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Energy flow analysis of a smart thermal grid at Leangen

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Energy and Environmental Engineering
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Task for master thesis

At Leangen in Trondheim, a new building area will be built in a few years. In order to provide heating and cooling for this area in an energy and cost efficient manner, a local low-temperature grid that utilizes excess heat from an indoor skating rink and alternative surplus heat sources.

The objective of the master thesis will be to gather data on the characteristics of the excess heat supply from an ice skating rink, and the heat demand for the planned building stock should be characterized in order to evaluate to which extent the heating and cooling demand can be covered by the available excess heat combined with local heat pumping devices. In addition, alternative methods for covering the entire heating and cooling demand should be included in the thesis to make a functional, complete and smart system which can be developed in the nearest future.

The building of the Leangen area has different heating and cooling demands over the course of a day, week, month and year. The master thesis will focus on developing a simplified process flow chart over the entire area and a dynamic model of the plant in the object-oriented modelling language Modelica. The work involves mapping the energy available in the surplus sites and the demands in the various buildings in the area. The results should be analyzed and the potential for storing cold and hot energy during off-peak periods should be investigated.

The following tasks are to be considered:

1. Review of relevant literature, e.g. industrial refrigeration systems, HVAC, thermal demands.
2. Develop model(s) representing the energy systems of the Leangen area
3. Perform dynamic simulations with varying thermal loads
4. Analyze the results in terms of system performance, energy consumption and thermal energy storage potential
5. Summary report (incl. discussion and conclusion)
6. Draft version of a scientific paper
7. Proposals for further work

Preface

This master thesis was written in the spring of 2019 in the 10th semester at the department of Energy and process Engineering at the Norwegian University of Science and Technology (NTNU). The thesis is a mandatory assignment for the field of study called Industrial process engineering, and constitutes 30 credits. The master thesis is based on the project work *Energy analysis of a smart thermal grid at Leangen* written the autumn of 2018.

I would like to thank my supervisor Armin Hafner and co-supervisor Hanne Kauko for great help throughout the period regarding theoretic guidance and simulations as well as contributing with great enthusiasm. I would also like to thank Michael Daniel Jokiel from Sintef Energi and Marcel Ulrich Ahrens at NTNU for good guidance with PI-regulators and help with Dymola in general, and Synne Kathinka Bertelsen from the municipality of Trondheim for giving necessary information for the task.

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Abstract

At Leangen in Trondheim a new residential area will be built. The purpose with this master thesis is to evaluate how a local low-temperature thermal grid can supply the area considering space heating, domestic hot water production and space cooling. Waste heat from a nearby ice skating rink and waste heat from greywater will be implemented as an energy source for the low-temperature thermal grid.

Two models for the energy distribution system have been developed to investigate which working fluid is best suited to produce domestic hot water. Model 1 is a system which uses ammonia or propane as well as preheating of tap water. Model 2 uses CO_2 and there is no preheating of water. The municipality of Trondheim have given data from two energy meters at the ice skating rink, and this information is being used further to evaluate the amount of energy demand for the area which can be covered by waste heat. The buildings is simulated in Simien as low-energy consumption buildings. The specific energy demand for space heating and cooling is 28,1kWh/m^2 and 10,6kWh/m^2, while the demand for hot tap water is 29,8kWh/m^2.

The results show that the annual energy demand for space heating and production of domestic hot water for one building can be covered by waste heat from the skate rink by 47,5%, while greywater can cover 33,0%. Energy must be supplied to the heat pumps in the form of electricity, and this amount of energy constitutes 12,9%. It remains 6,6% of the energy demand which must be covered in other ways. Heat supply from the high-temperature district heating network could be one solution. In the future, there may exist other waste heat sources that can be implemented and hence cover the entire energy demand. Model 2 using CO_2 heat pumps was selected for further evaluation due to CO_2 being the medium that achieves the highest COP when producing domestic hot water with a high temperature lift. It was desirable to create a simple and functional system, and model 2 involves fewer connections as it does not involve preheating of tap water. The heat pump including internal heat exchanger and system overheating of about 10K achieves a COP of 4,5 for the CO_2 heat pump connected to the space heating circuit and a COP of 4,3 connected to the space cooling circuit with the temperatures given in our system. Water tanks have been implemented for thermal storage of hot tap water to reduce the power peaks of the district heating network by having a continuous water production. High temperatures in a storage tank indicate lower volumes which can be an advantage in dense residential areas. In addition, the legionella bacteria can not live in water temperatures over 65°C. Therefore it has been decided to make storage tanks connected to each building with a temperature level of 75°C stored in a size of between 1460-2098L.

The system solution is based on a number of assumptions, and it is important to be critical of the results. The results will however give a picture of how external and internal waste heat can cover the energy demand of a large residential area and how a low-temperature thermal grid with smart solutions can lower the energy consumption in our society.

Sammendrag

På Leangen i Trondheim skal det bli bygget et nytt boligområde. Hensikten med denne masteroppgaven er å evaluere hvordan et lokalt lav-temperatur varmenett kan forsyne boligområdet med energi til romoppvarming, produksjon av tappevann og romkjøling. Spillvarme fra en skøytebane i nærheten, samt spillvarme fra gråvann er implementert som en energikilde til lav-temperatur varmenettet.

To modeller er utviklet for å undersøke hvilke arbeidsmedium som er best egnet til varmepumpen som skal produsere tappevann. Modell 1 er et system som tar i bruk ammoniakk eller propan som arbeidsmedium og forvarming av tappevann er introdusert. Modell 2 bruker CO_2, og tappevannet blir ikke forvarmet. Trondheim Kommune har gitt data fra to energimålere på skøytebanen, og informasjonen er videre brukt til å evaluere hvor mye av energibehovet til boligblokkene spillvarmen kan dekke. Boligblokkene er laget og simulert i Simien som lavenergi-bygg. Spesifikt energibehov for romoppvarming, tappevann og romkjøling er 28,1kWh/m^2, 29,8kWh/m^2 og 10,6kWh/m^2

Resultatene viser at det årlige energibehovet for romoppvarming og tappevannsbehov til én boligblokk kan dekkes av spillvarme fra skøytebanen med 47,5%, mens gråvann kan dekke 33,0%. Det må tilføres energi til varmepumpene i form av elektrisitet, og denne mengden energi utgjør 12,9%. Da har vi resterende 6,6% av energibehovet som må dekkes på andre måter som for eksempel kan være tilførsel av energi fra høy-temperatur fjernvarmenettet. I fremtiden kan det muligens finnes andre spillvarmekilder som kan implementeres og derav dekke hele behovet. Modell 2 som bruker CO_2 som arbeidsmedium i varmepumpen ble valgt for videre vurdering. CO_2 er det mediumet som oppnår høyest COP ved tappevannsproduksjon hvor høyt temperaturløft er nødvendig. Det var ønskelig å lage et enkelt og funksjonelt system, og modell 2 innebærer færre koblinger da det ikke innebærer forvarming av tappevann. Varmepumpen ble undersøkt når den var forbundet til varmekretsen og kjølekretsen. Ved rundt 10K overopphetning med intern varmeveksler er det mulig å oppnå en COP på 4,5 for CO_2-varmepumpen forbundet med varmekretsen og en COP på 4,3 forbundet med kjølekretsen med temperaturene gitt i vårt system. Det er blitt implementert vanntanker for lagring av varmt tappevann for å redusere effekttopper i fjernvarmenettet ved å ha en kontinuerlig vannproduksjon. Høye temperaturer i en lagringstank tilsier mindre volum noe som kan være en fordel i tette boligstrøk samtidig som legionella bakterien ikke klarer å leve i temperaturer over 65°C. Det er derfor blitt valgt å implementere en lagringstanks forbundet til en bygning som inneholder vann på 75°C som lagres i en tank på størrelse mellom 1460-2098L.

Systemløsningen er basert på en rekke antakelser, og det er derfor viktig å være kritisk til resultatene. Resultatene vil kunne gi et bilde på hvordan ekstern og intern spillvarme kan dekke energibehovet til et stort boligområde og hvordan et lav-temperatur energinett med smarte løsnigner kan senke energibruket i vårt samfunn.

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1 Introduction

1.1 Background and motivation

In Europe the energy use of buildings constitutes 40% of the total energy consumption, and 75% of the buildings which exists today are inefficient[12]. In Norway the energy demand represents 40% of the total energy consumption as well, and the energy demand of residential and non-residential sector represents 22% and 18% of this percentage[36]. It is desirable to decrease the consumption of energy, as well as increase the amount of energy efficient buildings. To achieve this goal a *smart* and integrated energy supply system is needed. A combination of a low-temperature district heating network, together with waste heat sources and heat pumps can contribute to an increase of the energy efficiency of a system.

This master thesis considers information about a building area at Leangen in Trondheim, which will include a local, low-temperature heating system and low-energy consumption buildings. The building area does not exist at the moment, and a reference building is made in Simien to evaluate the total energy demand of the area. The buildings extract the same amount of energy considering space heating and cooling, as well as domestic hot water. When thermal treatment of the legionella bacteria is considered, a low-temperature heating system with a supply temperature of 40°C can not heat up hot tap water to the desired temperature level which prevents the bacteria from living. Therefore, a heat pump is implemented to reach the desired temperature level for domestic hot water production. It is of interest to make an energy system which can contribute to utilization of existing but hidden energy sources in our society. Typical buildings with waste heat at low temperatures is data centers, grocery stores and ice skating rinks.

1.2 Goal and structure

At Leangen in Trondheim a new building area is being made by Koteng Eiendom AS. This master thesis considers a compact area which consists of mainly residential buildings which is supposed to be both sustainable and have a low energy consumption. Therefore a low-energy consumption building block is considered. The goal with this master thesis is to consider the potential of integrating waste heat from an ice skate rink nearby and thereby construct a low-temperature district heating network which is fed by this source and maybe other waste heat sources in the future. The municipality of Trondheim have provided data from two energy meters at the ice skating rink which is used to investigate the energy potential of the waste heat. Further on, two different models of energy distribution system within the building block is being considered which utilizes the waste heat in two different ways. The most profitable and relevant system is being investigated further in Dymola. Following intermediate goals are required to answer the questions in the master thesis assignment:

- Literature research of relevant themes
- Define a reference building with a low-energy consumption
- Investigate the energy potential in waste heat from ice skating rink and other potential waste heat sources
- Develop models in Visio of the low-temperature heating system and the building interface. The models should consider space heating and cooling units, as well as heat pump and storage tanks for domestic hot water

- Use Coolpack to investigate how different working fluids influence the heat pump cycle when producing domestic hot water
- Use Dymola and investigate one model with the given temperatures in our system
- Make a cost analysis and compare prices of our new system compared to electricity and conventional district heating systems

The master thesis has 7 chapters. The first one is an introduction to the master thesis and why it is a relevant assignment. Further on, chapter 2 deals with the literature review, and contains relevant themes for this master thesis. This is the chapter which gives a foundation for the system solution made. Chapter 3 presents the different methods used to make a system solution, and contains information such as how to make a reference building in Simien, how Coolpack is being used to compare different working fluids for the heat pumps and how the Dymola model should be made. In chapter 4 and 5 the results and discussion part are represented. The conclusion is given in chapter 6, while chapter 7 handles ideas for further work.

1.3 Limitation of project work

The simulations made in Simien is made for a fictive residential building with 20 apartments and 5 floors, and one apartment have a size of 70m^2. There may be some changes in real life. Only one user profile was simulated, while in reality each building will have users with different energy demands. However, the assumptions made is based on the data given from Asplan Viak AS and Lund Hagem Arkitekter AS and the average values for the area can give an indication of the energy consumption for an average building and thus the entire building area.

Another limitation of the master thesis is whether or not the waste heat is available for the project. The temperature of the waste heat is also unknown, and specifications of the ammonia refrigeration system should be investigated further. The temperature of the waste heat is based on discussions at meetings and the literature review. However, it is of interest for district heating suppliers and real estate companies to develop a local low-temperature grid to supply the energy demand of the building area and investigate the potential in waste heat sources in general for low-energy consumption buildings.

The models being investigated do not handle the losses in pipes and valves, and the temperature drops in the systems should be investigated as well. The amount of available greywater is also uncertain and depends on the user behavior of people living in the buildings. This may be considerations which have an impact on the conclusion.

2 Literature review

2.1 Heat distribution within the building area

In order to achieve the goals of international regulations, as well as supplying the required amount of energy to buildings, the implementation of waste heat and renewable energy sources is required in the heat distribution system[44]. Several studies[29] concludes that district heating systems have an important role when it comes to implementing future sustainable energy systems. This chapter contains information about district heating, low-temperature district heating, as well as existing smart thermal grid projects in Norway and Sweden. Chapter 2.1.1, parts of 2.1.2, 2.1.3 and 2.1.4 is based on the project thesis[1].

2.1.1 Review of district heating

Conventional district heating (DH) systems are based on central and large industrial waste or combustion plants[59]. These heat sources have a large capacity and a high temperature which can be utilized for space heating purposes in the community instead of vanishing into the surroundings. DH systems in Norway are based on the heavy industry sector which is mainly placed on the coast due to the industry having a large energy demand. Hydro power is available as an energy resource at the coast of Norway, and ships do have an easier access to the industry. However, it exists some present buildings in our communities in other areas than the coast of Norway which represents heat sources and can be utilized for space heating. Some of them is listed below[59]. The building types represented have large chillers and refrigeration facilities.

- Data centers
- Office buildings
- Food retail stores
- Skate rinks

Four generations of district heating
From the 1880s until 1930, steam was commonly used as a heat carrier in USA and Europe[29]. This is known to be the first generation of DH systems. Such a system provides big heat losses in the network due to the high temperatures of steam, causes danger due to the high pressure, and it often appears corrosion in the return pipes which decreases the efficiency of the system. DH systems based on steam is not commonly used nowadays. In the period of 1930 to 1970, the second generation of DH systems were introduced and this generation provided temperatures of about 100°C or more with a heat carrier consisting of pressurized water. This is a system which still can be found in old parts of the DH systems based on water. The third generation of DH systems was introduced in the 1970s, and is a system which replaces the outdated versions of DH systems in parts of Europe as well as being the new DH solution for DH systems in China, Canada, USA, and Korea. Temperatures of such a system is typical below 100°C. The development throughout these three generations have been to reduce the temperature in the system, prefabrication and to avoid corrosion using lean material components[29].

