the heating demand for DHW in the building, and increasing to 9300 kWh or 5.5 times the DHW demand at maximum production level. Several options to utilize this large potential was suggested, including other residential building buildings, a car wash, sports facility, school or swimming hall.

All things considered, the minimum integration that should be carried out for this building concept is recovering the heat resulting from refrigeration load in the supermarket. For small changes to a standard CO_2 booster system, a large energy saving can be achieved. If the hydrogen station is to be coupled with the building energy system, direct integration in case 3 is advised. This solution can best exploit the large energy potential that lies in the refrigeration load at the hydrogen station. Better data, more precise modelling of the hydrogen station and certainty in the demand for hydrogen fuel at the location is needed to confirm the results found in this study. Nevertheless, the potential for production of DHW is shown to be large and should be exploited.

Chapter 7

Further work

In this chapter the suggestions for further work on the topic is presented. From the results and simulation more aspects of the building and its energy system has been discovered that are worth investigating.

The largest uncertainties concerning the models used for simulations in this thesis are linked to the details of operation of the hydrogen refueling station. The calculated waste heat available for heat recovery is based on data from the manufacturer of the electrolyser and information given by the operator of the fueling station. More specific information on the actual operation and cooling demands of the refueling station have to be obtained to establish a better model to give more accurate data on available waste heat. More information and better data on scheduling and load gives an opportunity to develop a model for the station that is more dynamic than the constant-production model used in this thesis, and could therefore improve the accuracy of simulation results.

This thesis has considered and calculated energy savings due to heat recovery from different sources on the building property. The purpose is to demonstrate the potential energy savings that can be achieved by different variants of energy system. A cost analysis should be carried out to evaluate the savings in heating costs compared to additional investment costs the system requires for integration. This analysis can include payback times and evaluate the three variants of the energy system against each other to find the most profitable solution.

Utilizing solar energy to save energy cost is increasing in popularity, also in Norway. Investigation on the possibility for local electricity production by PV panels or by solar thermal collectors could be investigated further to limit the import of primary energy even more. Electricity production is particularly interesting for this building concept, as excess electricity from the PV panels can sent to the hydrogen electrolyser used for production of hydrogen. In this way the hydrogen fueling station can function as a battery for renewable power, and decouple the supply and demand of electric power.

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