The fourth generation considers the infrastructure of the DH system as well as new technologies entering the market. This generation is a low-temperature district heating (LTDH) system and should be able to integrate several heat sources, and be a part of an energy system integrated with the electricity and industry sector. Examples of additional heat sources added to the system together with the conventional DH system are industrial surplus heat sources and other waste heat sources available in the area together with renewable energy sources such as bio-gas, wind, geothermal and solar energy[31]. The system should be able to supply LTDH for both space heating and domestic hot water to both existing buildings and new buildings, as well as optimize the networks to ensure low grid losses. The fourth generation of DH is dependant of the development of future buildings to achieve a better energy efficiency of the total system[29]. In addition, this generation will take advantage of modern measuring equipment, as well as different thermal storage technologies and advanced information technology which will make the fourth generation a more intelligent system or *smart* DH system. A *smart* system is achieved by implementing three essential elements. The first one is the physical network (PN) which includes control devices, pipes, local meters and equipment for heating. Secondly, we have to include the internet of things (IoT) and this element includes sensors, the collecting of data from the system, devices for the transmission and more. The third element which needs to be included to achieve a smart DH system is the intelligent decision system (IDS). Based on the collected data, as well as heat demand, the system should be able to make optimal decisions[31].

Many studies[29] concludes that together with the synergy of different sectors, it is important to reduce the losses in the grid to gain an increase in efficiency of low-temperature production units. In order to minimize the heat losses and the distribution costs, it is of interest to have a concentrated area which requires a heat demand[29]. According to a study[31] which includes different DH systems, the percentage of heat loss from the system vary up to 32% of the delivered heat, in which 10 out of 13 systems was LTDH systems. These losses is one of the biggest concerns for future DH systems. The losses also appear in quite new systems from 2010 made to serve a LTDH system, and 10 out of 13 systems have heat losses above 15%. In Norway the losses are in the range of 8-15%[44]. Potential solutions which is widely used to solve these massive heat loss issues in the network are many[31]. It is possible to install pipes inside buildings and make house-to-house connections, but this is not welcomed by DH companies. It is also of interest to shut down pipes in the summer period, which is a concept being investigated at the moment. Other methods to reduce losses is to use booster pumps, or apply branched network with bypass. The latter solution can result in higher return temperatures. It is of interest to reduce the temperature difference between supply and return temperature to decrease the distribution losses due to less use of pump energy. Losses is to be found in the pipes of the system as well, and therefore a more suitable insulation can be a solution to reduce losses in this area. Implementing buffer tanks in the system is a solution which have its limitation due to the legionella bacteria. Water leakages appear, and the overall system will benefit from installing leakage detection to insure no hot water loss and fault detection.

2.1.2 Low-temperature DH system

The future buildings have a low energy consumption considering space heating which results in no need for high temperatures in a future DH system[44]. The heat supply determines the supply temperature, and the cooling processes at the customers determines the return temperature. In a DH system with water as carrier, there will be some heat supply units in the system in which water is heated, and at the customer substations the water cools down. The return temperatures of the network is dependant on the local conditions of where the system is operating. By improving the control of the system and mapping errors, as well as the improving the design, the temperature level can be reduced even further than the third generation temperature levels[31].

When having eliminated the errors of the system, together with improved control and new technology for the stations, it is in theory possible to reduce the supply temperature level to 69°C and have a return level of 34°C. A system in Denmark showed that the goal is reachable in real life as well, and in 2009 the Danish system reach an annual average supply temperature of 74°C and a return temperature level of 36°C[31]. Below, the advantages of a low-temperature distribution system is listed[59].

- Reduction of heat losses in distribution network
- Utilize waste heat sources of low temperatures
- Higher COP for heat pumps

2.1.3 Supply temperature requirements for buildings

Requirements for space heating
When heating up new buildings, 50°C as supply temperature should be enough to satisfy the demand together with low-temperature radiators and floor heating[31]. According to Sintef[83] a temperature of between 35-40°C should be enough to cover the space heating demand. This temperature range contributes to a floor surface temperature of about 23-28°C and a heat output of 30-40W/m^2. For cooling of buildings about 14°C is enough[83]. In Norway the share of older building stock would be about 50% in 2030, and old buildings have a system which requires supply temperatures of 70°C or more[31]. This is something which have to be considered due to the fact that a lower supply temperature can cause discomfort for people living in old houses. It exists studies which concludes that by having a supply temperature of 50°C, it is possible to supply old houses from the 70s and 80s without renovation when increasing the supply temperature to 60°C in some extra cold periods. However, by replacing windows of older buildings the supply temperature can be lowered below 60°C throughout the year, and the temperature can be further decreased by replacing old radiators with low-temperature radiators resulting in a supply temperature of 50°C through the year[31].

2.1.4 Heat sources

It exists a various diversity of heat sources which can be included in a low-temperature district heating system. The energy sources being discussed have the ability of being implemented in such a system.

Renewable energy sources
Solar energy in district heating is not commonly used, but it exists some developments in countries such as Austria, Germany, Denmark and Sweden. This is an energy source which is unpredictable and the technology crave big space. Solar collectors can be used to preheat the water, however this concept have a performing fraction of 5%. If the solar fraction on an annual basis make up 15%, the solar collectors have the potential of covering 100% of the summer load. Solar collectors are also an instrument to harvest heat to large seasonal thermal energy storages which is further used to heat up space in the winter period. This technology can be based on the ground as a big scale centralized operating area or on the rooftops as decentralized operating units connected to the DH system. Solar collectors implemented into the DH system as decentralized solar collectors will bring flexibility into the system by making the buildings either exporters or importers.

Geothermal energy only compose a small fraction of the energy supply in the world today. Storages with high temperature can be used for power generation, as well as being used directly for district heating. Even though this energy source has its benefits by being available all over the world and having minimal environmental impact it also has its limitations. This technology has high investment costs, can cause human made earthquakes, and the amount of which can be extracted decrease over time. The geothermal storage have different depths. Some are deep and have a depth of 500-5000 meter, while shallow geothermal systems have a depth of 300 meter. By using a heat pump, heat in for example a shallow geothermal system with aquifer thermal energy storage can be utilized[31].

Waste heat utilization in general
When implementing waste heat sources the heat source can be utilized directly, or by lifting the temperature using a heat pump. The industrial waste heat source can be unstable and depend on the operating hours and the season. Thermal energy storage is therefore one solution to this uncertainty. In 2010 Denmark used the industrial waste heat to supply their DH system, which covered 2-3% of the energy supply to the system. In Sweden waste heat covered up to 6% together with 27% waste heat from two oil refineries[31]. In the industry processes, much of the energy in the form of steam and combustible gas is of high grade which can be used within the industry itself for power generation. Another reason for why the percentage of industrial waste heat used in DH systems is quite low is the technical issues with integrating this heat source. Normally, the customers for DH can be far away from the industry which results in both big investments considering the network, as well as big transmission heat losses. The supply of heat to a small-scale town should be within 5-10 km, while supply to a medium or large city should be within 20-30 km[31].

Waste heat utilization of ice skate rinks
As mentioned in chapter 2.1.1, it exists some other building types which can contribute with low-temperature waste heat to the local LTDH system instead of only relying on waste heat from the industry. Westhills Recreation Centre in Langford (Canada) has ice skating rinks. Its refrigeration systems produces waste heat leaving the condensers. This waste heat is being utilized for space heating and production of domestic hot water for local buildings. The refrigeration system runs 11 months in a year, and the area consists of indoor and outdoor ice skating rinks, and a bridge connecting them both. In the same facility it exists a bowling hall, activity rooms, and a restaurant which all have different heating demands. The concept of this area is to integrate waste heat from the ice skating rinks together with the building with other facilities and further send the leftover heat to a residential area nearby. The project is reward winning and got first price of ASHRAE Technology Awards in 2015 within the category *The New Public Assembly building*[14]. In this project it appeared that 60% of all the energy added to the refrigeration system turned into waste heat. The building with the ice skating rinks and other facilities only needed 40% of this waste heat, and the rest was transported to a residential area for space heating in a building area. The building area used the leftover of the waste heat as an energy source for its heat pumps placed 365 meters away from the building. This project which permits the integration between ice skating rinks and society is the first project of its kind in North America. High efficient piston compressors was used together with ammonia as a working fluid in the heat pumps which is heating the residential area[14].

The building of the skate rink area consists of concrete floors, whereas temperatures between 22°C and 24°C are maintained to provide heat to the building. However, in the winter period, the glycol temperature need a temperature rise to provide the same heating comfort for the building. An energy recovery heat pump is applied to increase the temperature further up from 35°C to 40°C to keep the same level of comfort as in the other seasons of the year. The heat pumps in the building have an uninterrupted heat source from thermal storage and a long compressor run time. The heat from the desuperheating system has a range of between 38-49°C, and a heat pump is implemented to lift the temperature level up to 60°C to produce domestic hot water. The heat pump uses the energy recovery loop as a heat source, and at winter time the production of hot water gives a COP of 4.28[14].

2.1.5 Existing smart thermal grids

It is of interest to consider existing projects regarding smart, thermal grids with low temperatures and thermal energy storage. One project in Sweden and one in Norway will be mentioned. The projects have elements of the future DH system.

Ectogrid in Sweden
The first fourth generation DH system in the world was built in Sweden and is known as the Medicon Village. The thermal grid was made by E.ON and named *Ectogrid*[73]. The concept considers that each building in the area can either supply the low-temperature district heating system with heat or cold or take advantage of it to cover its own energy demand. The system is able to collect information about the users and thereby optimize use of energy. Future energy use together with weather conditions can be predicted and thereby stored in the ground[74]. In figure 1 an energy pyramid is illustrated.

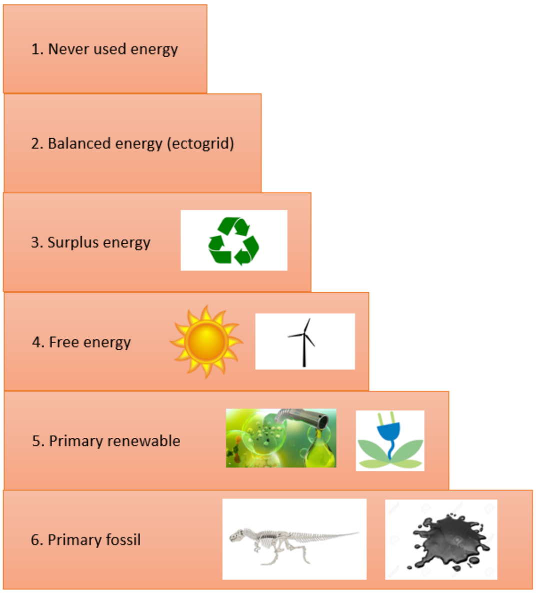

Figure 1: Energy pyramid

The fourth generation DH system at Medicon Village represents a new step in the pyramid of different energy sources[73]. Balancing the energy in such a way the ectogrid technology does makes this an energy efficient system. A balanced energy system is based on local industries and reuse of energy flows with its roots from waste water. This is the second best way of using energy. The best strategy is to not use energy at all, and the use of fossil energy is presented as the lowest step due to its green house gases[73]. The system at Medicon Village is a setup for the Mats Paulsson's foundation for Research, innovation and societal development. In the area, there are more than 1600 people working with science and improvement of the life and health of people.

The foundation demand that all of surplus energy must be reused, and the results from the project is being directly put into research and innovation[74]. The area is shown in figure 2.

Figure 2: Ectogrid at Medicon Village[74]

Medicon Village contains 15 commercial and residential buildings. Each building has its own specific demand for space heating and cooling and constitutes a yearly demand of 10GWh and 4GWh. When the concept of ectogrid is applied, it is possible to balance as much as 11GWh. The system will thereby require an energy input of 3GWh with such a smart, thermal grid[74]. The balance of the energy flow is shown in figure 3.

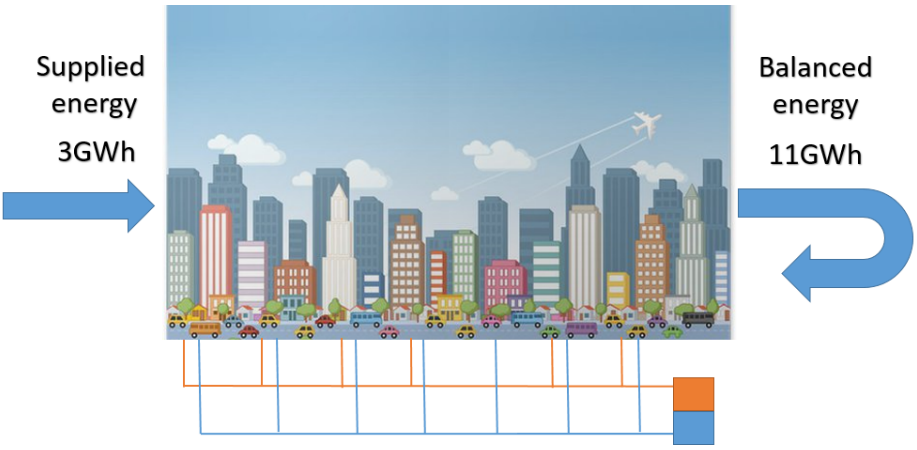

Figure 3: Energy balance at Medicon Village

Balancing thermal energy flows from different buildings and reuse of energy makes it possible to reduce the energy input of a system by 78,6%.

Smart micro energy system in Oslo

At Furuset in Oslo it will be made a new, thermal energy system which is called a smart micro energy system. The system contains seasonal thermal energy storage in which waste from incineration is implemented. In the summer, the energy recovery system at Klemetsrud produces more energy than needed, and the goal is to have a reservoir consisting of several deep wells with a depth of 200 meters. This stored energy will be used in the winter season when needed. Trond Moengen from Energidata Consulting and also the project leader of Klimaetaten have commented[95] that such a project has never been done in Norway before. The project has received 36,8 million NOK from Enova and is a project which gathers a lot of companies such as Fortum Oslo Varme, Hafslund Nett, JM Bygg Norge, eSmart Systems, Pixii and the municipality of Oslo.

The system will communicate with the electricity grid as well as the DH network and utilize local energy production such as energy from batteries and solar cells, as well as excess heat from DH production which will be stored for later use. The system should be able to cooperate with the whole energy system. When the autumn and winter is near, and the temperatures decreases, the thermal boreholes will be connected to the low-temperature thermal grid for local energy distribution to cover the energy demand of buildings. The dimension of the reservoir and amount of boreholes and the size of them is not yet decided. The capacity however will be about 15GWh of heat which is adapted to the heating demand of Furuset. The energy loss is not yet known, and depends on the conditions of the ground of the area. It all depends on the geological conditions. If it exists ground water which flows through the area, a lot energy will be wasted. According to an assessment of the area it should be possible and attractive to implement the concept of thermal energy storage in the area. Moengen comments[95] that there will be heat losses, but the heat stored is actually heat which would be sent to the ambient air if not stored. He compared theoretical studies and experiences from Sweden in which 70-80% of the energy stored in boreholes could be used.

The energy recovery system at Klemetsrud have a nominal effect of 151MW, and the production is based on waste heat from waste incineration. The system produces both electricity and district heating, in which 115MW goes to the district heating system. Eirik Folkvord Tandberg, administrative director for Fortum Oslo Varme, comments[95] that the idea behind the concept of thermal energy storage is to utilize the waste heat of Oslo in the best possible way. It exists 150GWh with waste heat at the summer time which should be cooled down. It is also being investigated through other projects for technology development to see if the heat can be used for cooling which would introduce the system to the concept of flexibility. The micro energy system project at Furuset is one of eight demonstration projects for the future energy systems. The eight projects received all together 210,3 million NOK from Enova because they are seen as highly useful for the future energy distribution system in Norway. The goal is to demonstrate digital -and technological solutions as well as business models which utilizes the flexibility of the energy system.

2.2 Components of a DH grid

This chapter includes information about important components in the DH grid such as heat exchangers, circulation pumps, pipes, valves and PI-regulators. Chapter 2.2.1 and 2.2.4 is partly based on the project thesis[1].

2.2.1 Heat exchangers

Heat exchangers is used for several applications such as space heating and cooling and production of domestic hot water. The amount of heat stored in a material is determined as shown in equation 1[8].

$$Q = m * c_p * \Delta T \tag{1}$$

Equation 1 indicates that the capacity Q (kW) is equal to the mass flow (kg/s) multiplied by specific heat capacity of water (J/kgK) and the temperature difference T (K).

Heat transfer is the process when energy flows from one region of higher temperature to another region of lower temperature. The process will not stop until the temperature level is equal to one another. It exists different types of heat exchangers with different flow configurations such as counterflow, crossflow and parallel flow. Heat exchangers have different names depending on their purpose; condensers, gas coolers, evaporators, superheaters etc. Heat transfer appears in three different forms[60]:

- *Conduction*: Heat transfer in solids, motionless gases or liquids. Kinetic energy in molecules is transferred from a molecule to another nearby
- *Convection*: Heat transfer in gases or liquids. Bulk fluid motion
- *Radiation*: Electromagnetic waves. No presence of an intervening medium

Heat exchanger devices will have an impact on both the overall efficiency of the system as well as the size of the system. To achieve the best possible design of a heat exchanger a balance between the pressure drop and the effectiveness should be obtained[76].

Calculations of heat transfer of counterflow heat exchanger
The calculation of heat transfer in a counter flow heat exchanger involves equation 2 and 3[8]. Important units is the outer surface area of heat exchangers (A_o), fluid temperatures (T), the overall heat transfer coefficient (U_o) and the log mean temperature difference (ΔT_{lm}). These units multiplied together equals the capacity (Q).

$$Q = U_o A_o \Delta T_{lm} \tag{2}$$

The heat transfer process is shown in figure 4 which is inspired by a textbook[60]. Equation 3 is the euation for the log mean temperature and have parameters which is equal to ΔT_1 = T_{h1}-T_{C2} and ΔT_2 = T_{h2}-T_{C1}[60].

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{ln(\Delta T_1 / \Delta T_2)} \tag{3}$$

Figure 4: Counter-flow heat exchanger

Different types of heat exchangers
Plate heat exchangers uses several metal plates to transfer heat between different fluids[23]. The advantage with this heat exchanger is the use of big area in which the fluid can transfer heat across. This heat exchanger is available in small sizes and it has a high transfer efficiency. It is suitable for medium and low-pressure fluids. If the heat exchanger is going to be used for high-pressure fluids a welded or brazed plate heat exchanger should be used. The plate heat exchanger can have a temperature approach of 1°C[19]. The geometry of the plate is important factor due to pressure drops and heat transfer. High heat transfer and low pressure drop is good[23].

Tube in tube heat exchanger is nice to use when the system is using high temperatures and high pressures as well as a low flow[20]. The heat exchanger have an inner and an outer tube which is coiled together and makes the design quite compact design. The compact design contributes to a smaller size and an increased efficiency. Its easy to clean and it is a low maintenance of the heat exchanger. Typical application is water cooling, sewage sludge heating, and milk, cream and juice heating. It is also suitable to high temperature and pressure applications.

Shell and tube heat exchanger is typically used in the oil industry due to the fact that its suits high pressures but can also be used for heating of swimming pools. The unit consists on a large pressure vessel (the shell) and tubes inside of it[21]. One fluid flows over the tubes while the other fluid flows inside the tube. The shell and tube heat exchanger is a good choice of heat exchanger due to its easy service maintenance with the models of floating tube bundle. The temperature approach of a shell and tube heat exchanger requires 5°C or higher[22].

2.2.2 Circulation pumps

When implementing space heating and cooling technology based on a heat transfer fluid in a building, it is necessary to add circulation pumps in the system to transport the fluid. With this equipment it is possible to overcome the hydraulic resistance which exists in the system. There is a need of an over-pressure in the system to create the motion, and the circulation pump is often driven by an electrical motor[67]. When energy is added to the fluid, the discharge pressure at the exit of the pump will be higher than the inlet pressure and we get a movement[77]. The most common heat transfer fluid for space heating and cooling of a building is water with a temperature range up to 110°C. The pump characteristics and its performance will vary depending on the heating system and its characteristics[67].

2.2.3 Pipes and valves

The material of the pipes must be chosen according to the fluid properties when designing the system. The fluid can have the ability of being corrosive or erode the pipe. It should be investigated if the system can implement plastic pipes instead of metal pipes due to its ability of being corrosion resistant and having a light weight. Polyvinyl chloride (PVC) is not recommended for hot water piping, but can be used for cold water systems. Chlorinated polyvinyl chloride (CPVC) however can be used for temperatures up to 60°C while reinforced thermosetting resin plastic (RTRP) is recommended for temperatures up to about 93°C. Cross-linked polyethylene (PEX) is recommended for domestic hot water piping, which is not considered to be a good alternative for use as residential water pipes instead of copper tubing and PVC and CPVC. A system with high pressures needs higher pipe strength, and the velocity of the fluid must also be considered. Depending on the application the pipe velocity have different ranges and requirements. It is necessary to have a velocity of 3m/s for general building service to reduce pipe material erosion as well as noise. The determination of the diameter of the pipe is usually set by the volume flow rates of the pipes[77].

A valve is a unit which can regulate the mass flow of the fluid by opening or closing the passageway. It is also possible to partially open or close the passageway. Many valves an be controlled manually, and others are controlled actuators. Actuators is used to obtain an automatic control of the system, and is driven by changes in measures of pressure, flow or temperatures. Most valves are control valves. Often, a control valve is operated in combination with independent sensing devices which register the flow, pressure or temperature level in the system[68].

2.2.4 Regulation and control equipment

The customer substation of a district heating system consists of different units such as heat exchangers, valves, pumps and pipes. There must be applied a control system for regulation of how heat and cold supplies a building, as well as how the domestic hot water system and ventilation system should be regulated. A control system will make sure that the set point values will be maintained. In this chapter, the regulation of space heating will be the main focus.

Principles of regulation
The control and monitoring of larger systems such as a district heating system have a few main goals. Firstly, the return temperature difference across each costumer substation should be maximized to obtain heat delivery to the customer. Secondly, a low temperature in the supply line of the district heating system will give the benefit of reduced temperature gradient to the soil or air in contact with the system which decreases the losses in the distribution system. Finally, in times of large heat demand, shift heat usage could be applied to reduce the production of energy in the first place. This could be either pre-heating or pre-cooling of the building. All of those efforts will contribute to a better overall efficiency of the DH system considering economy and environment[69].

Control equipment
It is desirable to have a minimum of additional computational power and control parameters for the system when trying to achieve a good indoor climate. A substation (or nearby the substation) have several units which controls and measures the system:

- Space heating control units
  - Control valve
  - Temperature sensor
- Heat meter
  - Temperature sensor
  - Flow sensor
  - Remote communication unit
- Circulation pump
  - Regulates the space heating system

It is possible to obtain the desirable indoor temperature by measuring the outdoor temperature, and thereby utilizing valves to achieve the desirable temperature in the heating system. The valves regulates the flow on the primary side of the heat exchanger which decides the heat delivered on the secondary side of the heat exchanger. Such a regulation process is shown in figure 5

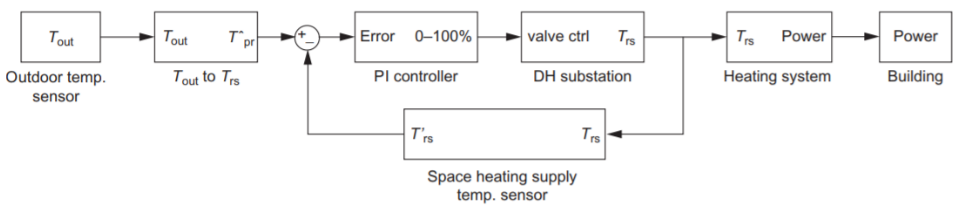

Figure 5: Regulation system of space heating[69]

In figure 5, the difference between outdoor temperature (T_{out}) and system supply temperature for space heating for radiators (T_{rs}) is predefined by a linear relation range of relevant temperatures in the geographical area of the district heating system. T_{ts} and T_{pr} represents the supply and return temperature for the primary side of the system. T_{rs} can be controlled by being measured and sending feedback to the controller as shown in figure 5. The control system adjusts the supply temperature to suit the thermal power supply according to the demand. In this way, we do avoid both discomfort by heating too much or too little, as well as we prevent energy waste[69]. In figure 5 a PI controller is shown. A PI-controller is a controller used when the process returns to the same output value when the input values and disturbances is the same[71].

Tuning regulators with experimental method
Without knowing anything about a process, *The Good Gain method*[98] can help tuning the regulator in an optimal way according to your system. The method is done as an experimental method. This method is known to achieve an even more stable control loop than the Ziegler Nichols methods which gives the amplitude ratio between subsequent oscillations after a step change of the set point equal to 1/4. The experimental method is done with the following steps:

1. Adjust control signal u_0 close up to the operation point
2. Start with K_p=0, T_i=∞, T_d=0. Increase K_p until control loop is stable. Increase K_p until overshoot and barely undershoot. This is good stability.
3. The time between under -and overshoot is represented as T_{ou}. The integral time is set to 1,5*T_{ou}.
4. A PI-controller have to care about the I-term as well in which the controller will be less stable than the P-controller. Therefore, to compensate, we introduce the K_p term which is set to K_p=0,8K_{pGG}
5. A PID controller includes a D-term and the T_d is set to T_i/4
6. Apply a step change of the set point. When the stability is poor, it is possible to get it more stable by reducing the Kp value and increasing the integral time

Night-set control
When lowering the indoor temperature at night time, this is known to be *the night-set back control*. Such a concept contributes to a decrease of the total heat demand. The reduction of the heat demand will be noticeable, while the reduction of temperature at night will not be very noticeable. It is known that night-set control will be profitable only for building types with a specific and high demand. The insulation of the building will also be a factor of importance, as buildings with non-airtight building envelopes and bad insulation will be the ones with a high saving potential. The night-set control results in a low head load during the night and a high heat load in the morning which means a high peak load. However, this peak load will vanish quite fast[70].

2.3 Building specifications of low-energy consumption buildings

Before choosing a heating system for a building it is important to evaluate the energy demand of the building. Therefore it is important to investigate building specifications in different areas such as energy -and power demand of a building, indoor climate specifications, and ventilation.

2.3.1 Energy -and power demand of a building

The energy demand constitutes the necessary heat output to keep a certain set point temperature as well as a certain airflow of ventilation to maintain an adequate replacement of air. Regulations of TEK17, indicates certain requirements for the building[87], and the requirements for total net energy demand of different building types is listed below in table 1[85].

Table 1: TEK17 energy requirements

Building category	Total net energy demand [kWh/m^2]
Small building and house over 150m^2	100 + 1600/m^2
Apartment building	95
Children's garden	135
Office building	115
School building	110
Nursing home	195 (230)

The parenthesis in table 1 indicates requirements for areas in which heat recovery of ventilation air entails a risk of spreading contamination or infection. The energy measures remains if the building's heat loss do not increase[87]. According to sources from Sintef Byggforsk[84] a passive house building have a power demand of 10W/m^2 for space heating.

2.3.2 Building specifications

A low-energy consumption building needs strict specifications of the construction. The energy measures is shown in table 2 which is the requirements of TEK17[87]. The data is given for both small buildings and apartment buildings.

Table 2: TEK17 building requirements for small buildings and apartment buildings

Energy measures	Small building	Apartment building
U-value outer wall [W/(m^2K)]	$\leq$ 0,18	$\leq$ 0,18
U-value roof [W/(m^{2}K)]	$\leq$ 0,13	$\leq$ 0,13
U-value floor [W/(m^{2}K)]	$\leq$ 0,10	$\leq$ 0,10
U-value windows and doors [W/(m^{2}K)]	$\leq$ 0,80	$\leq$ 0,80
Amount of window -and door area of heated BRA	$\leq$ 25%	$\leq$ 25%
Efficiency for heat recovery in ventilation system	$\geq$ 80%	$\geq$ 80 %
Specific fan power in ventilation system (SFP) [kW/(m^3/s)]	$\leq$ 1,5	$\leq$ 1,5
Air leakage per hour at 50 Pa pressure difference	$\leq$ 0,6	$\leq$ 0,6
Normalized thermal bridge value [W/(m^{2}K)]	$\leq$ 0,05	$\leq$ 0,07

The requirements of low-energy consumption buildings and passive house buildings is shown below in table 3. It exists some differences between these two building types regarding the construction[86].

2.3.3 Specific energy demands

It is of interest to know specific values when investigating the energy demand of a building. By looking at standard NS 3700:2013[86] for low-energy consumption buildings and Sintef

Table 3: Requirements for passive houses and low-energy consumption buildings

Minimum requirement	Passive house	Low-energy consumption building
U-value windows/doors	$\leq$ 0,80W/m^{2}K	$\leq$ 1,2W/m^{2}K
Normalized thermal bridge value	$\leq$ 0,03W/m^{2}K	$\leq$ 0,05W/m^{2}K
Efficiency of heat recovery	$\geq$ 80%	$\geq$ 70%
SFP-factor of ventilation	$\leq$ 1,5kW/(m^3/s)	$\leq$ 2,0kW/(m^3/s)
Air leakage per 50Pa pressure difference	$\leq$ 0,60h$^-$1	$\leq$ 1,0h$^-$1

reports[3] about the criteria of low-energy consumption buildings and passive houses the power demand for domestic hot water and internal heat loads can be investigated.

Table 4: Power demand requirements for different applications

	Power demand [W/m^2]	Power demand [kWh/m^{2}year]	Heating subsidies[W/m^2]
Lightning	1,95	11,4	1,95
Equipment	3,00	17,5	1,80
Hot water	5,10	29,8	0,00
Persons	2,00	-	1,50

Different types of buildings have different energy demands which depends on the habits of people and the size of the building. In table 5 specific energy demand of different building types is listed given as kWh/m^2[86].

Table 5: Heating demand for passive houses and low-energy consumption buildings

Yearly mean temperature	Building type	Heating demand [kWh/(m^{2}year)]
$\geq$ 6,3°C	Passive house	15
$\geq$ 6,3°C	Low energy building class 1	30
$\geq$ 6,3°C	Low energy building class 2	45

Indoor climate requirements
The human body needs to maintain a certain temperature and uses energy for this purpose. It exists certain parameters which influence our temperature such as the ambient temperature of the air, radiation temperature, activity level and clothing. The air temperature combined with radiation temperature constitutes what we call *operative temperature*. The operative temperature is the uniform temperature of the surrounding surfaces and air. This temperature will give the same heat output as for the human body. The temperature can be calculated as the arithmetic means of average radiation temperature and air temperature given in equation 4.

$$t_{op} = 0,5(t_a + t_r) \tag{4}$$

In equation 4 the t_a and t_r represents the air and radiation temperature. The optimal temperature depends on the building construction and activity of the people living in it. The temperatures can be measured directly.

Optimal distribution between air temperature and radiant heat
A Japanese study[85] from 2009 presents the optimal relationship between air temperature and radiant heat that gives the best thermal comfort. The lowest exergy consumption occurs when the air temperature is about 18-20°C and the radiation temperature is about 23-25°C. This indicates that low temperature radiant heat is comfortable for the human

body. Such low temperatures can be combined with good thermal insulation, heat recovery of ventilation air as well as other low energy and passive house solutions. Radiant heat systems have higher investment costs than passive heating through the ventilation system for passive houses, however it is an alternative to the traditional heating in the floor and spot heat sources.

Ventilation
A building has ventilation system for two different reasons. The first one is to obtain a good air quality, which means achieving a comfortable and healthy air which does not contribute to health issues. We need fresh air which is not polluted by animals, materials and humans. Sometimes indoor excess heat is also considered to be polluted air which needs to be removed by ventilation. The second reason why to have a ventilation system is to remove moisture from the building. The moisture can contribute to fungus damage and mold in the building construction. It is also necessary to remove unwanted particles from the incoming air and avoid cold air by tempering the supply air. The simplest, most economical and operational ventilation system is the one with filtration of fresh air and heat recovery of exhaust air[85].

2.4 Heat distribution inside the buildings

When considering low-temperature district heating systems it is of importance to recognize new ways of designing buildings which are compatible to such a system. This chapter contains information about underfloor heating and thermal active building systems. Chapter 2.4.2 is based on the project thesis[1].

2.4.1 Underfloor heating

Underfloor heating (UFH) is a heating system of low temperature which is suitable for low-energy buildings. The large surface represents a large thermal capacity, and due to this large surface over-heating can occur if the temperature exceed the set point due to for example an increase of internal heat load or solar radiation. The thermal resistance of the upper layer of the floor will have an impact on the heat output of the UFH system, and the temperature of the water being circulated must be regulated accordingly. The transfer performance, as well as the conductivity of the floor material can be measured for obtaining the optimal material with high thermal conductivity[62]. The water temperature for UFH is usually set to 30°C, and the temperature of the room is set to about 25°C. However, the respond time of such a system is slow[85].

Benefits of using UFH
A system utilizing the concept of underfloor heating can contribute to an even more comfortable environment in the building. The source of heat will be distributed at a bigger area, and therefore the heat is distributed to the room in a more even way[66]. According to Sintef Byggforsk[85] it is possible to regulate temperatures individually in each room, and furniture can be placed regardless of heating devices. According to a study by Maivel and Kurnitski[63] there will be a correction of the operative temperature when using radiators by 0,25K, but no correction factor when using UFH. It is also of interest to investigate the impact on heat pumps when investigating UFH systems. The heat pump performance is affected by the return temperature of the UFH system. In a study[64] it was confirmed that a low return temperature would result in the highest heat pump efficiency.

2.4.2 Thermal active building systems

When designing new buildings a technology called thermal active building systems (TABS)[2] or Building-integrated thermal energy storage (BITES)[6] can be implemented. This is a system which reduces the CO_2-emissions as well as it creates a comfortable indoor air quality. The system regulates itself and consists of concrete floor and ceiling with integrated water pipes. Through those pipes, both warm and cold water floats to either heat or cool the building. Over 50 % of the heating happens through radiation from the floor and the roof, and the surface temperature should be the same as the temperature in the room. The water in the system requires 35°C to heat the building, and 15°C to cool it down. One additional solution to this system is to implement thermal energy storage and use the thermal energy in periods where the energy demand of the electrical grid is high. A TABS system allows the installations in the building to be minimal, due to the installations being implemented in the ceiling and floor. However, there will be a limit to the amount of energy such a system can generate in a room. In average, the generated effect is normally between 40 and 50 W/m^2 and it will be difficult to provide different temperatures for individual rooms. TABS is sustainable for new buildings often referred to as low-energy, passive or near zero emission buildings. Such a method for heating and cooling housings cause more energy savings and are more energy efficient than conventional air conditioning. The system is commonly used for office buildings where the rooms require the same temperature levels[2].

Optimal design
When designing a TABS system one have to consider following five critical parameters: thermal energy storage capacity, active charge capacity, thermal output capacity, and thermal output time lags and magnitudes. By using a frequency domain analysis the last two parameters can be calculated. The analysis can investigate which signal lies within the different frequency bonds. The other parameters can be calculated by inspecting the construction of the building, as well as how the buildings operates in active mode[6]. The operative temperature which is required to maintain a thermal comfortable climate can be determined by the thermal capacities, and is calculated as in the given equation 5[5].

$$ORT = A * t_a + (1 - A) * t_r \tag{5}$$

When calculating the operative temperature of the TABS system, the mean air temperature (t_a) and radiant temperature (t_r) are required as well as the heated area (A)[5].

Comfort of the inhabitant
TABS can have a rather undesirable acoustic. When designing TABS it is of high importance to consider the comfort of the inhabitant. Such a system as described needs to evaluate the use of sound absorbing surfaces. A coating which covers the whole surface will have a impact on the thermal comfort. By covering the roof areal by 35% with absorbing material the cooling capacity will have a reduction by 12% according to a master thesis by Pittarello[5]. With a coverage of 70% the reduction of the cooling capacity would be equal to 22%.

2.5 Domestic hot water production

Heating systems of buildings and domestic hot water (DHW) production covers 80% of the total consumption according to Federal Office for the environment of Switzerland[42]. This chapter will provide information about domestic hot water production considering water quality, methods of heating and reuse of tap water. Chapter 2.5.1 is partly reused from the project thesis[1], while chapter 2.5.2 and 2.5.3 is based on the project thesis.

2.5.1 Specifications of water temperature and quality

The most common way of preventing the legionella bacteria is to regulate the temperature to a specific level[81]. The temperature of the water should be 65°C, due to the legionella bacteria[40]. Above this temperature, the bacteria can not grow properly. We also do have to consider the acidity of the water, due to the bacteria mentioned. It grows in the range of 5,5 -9,2, the optimal number of acidity being 6-7. Norwegian water have a pH value in the range of 7,5-9,0. The amount of oxygen in the water which is optimal for the bacteria is 2,2mg/L, and the oxygen amount can be drained to a lower amount in the system. The amount of metals is also important to investigate as iron, zinc, magnesium and calcium makes the legionella bacteria grow better, while lead and copper is preventing the growth[40]. A low velocity of water flow results in more types of microorganisms and nutrients which the legionella bacteria feeds on. This can be avoided by designing the system in such a way that makes the standby time as short as possible. Another factor which can decrease the growth of the bacteria is by placing two hot water tanks in series instead of parallel. The bacteria grows better in accumulators and preheat boilers compared to battery heaters. In systems where water is heated according to the demand, the growth of bacteria is less widespread. The pipes of the system must be isolated to prevent temperatures preferred by the legionella bacteria, and over-sizing of the system can cause low velocity and sludge deposits. Under-sizing can cause high velocity and corrosion. Dead ends of the pipework should be avoided due to accumulation of oxygen pockets in the system, and every outlet station for tapping equipment should include filters due to the temperatures in these circumstances[40].

It exists however some alternative solutions in addition to temperature level control to prevent the legionella bacteria. Three methods can be used for the treatment which is thermal, chemical and physical treatment[81]. Thermal treatment is to keep the temperature of the water above 65°C, which is a simple method. However, a LTDH system required additional devices to lift the temperature to the desired temperature level. The chemical treatment consists of ionization, chlorine, chlorine dioxide, UV-light and photocatalysis. This method requires that the water must be controlled on a regular basis in order to avoid violating the water quality. The physical treatment consists of filtration which will contribute to the bacteria not entering at all. The filtration method have operational costs which is much higher that other methods due to the filter being replaced on a regular basis[81].

2.5.2 Direct and indirect heating

Direct heating
A direct heating system is well suited if the building is supplied by district heating or gas. A model of a direct heating system is shown in figure 7 which is inspired by a book[40]. Small changes of the amount of water flow in such a system can reduce the effect and energy demand a lot, and it is not common to heat water directly this way due to this issue[40].

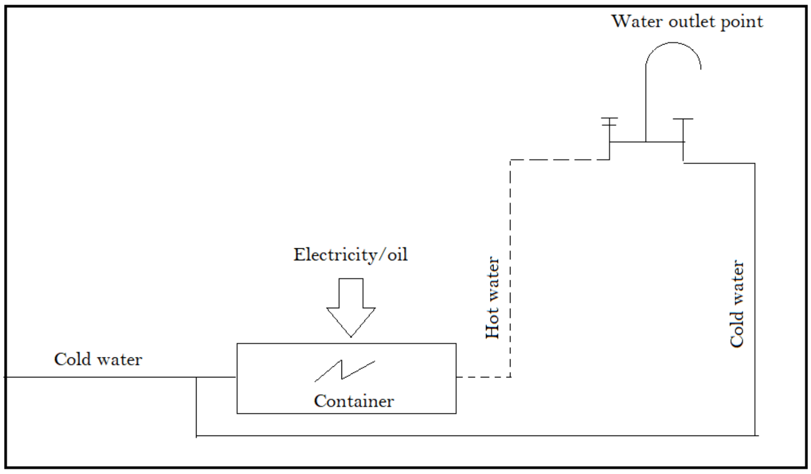

Figure 6: Direct heating of tap water

Indirectly heating
If the energy supply is based on electricity, a hot water tank is often preferred. The main goal with a hot water tank is to accumulate hot tap water which can cover the demand at all times[40], and below a system sketch is presented in figure 7 inspired by a book[40]. Such a system can make sure of DHW during peak demand hours[42], and the system is able to obtain power savings.

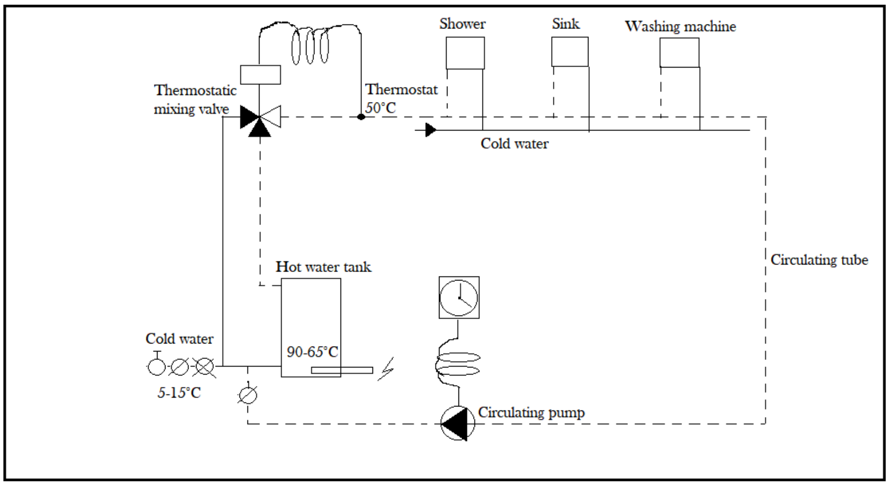

Figure 7: Accumulation of hot water

2.5.3 Heat pump water heater application

DHW production in low-energy and passive houses in Scandinavia constitutes about 50-80% of the total energy consumption of the building[79]. Due to this an implementation of heat pumps for DHW production contributes to energy savings in new buildings. The energy consumption of a CO_2 heat pump water heater (HPWH) can reduce the energy consumption of the DHW production. Compared to the conventional electrical or gas based systems, a CO_2 HPWH can reduce the energy consumption with 75% when the temperature of domestic hot water is set to 60°C. This saving potential is based on experiments[72] at Norwegian conditions with temperatures in which water was heated from 10°C to 60°C with ambient air as heat source[72]. As mentioned in chapter 2.4.1, the COP of the heat pump will describe the efficiency of the heat pump. By investigating the COP of the heat pump it is possible to do calculations on the energy saving potential of implementing a heat pump instead of a conventional boiler for production of DHW. Further on, it is possible to do a cost analysis on the HPWH system.

2.5.4 Reuse of tap water

Greywater (GW) is a combination of waste water coming from showers, WC basins, bathtubs, dishwashers, kitchen basins and washing machines[80]. By utilizing the waste heat coming from greywater it is possible to harness heat which is non-industrial. On a daily basis a person produces in average 136L of GW, and about 64% of the water consumption is GW with potential for heat harnessing. The GW temperatures varies a lot depending on the user living in the building and their habits. According to studies[80] a general rule is that the output GW is between 5-10°C lower than the input temperature. In China, an average temperature of 32,5°C of GW was measured at a hotel during a summer. Below, a list of different temperatures and amount of water used is listed for some applications[80]:

- Shower: 40-50°C, 12L/min
- Washroom basin: 50-60°C, 2L/min
- Dishwashers: 60-85°C, 10-25L per wash

2.6 Heat pumps and different refrigerants

This chapter contains information about heat pumps which provides natural working fluids due to the regulation of refrigerants provided by the European Union F-gas directive. The chapter includes information about heat pumps in general as well as heat pumps with different refrigerants such as Ammonia, Propane and CO_2. All of the subsections is partly reused from the project thesis[1].

2.6.1 Heat pumps

Heat pumps use less energy than conventional systems for heating, and can be applied both for heating and cooling of residential buildings and other purposes. Using heat pumps is the most energy efficient method of heating[41]. Compared to the conventional methods of heating systems the heat pump technology can reduce the energy consumption by 50-90%. In addition the energy supply to the heat pump is either waste heat from another process or a renewable energy source such as groundwater, seawater or rocks[53]. It exists three types of heat pumps, and they are listed below.

- Air-to-air
- Water-to-air
- Water-to-water

The choice of heat pump is dependant on what kind of operation the heat pump will provide. The choice of type depends on where the heat is being taken from and where it is being delivered to.

The heat pump concept and applications
Heat flows from an area of high temperature to an area of low temperature. However, by using a heat pump it is possible to force the heat flow to go in the other direction by introducing an amount of energy into the heat pumps[41] and gives the heat pump the ability to operate with a reverse technique compared to a heat engine. It is required to give the heat pump external drive force to enable such a process, and force the extracted heat from the surroundings into a sink of heat. In theory, the energy delivered to the desirable area should be the same amount of energy extracted from the nature in addition to the input energy for the driving process[46]. Heat pumps can be utilized for following purposes[53]:

- Space heating (floors and ceilings, ventilation air)
- Domestic hot water production (for residential buildings or industrial purposes)
- Space cooling (water for industry, computer cooling, cooling of residential buildings)

The heat pump have two different application areas which is heating and cooling; the heat pump process and the refrigeration process. Figure 8 shows a sketch of the four main units in a heat pump and is inspired by the compendium of the subject TEP4255[56].

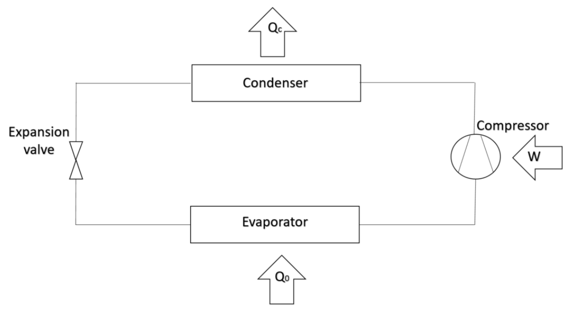

Figure 8: Heat pump process

The difference between the two processes is shown in figure 9 at the same ambient temperature level. In the heat exchangers (evaporator and condenser) the working fluid is in a two-phase stage, being a mixture of both liquid and vapour. The working fluid temperature is dependant on the pressure in the system, and the ambient temperature outside the heat exchangers have a huge influence on the process[56].

The temperature outside the heat exchanger determines if the heat pump operates as a Carnot heat pump process or a refrigeration process:

- Heat pump: The temperature of the working fluid is lower than the temperature outside the heat exchanger. Fluid will evaporate and extract heat from the surroundings
- Refrigeration: The temperature of the working fluid is higher than the temperature outside the heat exchanger. Fluid will condense and deliver heat to the surroundings

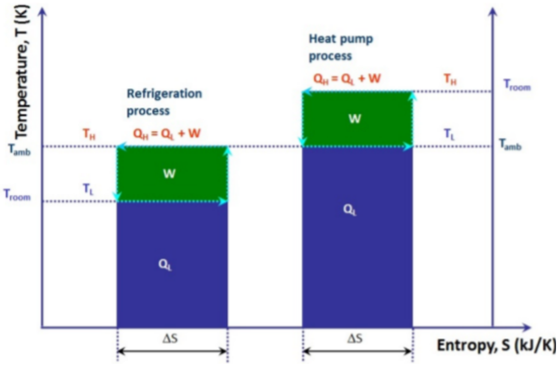

Figure 9: Carnot refrigeration process and heat pump process[56]

Refrigerants

The duty of a refrigerant or the working fluid in a heat pump is to circulate in the sealed processing plant and absorb heat from the cold side as well as delivering cold to the warm side[56]. It exists a great selection of refrigerants which contributes with ozone depletion (ODP) and global warming (GWP) such as CFCs, HCFCs and HFCs. HFCs have replaced CFCs and HCFCs due to their ozone depletion, however HFCs still contribute to the global warming[26]. According to the European Union F-gas directive, HFCs are being phased out by 2030[18]. Due to the attention directed to zero ODP and negligible GWP, the use of natural refrigerants in heat pump systems have become more attractive. Ammonia, hydrocarbons and CO_2 are examples of natural refrigerants[26].

Explanation of components in a refrigeration system

The closed process of refrigeration consists of four main units: evaporator compressor, condenser, and expansion valve. As the working fluid enters the evaporator the working fluid is in two phases; liquid and gas. The pressure in the evaporator is low since we want a low temperature (lower temperature than the frozen storage medium temperature). In this way, the heat from the cold storage (Q_0) will flow to the evaporator which works as a heat exchanger. The liquid in the process is being evaporated after the working fluid has absorbed the heat. This leaves us with pure vapour. The evaporation process is known to be an isobaric process where the pressure is constant. Further, the working fluid enters the compressor. The compressor maintains the low pressure in the evaporator by removing the working fluid vapour which is evaporated. The working fluid is being compressed to a lower pressure level which makes the temperature increase as well to a temperature higher than the ambient temperature. To compress the working fluid, mechanical work is needed. The work input (W) is usually generated from an electric motor. After the compressor, the fluid which now has a high pressure and temperature as vapour enters the condenser. The condenser temperature is higher than the ambient, and therefore heat is given off to the environment. The working fluid leaves the condenser as saturated liquid, and the pressure in the condenser is constant which indicates that this is an isobaric process. Finally, the fluid enters the expansion valve. This valve reduces the pressure and thereby temperature and returns the working fluid to the evaporator as a mixture of liquid and vapour. The pressure difference over the expansion valve defines the ratio of gas/vapour. By using an expansion valve, it is possible to regulate the mass flow coming through and thereby the right amount of working fluid to the entrance of the evaporator. The cycle is repeated[56].

Heat pump modes

The heat pump have different modes; the *monovalent* mode and the *bivalent* mode. A heat pump which operates with only one source of energy is operating in the monovalent mode, whereas heat pumps operating with several heat sources and do have an electric resistance is operating in the bivalent mode. Heat pumps in a large scale, such as industrial heat pumps, do work in the bivalent mode due to utilization of combustion furnance and other heat sources, and are becoming more popular due to their high performance. An electrical heat pump can supply a heat load of 100kWh with an input of 20-40kWh of electricity compared to the industrial HP which only uses 3-10kWh[46].

Coefficient Of Performance

The efficiency of the heat pump is defined by the coefficient of performance (COP)[56]. The COP is determined below in equation 6, and determines the relationship between the output of delivered energy (Q_c) over the amount of electrical energy input to the compressor (W). However, the calculation of the COP is not able to take into account the defrosting processes which should be done when the temperature outside drops below 6,4°C. If the heat pump builds up a barrier of ice on the coils the COP will decrease.

$$COP = \frac{Q_c}{W} \tag{6}$$

Another factor which decreases the COP of air-to-air heat pumps is the inability to provide enough heat at cold days[46]. If the temperatures are expected to be low, a heating system as a back-up plan should be implemented. The back-up heat are often electricity-based, and when the temperature reaches the balance point the supplementary energy source is being utilized. When calculating Q_c given in equation 7 and W given in equation 8, we need to look into the thermodynamic properties of each specific refrigerant. In figure 10 the ideal process is shown in a pressure-enthalpy diagram[56].

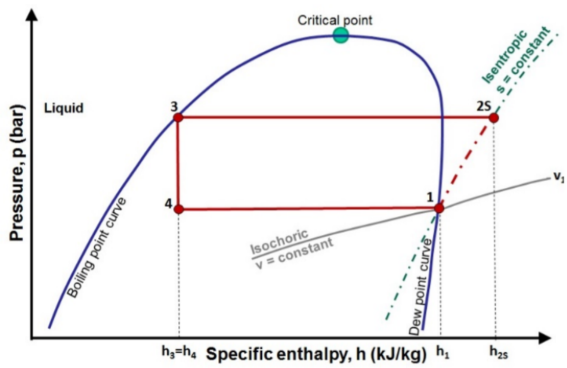

Figure 10: Heat pump process in p-h diagram[56]

In figure 11 the ideal heat pump process is shown in a temperature-entropy diagram. The calculation of Q_c is dependant of the enthalpy values of h_2 and h_3 in the process[18]. The value of h_2 represents the real process, while h_{2is} in the diagrams represents a loss-less isentropic and adiabatic process. The calculations regarding a real process do need recalculations of h_2.

We need the values of the mass flows for water (m_w) given in equation 9 and for the refrigerant being used (m_R) in the heat pump given in equation 10 to solve equation 7 and thereby equation 8[18].

$$Q_c = m_w * c_p * \Delta t = m_w(h_2 - h_3) \tag{7}$$

$$W = m_R * w = m_R * (h_2 - h_1) \tag{8}$$

$$m_w = \frac{V_w}{\rho_w} \tag{9}$$

$$m_R = \frac{Q_c}{q_c} = \frac{Q_c}{w} \tag{10}$$

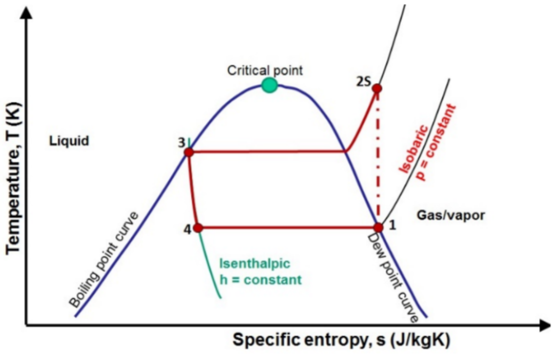

Figure 11: Heat pump process in a T-s diagram[56]

Superheat
The definition of evaporator superheat is a temperature difference between the evaporator saturation temperature and the evaporator outlet temperature[58]. It is desirable to have a low amount of superheat present in the heat pump system due to a high reduction of efficiency as a consequence of reduced capacity of the system. It is however important to maintain some superheat at the outlet of the evaporator to ensure that the refrigerant have vaporized entirely. If this is not the case, the compressor can undergo serious damages. This case can cause damages due to the fluid entering the compressor in a mixed phase of liquid and vapor. It is therefore important to measure the superheat and regulate it[58]. A suction superheat control of about 5K is used in normal heat pumps[57]. Another term for superheat is the total system superheat which is all the superheat measured from the outlet of the evaporator and to the inlet of the compressor. This is the superheat in front of the compressor and depends on which components being used in the system. An internal heat exchanger can increase the system superheat[56].

Internal heat exchangers
An internal heat exchanger (IHX) can be placed at the outlet of the gas cooler as a tube connected to another tube placed at the inlet of the compressor. IHX have several purposes. It can subcool the liquid at the outlet of the gas cooler as well as superheat the gas which appears at the inlet of the compressor. The superheat losses will increase and in return the expansion losses will decrease by utilizing such a unit[56]. The system sketch is shown in figure 12.

Heat pumps and district heating
Heat pumps in combination with a DH system contributes to the development of smart grids[39]. It is preferred to have a low supply temperature due to this being profitably, and by implementing a heat pump this will have a great impact on the efficiency of the system. A general rule is that the COP of the heat pump will increase by 5% or more if the supply temperature is lowered by 5 degrees. If we connect two heat pumps in series, this will be another way to gain a higher COP, in which both heat pumps will operate with a smaller temperature difference. The COP will be influenced by the DH return temperature, and a lower return temperature is favored due to a lower condensing temperature for the first heat pump in the series.

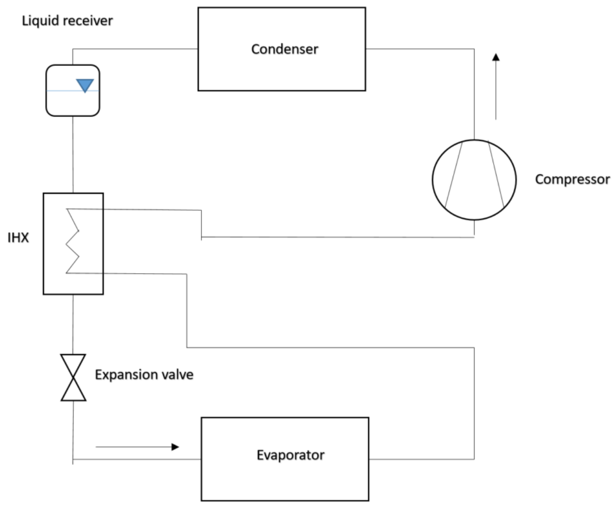

Figure 12: Internal heat exchanger in heat pump

2.6.2 Ammonia heat pumps

Ammonia as refrigerant
Ammonia have a molar weight which is 75% less than the molar weight of HFCs[54]. Ammonia have a high critical temperature and pressure of 132,3°C and 113,3bar as well as a low normal boiling point (NBP). The density of both the liquid and vapour of this fluid, ρ_L and ρ_v, is low compared to the HFCs. Ammonia have very high values of specific enthalpy of evaporation (Δh) compared to other working fluids. The properties of ammonia enables the ammonia heat pump to cover a wide range of demands; both refrigeration and heat pumping applications. Figure 13 shows the large Δh compared to the Δh of HFCs.

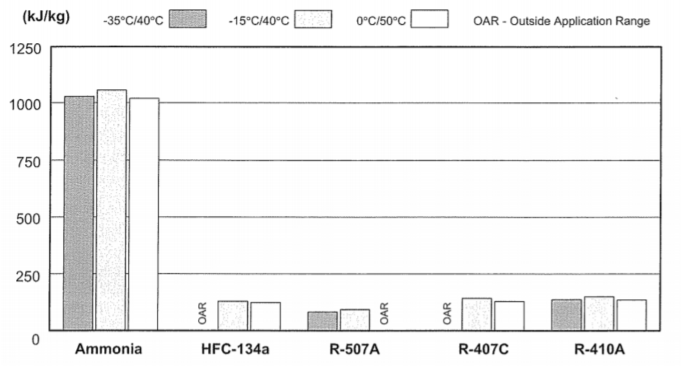

Figure 13: Specific enthalpy of ammonia and HFCs[54]

We do need to consider the pressure ratio in the compressor. The pressure ratio (defined by the condensation pressure over the evaporator pressure) influences the discharge gas temperature and the isentropic and volumetric efficiencies of the compressor[54]. In figure 14 a comparison of a two-stage cycle of ammonia and HFCs is shown. We recognize that the pressure ratio of ammonia with respect to different evaporation and gas outlet temperatures is lower in average compared to the other refrigerants. OAR means *outside application range*.

System design
The low molar weight of ammonia will have a positive effect on the sizing of valves,

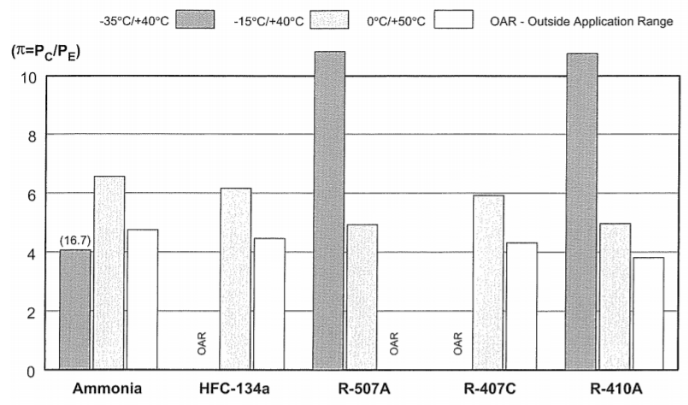

Figure 14: Pressure ratio of ammonia and HFCs[54]

compressors and pipelines in the system as well as the heat transfer properties, making the system a more compact system[54]. The mass flow rate is proportional to the specific enthalpy of evaporation. Due to the specific enthalpy of evaporation being big for ammonia the mass flow rate will be low, and compared to HFCs, it will be reduced by 15%. When designing a low or medium capacity heat pump with ammonia, a low mass flow rate in the system can be a challenge. When we do consider a direct expansion evaporator system, a stable control of the liquid fed into the evaporator is difficult to maintain. We do want a constant low superheat at all times during the operating time, which makes a stable control of liquid important. The low density for both liquid and vapour ammonia have a direct connection to the pressure drop in the components, pipelines and valves of the systems, and it results in low pressure drops.

Safety measures
Machine rooms containing an ammonia heat pump need some safety measures. It is always a risk of leakages. Ammonia has a pungent odour which can cause panic, as well as risk of fatal injury and intoxication[54]. Therefore, following requirements should be applied for the machine room containing an ammonia heat pump.

- The machine room should be localized on the roof or top floor of a building (in a container) to prevent dispersion of the fluid to the public. Restricted access, only authorized personnel
- Gas-tight machine room. Should prevent the fluid from circulating to other parts of the building
- Sufficient number of emergency exits (self-closing and fire-proof doors)
- Use of fire-proof materials in the building structure (walls, ceiling and floors)
- Ammonia leak detectors above heat pump. Will activate a visual and audible alarm system. Disconnect electrical equipment
- Fail-safe ventilation system with two-speed fans. When concentration of ammonia exceeds 500ppm, the fans are switched on to maximum capacity
- Ammonia absorption systems. Ammonia vapour is highly absorbed in water. Example of systems are scrubber or sprinkler system.
- Emergency lighting, fire extinguisher, and personal equipment for protection

Example of residential heating and cooling with ammonia
A district heating and cooling system was designed in 1991 for space heating and cooling,

as well as domestic hot water production for a university in Bergen[54]. The design conditions for heating and cooling capacity was set to 3.5MW and 1,5MW. In this project a seawater heat pump was used as a heat source for the ammonia heat pump with a water temperature of between 15-25°C. The costs of the ammonia heat pump versus a HFC heat pump was at this time 30% higher. Even though the costs of the heat pump was higher, the properties of ammonia made the COP increase with about 20-25% compared to the COP when using HFCs as working fluid. The system sketch is shown below in figure 15.

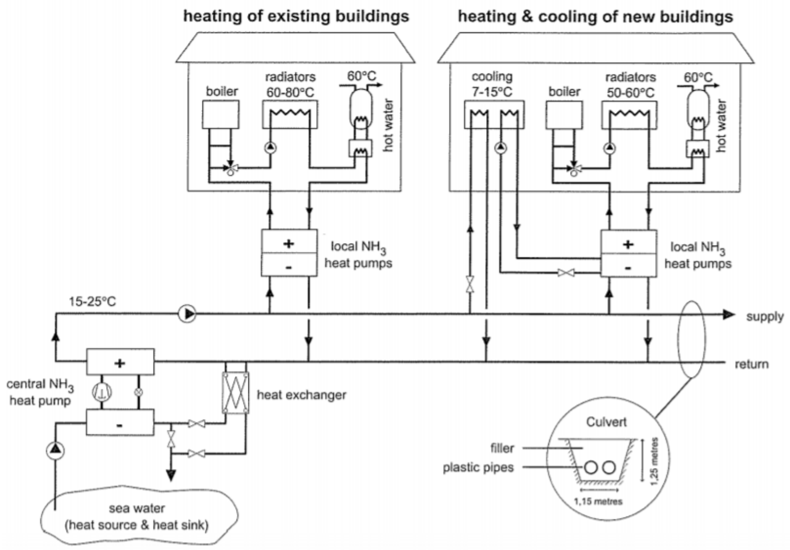

Figure 15: District heating and cooling system[54]

2.6.3 Hydrocarbon heat pumps

Hydrocarbons as refrigerants
The most common hydrocarbons used for heat pumping systems are ethane, propane, butane, iso-butane, propene, CARE 30 and CARE 50[54]. In table 6 the hydrocarbons are classified considering the different application areas.

Table 6: Different hydrocarbons and its application areas

Hydrocarbon type	Application area
Etane	Low-temperature systems, < 80°C
Propane	Low/medium-temperature refrigeration, heat pump and AC systems
Butane	High-temperature systems
Iso-butane	Domestic refrigeration and freezer systems
Propene	Low/medium-temperature refrigeration, AC and heat pump systems
CARE 30	Performs similar to CFC-12 and HFC-134a
CARE 50	Performs similar to HCFC-22, R-502, R-407C and R-404A

Safety measures
Hydrocarbons can be flammable in contact with air. As a cause of this there must be implemented some safety measures to the heat pump system[54]. In most workplaces,

as well as other environments, flammable liquids are to be found. Hydrocarbons can be found in heating fuels, alcohol, paints and more.

Flammable liquids should be packaged, and the following *main* safety measures must be applied for a heat pump system containing hydrocarbons as working fluid[55]:

- Leak-tight, robust system
- Safety of equipment that uses or comes in contact with flammable atmospheres
- Workers that gets in contact with flammable atmospheres should be protected

2.6.4 CO_2 heat pumps

In particular there are two characteristics regarding CO_2 as refrigerant which makes the medium quite special: the critical temperature and the high working pressure required under heat pump conditions[24]. Compared to other refrigerants used in heat pumps, CO_2 has a remarkable low critical temperature of 31.1°C. The critical temperature indicates the upper limit when condensing vapor and deliver heat. To get a sufficient amount of latent heat per unit mass, it is necessary to make the working fluid condense at a temperature level below the critical level[26]. The low critical temperature of the refrigerant permits the heat pump to operate in a transcritical stage which allows the gas cooler to operate above the critical pressure.

The transcritical process
In a transcritical stage the heat delivery temperatures are no longer limited by the critical temperature, which includes operating the evaporator below the critical temperature as well[24]. In vapour compression systems working in normal ambient temperatures this refrigerant can additionally operate closely and also above the critical pressure of 73.8 bar. As mentioned the refrigerant requires a high working pressure in comparison to other refrigerants, and CO_2 operates at a pressure level of 60-70 bar in a subcritical stage and at 80-110 bar in a transcritical stage. Although high pressures can have an impact on the compressors and its components and capability, high pressures also have its advantages such as keeping CO_2 at a relatively high volumetric heating capacity and high vapor density level[24]. Due to the properties of CO_2 it is possible to achieve a high COP of the system. An important characteristic with supercritical fluids is that the heat transfer in heat exchangers can be difficult to calculate due to the value of specific capacity of the fluid varies a lot. Therefore, the logarithmic mean temperature difference (LMTD) can not be calculated correctly. As a result it is easy to under-size the gas cooler of the heat pump. When sizing the gas cooler data simulations should be implemented and it should be divided into many number of sections which should be investigated[24].

Application areas
A study from 2004 considering the COP and thermodynamic efficiency of space heating and domestic hot water production shows that using CO_2 as working fluid in heat pumps is considerable better for production of domestic hot water (DHW) than for space heating[53]. The results of the measurements of two residential CO_2 heat pumps is shown in table 7.

Table 7: COP evaluation of CO_2 heat pump for different purposes

Heating demand	T_0	T_{2S}	T_3	COP_{HP}	COP_{LZ}	η_{LZ}
Space heating	-5°C	35°C	30°C	3.0	8.15	0.37
DHW heating	-5°C	70°C	6°C	3.6	7.40	0.49

As shown in table 7 it is noticeable that the efficiency increases with 30% when the goal is to produce DHW compared to the space heating[17]. Below there is a list of reasons why this is the case[18]:

- Good temperature fit regarding the counter-flow gas cooler and CO_2 vapour. Large temperature glide for CO_2 and water which means low average temperature during heat rejection
- High isentropic compressor efficiency, and therefore low pressure ratio due to the high pressure level in evaporator and gas cooler
- Low average temperature difference in heat exchanger due to superior heat transfer properties for CO_2 results in high evaporation temperature and reduced optimum pressure in gas cooler
- Able to supply high temperature heat 60-90 degrees celsius, no need for reheating with electric immersion heaters or other peak load sources that has a COP < 1.
- Especially designed for high temperature water heating. Long lifetime for compressor and other components.

Figure 16 shown the temperature glide between vapour CO_2 and water in which CO_2 is being cooled, and the water is being heated. The best temperature fit between CO_2 and water is when the curves are parallel.

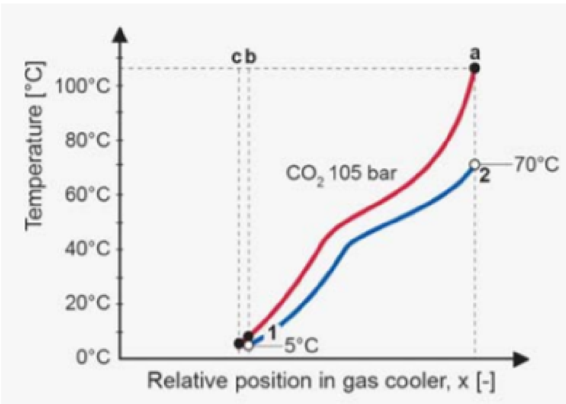

Figure 16: Temperature glide of CO_2 heat pump for DHW production[53]

Optimization of design

By operating the system in a transcritical region we get rid of the issue concerning a low critical temperature. The critical temperature of a CO_2 system makes it impossible to transfer heat to the ambient above this critical temperature by condensation[34]. By replacing the condenser with a gas cooler can solve this problem. When heat rejection happens over a large temperature glide it is easier to control the capacity, as well as it opens up the possibility of hot water heating/steam production and simultaneous refrigeration[33]. Figure 17 show both the real and isentropic process of a transcritical CO_2 heat pump cycle, in which the large temperature glide is shown. Due to the special behavior of CO_2 the slope of the isotherms beyond and around its critical point is modest for a specific pressure range. Otherwise, when looking at the isotherms outside this range, the isotherms are steep.

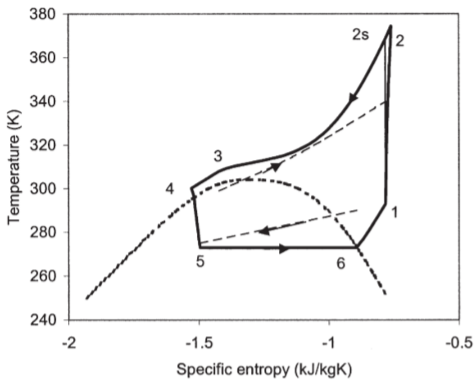

Figure 17: T-s diagram of transcritical CO_2 heat pump cycle[33]

Another important factor to investigate is how cold affects the gas cooler outlet temperature. The isotherms becomes flatter with a cooler outlet temperature, and therefore this outlet temperature is of big interest while trying to estimate the optimum operating conditions. A study[33] made by J. Sarkar et al. showed that the evaporator temperature has a big impact on the COP, as well as the gas cooler outlet temperature, compressor efficiency, compressor discharge pressure and heat exchanger effectiveness which is shown in figure 18 and 19.

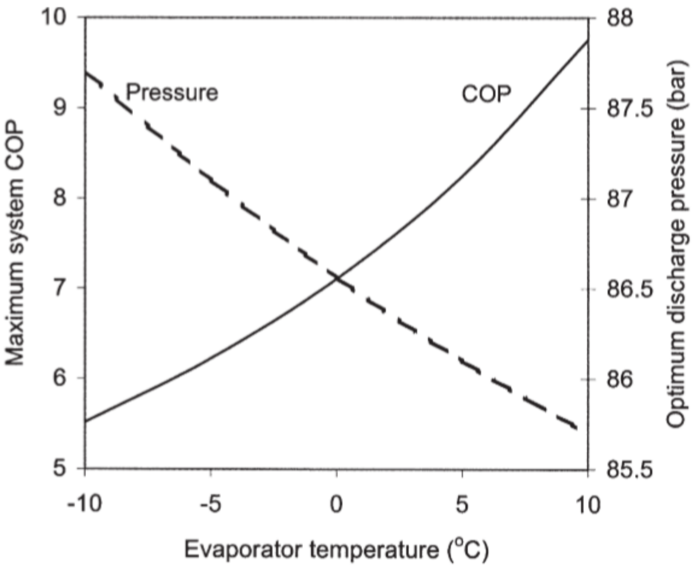

Figure 18: CO_2 heat pump cycle and varying evaporating temperature[33]

Figure 18 and 19 are an outcome of a system model based computer code “CO2PROP” made to estimate thermodynamic properties of CO_2 in both sub-critical and super critical stage. The model is developed by Span and Wagner which rely on another extensive study based on experimental data for thermodynamic properties of CO_2[33]. Ideally, it is possible to reach a COP of about 7.0 when optimizing the values of discharge pressure and the evaporator temperature. However, when comparing the cooler outlet temperature and the optimum discharge pressure, we get a lower possible COP of about 6.0 by using this model[24].

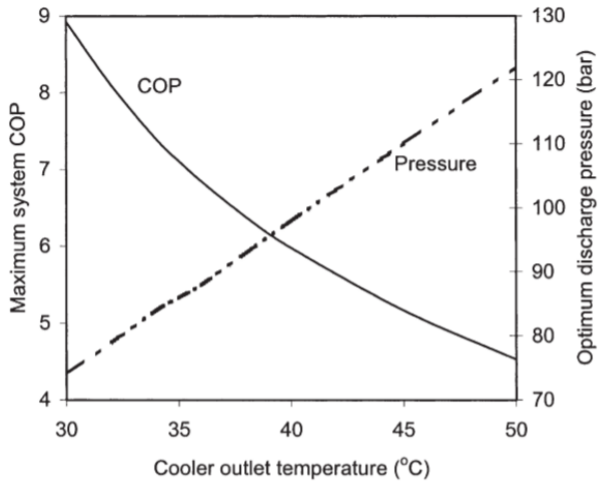

Figure 19: CO_2 heat pump cycle and varying gas cooler outlet temperature[33]

The conclusion of the impact of gas cooler temperature is that lower temperatures out of the gas cooler constitutes smaller enthalpy values and therefore a bigger heat capacity value which results in a better COP[17]. This reasoning is illustrated in figure 20.

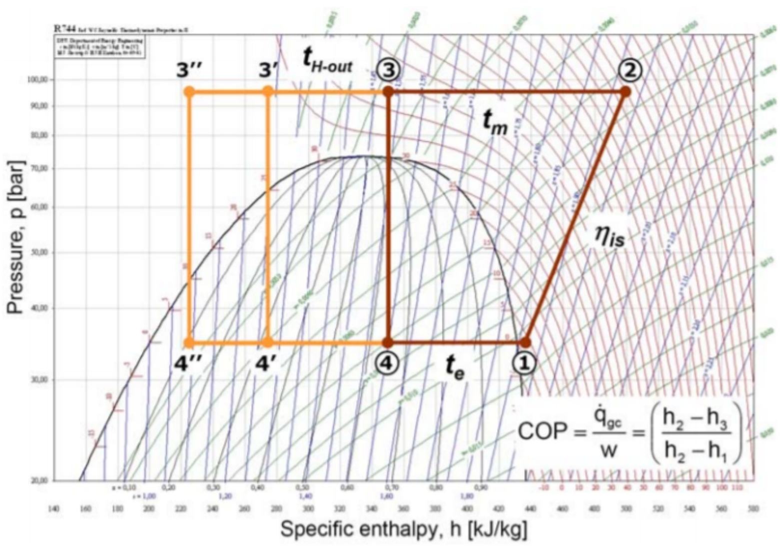

Figure 20: Heat pump cycle with different gas cooler outlet temperatures[17]

2.7 Thermal energy storage

This chapter contains information about thermal energy storage and its methods, as well as seasonal and diurnal thermal storage and costs of thermal energy storage. Chapter 2.7.1, 2.7.2, 2.7.3 is partly based on the project thesis[1].

2.7.1 Thermal energy storage in general

Another possibility for reducing the net energy consumption in a building is to implement thermal energy storage (TES). The general principle of TES consists of the following three steps: charge, store and discharge thermal energy. The process is shown in figure 21. This technology contribute to an increase of the performance for both heating and cooling systems by storing heat or cold and release it when heat or cold is desired. Another potential benefit with the thermal storage method is using the technology to reduce the power demand of the electrical grid during the peak hours. The implementation of TES is easy to implement in older buildings as well.

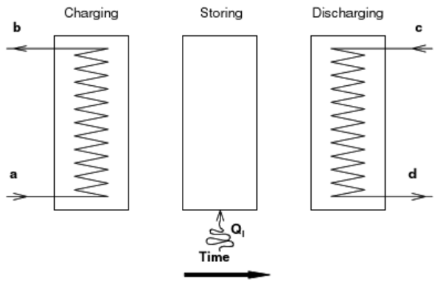

Figure 21: Thermal energy storage principle[47]

Depending on the circumstances, the technology can be implemented in the structures of the building or be placed as a separate unit. In addition, smaller air-handling units, ducts and variable air volume boxes makes TES a cost-effective alternative[47]. TES has many applications such as space heating and cooling, DHW production, and air conditioning. The classification of TES technologies is comprehensive, and will be discussed further in chapter 2.5.2 and 2.5.3.

Methods of TES
It exists many different methods of thermal energy storage. The most common methods are being discussed which is the sensible, latent and thermochemical heat storage. The *sensible heat storage* involves the transfer of heat to a storage medium, and the medium increases its temperature and stores the heat[9]. The storage medium being used is usually water, soil, stones, gravel or pebble[31]. Equation 11 and 12 indicates the amount of thermal energy which can be stored in the material[9].

$$Q = mc_p(T_h - T_i) \tag{11}$$

$$\rho = \frac{m}{V} \tag{12}$$

In general, the value of sensible heat is quite easy to find for solids and liquids, while for gases it is more complicated due to expansion of volume and increase of pressure when the temperature increases[47]. The letters of equation 11 and 12 have following explanation listed below:

- m - the mass flow of the fluid $[kg/s]$
- c_p - the specific heat capacity of the material $[J/kgK]$
- ρ - density of storage material $[kg/m^3]$
- V - volume flow rate of material $[m^3/s]$
- T_h - maximum temperature stored in the material [K]
- T_i - initial temperature in the material [K]

- Q - the capacity of storage

Sensible heat storage is known to be the most common storage method, and has several advantages such as being environmentally friendly, the material used is cheap, it is a relatively simple system, and the control system is easy and reliable[31]. The heat from such a storage can in addition be stored and released on demand[47]. The disadvantages of such a system is the low energy density, the huge volumes required, its self-discharge and heat loss problem, as well as the high cost of site construction and the geological requirements. The sensible heat storage is known to be a mature storage method, and it exists many test project in large scales which confirms it. However, further work must be done for this method such as optimize the storage temperature and control strategy to reduce heat losses and to advance the solar fraction and reduce the power consumption[31].

The *latent heat storage* uses organic and inorganic mediums[31]. It changes its phase at isothermal conditions which allows the material to store and release heat[47]. The method has its advantages such as higher energy density than sensible heat storage, and it can provide thermal energy at constant temperature as well. The disadvantage of this system is the lack of thermal stability, corrosion and crystallization which can occur. The storage materials being used do have high costs as well. This type of storage is being investigated by using laboratory-scale prototypes, and further it will be interesting to focus on and investigate screening for better suited phase change materials which has higher heat of fusion[31]. However, the method has obtained an increased interest due to the most effective measures of reduction in energy consumption[9].

The last method, the *thermochemical energy storage* considers the chemical aspect and the method uses storage mediums such as metal chlorides and metal hydrides. The process includes thermal sorption and chemical reaction, and the advantages which follows by the use of this method is the high energy density which is higher than for the two first methods. Further, it is also proven to be a quite compact system with negligible heat losses. However, there are some disadvantages as well such as the poor heat and mass transfer property under high density condition, the high cost of the storage material and the uncertain cyclability. This method needs to be considered further in the future by investigating how to optimize the temperature level during the charging and discharging process, as well as optimize the particle size and reaction bed structure which can provide a constant heat output[31]. The methods of thermal energy storage depends on the period of storage required, and is divided into diurnal and seasonal storage. The choice of method is also dependant of economy, and operating conditions[47].

2.7.2 Diurnal TES technologies

It is possible to moderate the short-term daily net load variation by building an integrated thermal energy storage into buildings, and thereby shift loads away from peak demand hours[31]. A short term storage can meet the load changes and provide less losses considering the generating equipment which starts and stops. Such a storage contributes to a boosted electric power output when increasing the heat extraction. The most common thermal energy storage technology is *the hot water storage tank* which is a *diurnal* or *short-term* storage technology. This technology is shown in figure 22.

Figure 22 shows a CO_2 heat pump connected to hot water storage tanks. The CO_2 heat pump operates in a transcritical stage, which operates at a significantly lower evaporation temperature than heat pumps with other working fluids. This special cycle was discussed in chapter 2.4.4. The storage is made due to the varying demand of hot water. The technology is based on the sensible heat storage method and is both reliable and cheap

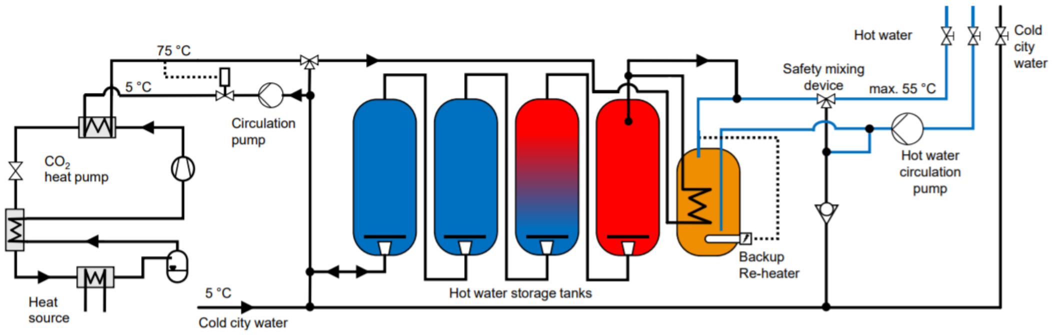

Figure 22: Domestic hot water storage tanks[65]

compared to other methods as well as it is a simple system. The height of the inlet and outlet of such a tank can be varied according to the available supply temperature and desired storage temperature[61].

Auxiliary heat sources
The most common way of heating domestic hot water is by using electricity. Solar energy in combination with heat pumps, biomass or fossil boilers are also being used. A solar system for domestic hot water production and space heating will improve the annual efficiency of a solar system with about 20%[61].

2.7.3 Seasonal thermal storage

The seasonal thermal storage is often used to store heat for winter season and cold for summer season, as well as the source functioning as a thermal storage for a ground heat pump[31]. Most of the technologies of the seasonal TES being used nowadays is based on the sensible thermal storage method due to renewable energy being the energy source and it is required to install large storing equipment to meet the energy demand for the district heating and cooling systems. The possible solutions of seasonal storage is many.

Aquifer thermal energy storage (ATES) is an energy storage technology where groundwater is often being used as a medium storing heat from the summer, and releasing it in the winter[52]. The system is mainly used for covering the energy demand of space heating and DHW production, and is often combined with a solar collector. It is proven that ATES can reduce the energy use with between 50 and 60% of the natural gas consumption, and the lifetime of the system is between 20 and 25 years. The groundwater typically have a temperature of between 5 and 10°C in the summer, and the system contains extraction wells, injection wells and a heat exchanger as shown in figure 23 which is inspired by a book[52]. When using groundwater as a storage medium, the temperature can rise to 30°C. The temperatures are depending on the wells installed in the system, and the capacity of such a system can be 15kWh/m^3. The water is being pumped from an extraction well and brought to a heat exchanger where heat is extracted from air or a fluid and brought into the injection well for storage while the cold from the groundwater is being supplied to the building as a cooling application.

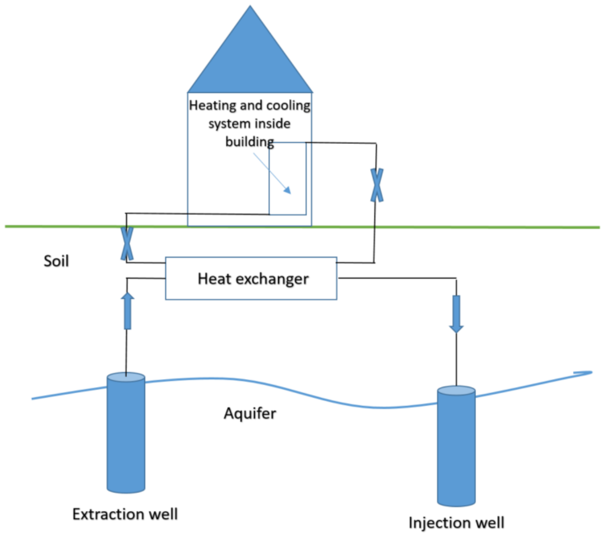

Figure 23: Aquifer TES

Borehole thermal energy storage (BTES) is another method of storage quite similar to ATES, although the design is a little bit different. The investment costs for BTES is big, there will be some heat imbalances in the mass of the ground, as well as disturbances in the ground and its hydrology. This system uses a structure of pipes which is of high-density polyethylene. This structure of pipes has a contact with boreholes in the ground, and typically glycol (or another heat transfer fluid which is of the antifreeze solution) is being used to extract the thermal energy from the ground, both heat and cold depending on the season. The system i shown in figure 24.

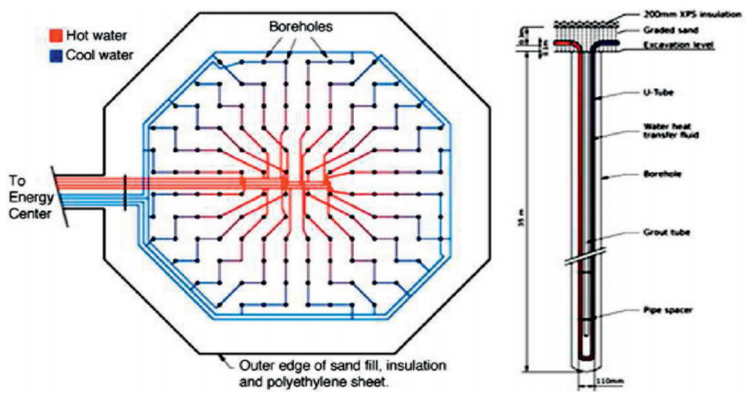

Figure 24: Borehole TES[52].

The system consists of several loops which is connected to the boreholes, and the structure is organized both horizontally and vertically. In the summer, solar collectors can be used to store heat in the boreholes. Cavern thermal storage is another method of seasonal storage as in which heat or cold is stored in large water reservoirs and later on distributed to where the energy demand is taking place[52]. It is of the sensible heat storage method, and it exists two commonly used systems of cavern TES; the hot water storage and the gravel/water storage. The storage takes place under the ground in an insulated tank in the form of a cavity or a pit, and water is being the storage medium for heating and cooling. The Cavern TES concept is shown in figure 25.

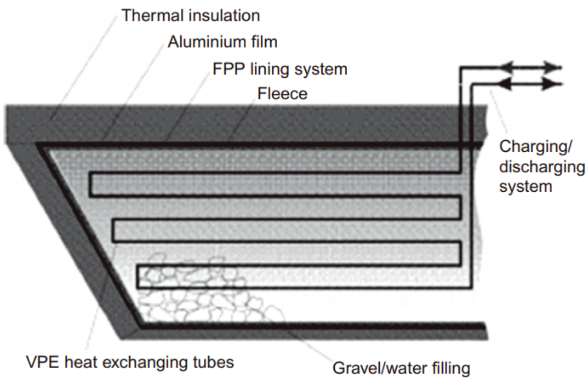

Figure 25: Cavern TES[52].

The method has a big potential considering the heat and cold storage of thermal energy, and the potential is depending on the thermal conductivity of the rocks, gravel or pebbles used. A Cavern TES in Steinfurt based on gravel and water, with a size of 1500 square meters have the quality of dealing with operative temperatures up to 90 degrees celsius and provides 34% of the annual heating demand combined with solar energy. However, this method is also considered as expensive due to its construction and operational strategy. A better insulation of the system increases the net output of the TES, and in addition by using porous material to insulate the system this also increases the outlet temperature of the TES technology. The material used for this methods can additionally to polyethylene be materials such as: Asphalt, elastomers, bitumen, resin, clay and concrete with high density[52].

All of the thermal energy storage methods can be qualified as a contributing element for the development of energy sources towards a sustainable future, in which the preferred systems are: water tank in the form of cavern TES, gravel or water TES, ATES, BTES or the storage in ducts[52]. It exists many other methods for seasonal thermal storage. A comparison of some of the methods is shown in table 8 which is a table inspired by a book[52].

Table 8: Seasonal thermal energy storage technologies

Method	Medium	Heat capacity $[kWh/m^3]$	Storage volume $[m^3]$	Depth[m]
Hot water	Water	60-80	1	5-15
Gravel-water	Gravel-water	30-50	1.3-2	5-15
BTES/Duct	Soil/rock	15-30	3-5	30-100
Aquifer	Sand/gravel-water	30-40	2-3	20-50

Thermal seasonal energy storage, heat pumps and solar energy
It exists many ways to combine heat pumps with seasonal thermal energy storage as well as solar collectors. The thermal storage will be the heat source of the heat pump, and studies[96] shows how the COP of a heat pump gets influenced by storage sizes, the earth type, and the temperature of the hot storage tanks. A larger area of solar collectors and bigger storage volumes attached to the system will result in an increased COP of the heat pump. This is especially the case after five years of operation. A project at a hospital in Belgium showed that when including an aquifer thermal energy storage combined with a heat pump for both heating and cooling of the hospital reduced the energy usage of 71% over a period of three years compared to conventional heating and cooling methods such as gas-fired boilers[96]. There is large cost savings with implementing such a system, however

it is important to denote that the operating costs of renewable energy sources is a small part of the total cost of the system. The initial cost of such a system is larger, but has decreased during the last years. The influence of the heat emission source temperature is however undetermined when looking at the connection to the design parameters of seasonal thermal storage and efficiency of the system. The system considered is shown in figure 26 which is inspired by a drawing [96].

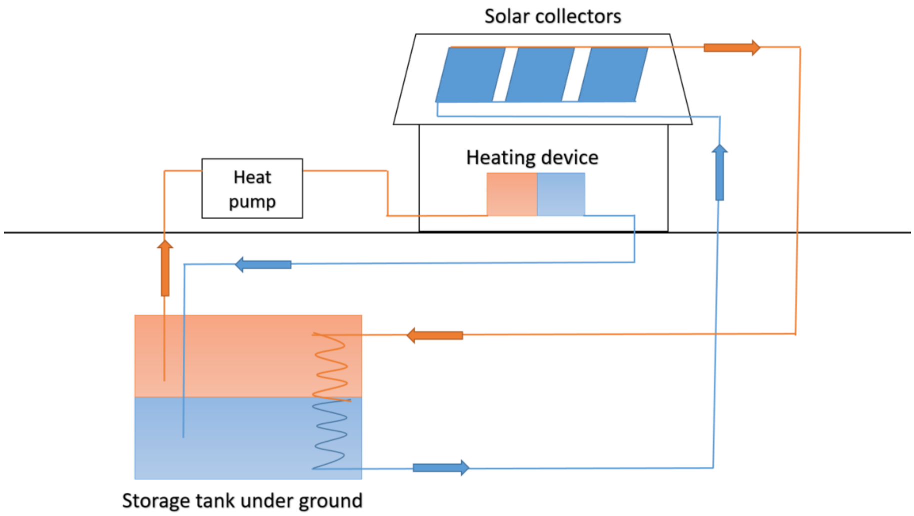

Figure 26: Solar collectors, heat pump and thermal storage under ground

2.7.4 Costs of TES

The implementation of thermal energy storage technologies can reduce the costs of the electrical consumption. When the demand of electricity is high it is possible to shift the load to reduce demand charges. For instance this is done during night time. TES contributes to energy savings for different sectors: the industry, commercial buildings and the utilization of solar energy[78]. Even though the TES technology has been investigated for many years it exists no broadly valid basis for comparing the performance of two TES technologies due to their operation in different conditions. A general basis have been investigated, and exergy analysis is proven to be a technique that have gotten increased attention for evaluating thermal performance of a TES system. The analysis is based on the second law of thermodynamics. Energy savings by using TES is done by several applications listed below[78]:

- Store waste or surplus thermal energy at some times for use at other times
- Use electricity at off-peak periods for production of thermal energy for storage. This stored thermal energy can be used during during high-demand periods
- Reduced size of equipment in new facilities and deferred purchase of additional equipment for different applications (heating, cooling, AC). Additional equipment can be used when thermal loads can not charge the system of TES.

A project in the Netherlands provides an understanding of how a TES system contribute to energy savings as well as it reduces the pollutant emissions. The system consists of a ground-water aquifer TES system. This technology was installed to provide space-conditioning for a new office owned by ANOVA Verzekering Co. The Dutch government subsidized the project by 212,000 US dollars, and it was implemented an electric heat pump for space heating and cooling. The subsidy constituted 20% of the total cost, which refers to a payback time of 6,5 years[78]. An energy budget was made for considering the energy saving potential both environmental and economical which is shown in table 9.

Table 9: Conventional energy systems and TES system

Consumption and emission	Conventional system	TES	Reduction	Reduction in percent]
Electricity [kWh]	395550	511500	84000	20%
Primary energy [m^3]	322000	179000	143000	40%
CO_2 [kg]	608000	346000	262000	43%

The energy saving potential can be presented as in table 9 and gives an indication whether or not it is profitable to invest in TES technologies.

3 Methods

This chapter presents the methods needed to evaluate the energy saving potential of a smart thermal grid at Leangen. Chapter 3.1 presents a flow sheet and approach of the master thesis, while chapter 3.2 presents an explanation of different programs being used to examine the model of a smart, low-temperature thermal grid. Chapter 3.3 contains necessary information about the building area which is being considered, and 3.4 presents two possible models of the building interface. Chapter 3.5 contains information about other waste heat sources which can be investigated in addition to waste heat from the ice skating rink. Chapter 3.6 contains information about necessary energy demand calculations, while chapter 3.7 and 3.8 contains information about how to investigate the models in Coolpack and Dymola. Chapter 3.9 is the last chapter and will consider a cost analysis of a smart low-temperature thermal grid.

3.1 Flow sheet and approach

Figure 27 represents the approach of the methods for evaluating a smart, thermal heating grid at Leangen. The blue boxes represents the main steps, while the white boxes represents relevant information for the approach.

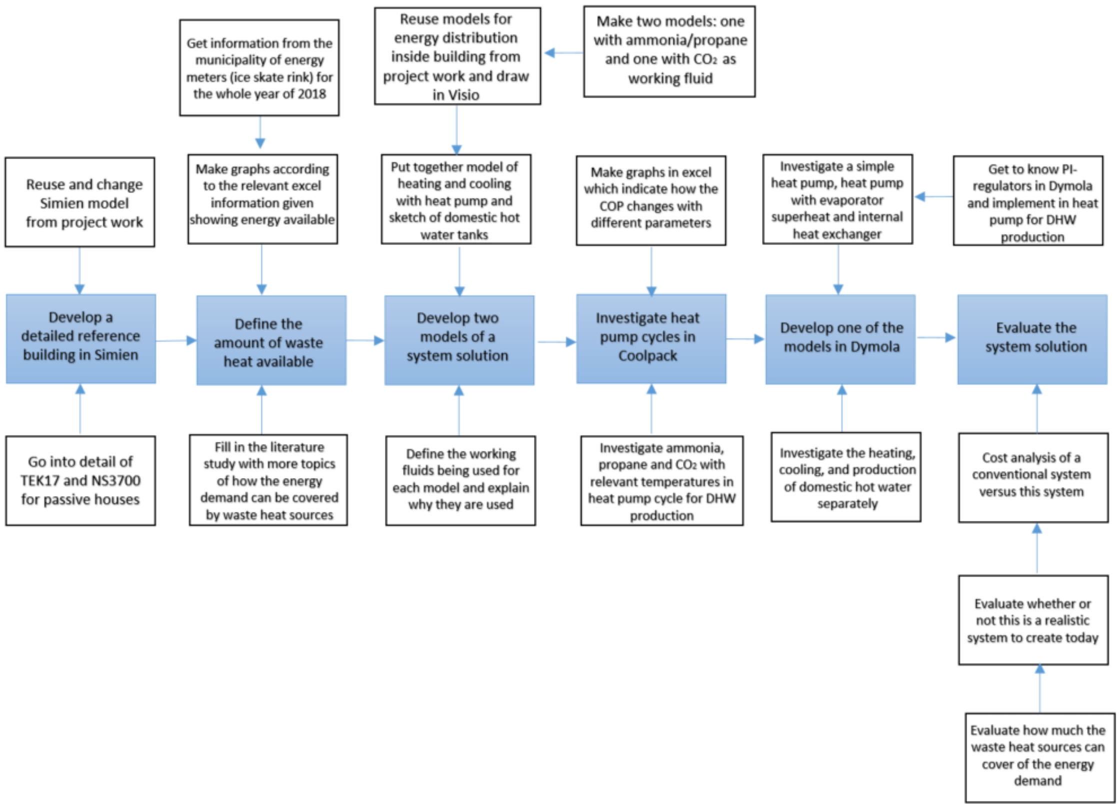

Figure 27: Flow sheet and approach

3.2 Programs

This chapter contains information about several programs which have been utilized in the master thesis. It is desirable to give the reader a brief overview, and therefore each program get a short introduction.

Simien
Simien is a simulation program which is being used to calculate the energy demand of buildings. A building can be constructed by different measures, and further on the program will simulate the total energy demand. The program is able to generate data of different demand categories such as space heating and cooling, as well as domestic hot water demand and more.

Coolpack
Coolpack is a program which can be used for the investigation of heat pump cycles with different working fluids. The program is able to calculate the COP of the heat pump, and it is possible to regulate pressure and temperature levels of the cycle. The heat pump cycle is being shown in different types of diagrams.

Dymola
Dymola is a simulation program which uses the language Modelica. The program is able to implement the TIL library which allows the use of thermal components. The program is being used to investigate heat transfer between water and space as well as heat pumps for the production of domestic hot water. The program enables the user to look at the enthalpy, pressure and temperature of different stages in the system. It is also possible to implement other components which increases the system superheat of a heat pump cycle. Due to given data of enthalpy, it is possible to calculate the COP of the heat pump, and the program allows the user to make different graphs depending on what information the user finds interesting.

DaVE
DaVE is a program which can be connected to Dymola. This program enables the construction of visual representations of different diagrams such as pressure-enthalpy, pressure-volume,temperature-entropy,temperature-enthalpy, -and pressure-temperature state charts.

Visio
Visio is a program for making drawings. This program do contain different components for engineering in the area of mechanical, electrical and process engineering. This program is desirable to use due to the components of heat exchangers, pumps, compressors and valves.

3.3 Building area of Leangen

To demonstrate how a smart, low-temperature thermal grid at Leangen should be implemented, it is of interest to gather data of the building area. The illustrations of the building area is thereby shown, and the relevant information of the building process, energy consumption and the network is presented in this chapter. The information used in this subsection is given by reports from Asplan Viak As[91] and Hagem Lund Arkitekter AS[93].

3.3.1 Building process

A racecourse at Leangen is being closed down, and the area has been sold out to Koteng Eiendom AS. The area will be used to establish mainly residential buildings in addition to offices, children's garden and commercial buildings. The area is supposed to be both sustainable and have a low energy consumption. It is estimated that the building process will start in 3 years. The building area have a long building stage which will last for 15-20 years. According to the information given by Asplan Viak[91] the development of apartment buildings will be the same in each stage, and will constitute 56268m^2 in each stage. Table 10 gives a brief overview of the different building types[91]. Figure 28 gives an indication on how the area will look like when finished.

Table 10: Amount of buildings for three building stages

Building types	Stage 1 [m^2]	Stage 2 [m^2]	Stage 3 [m^2]	Total [m^2]
Commercial buildings	1812	1812	1812	5435
Office	2500	2500	2500	7500
Children's garden	1980	0	0	1980
Apartment buildings	56 268	56 268	56 268	168 805
All buildings	62 560	60 580	60 580	183 720

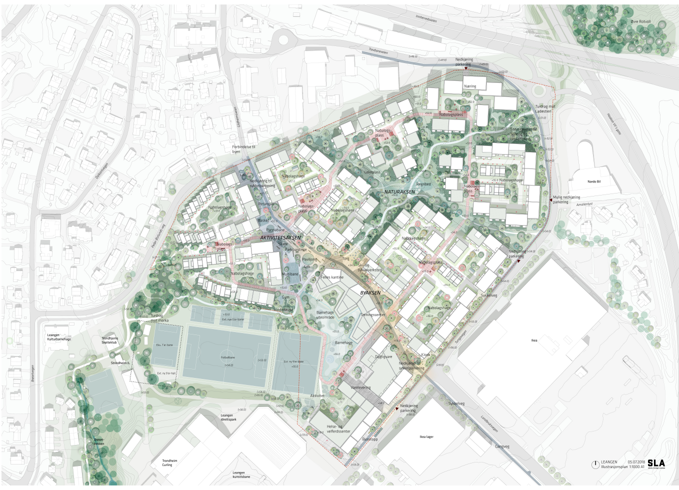

Figure 28: Illustration of building area given by Koteng Eiendom AS

3.3.2 Energy consumption of area

It is expected that it will be more frequently changes of the technical regulations of buildings. The standards which will be used in each building stage is presented in table 11.

Table 11: Standards for three different building stages

Stage 1	Stage 2	Stage 3
TEK17	Passive house standards	"Estimat 2030+"

It is assumed that there will be three stages of construction as presented in table 11. The first stage will be built according to TEK17, the second one according to the passive house level and the last one according to a standard named *Estimat 2030+*. The reference building being made in Simien will follow the regulations of TEK17 and NS 3700:2013-, the low-energy consumption and passive house standard. Both of the standards is mentioned in chapter 2.3 considering building specifications.

3.3.3 Network structure

It is of interest to implement waste heat from an ice skating rink nearby. Therefore it is made an illustration of how a central supply and return line of how water can cover the demand of the building area considering both heating and cooling which is shown in figure 29.

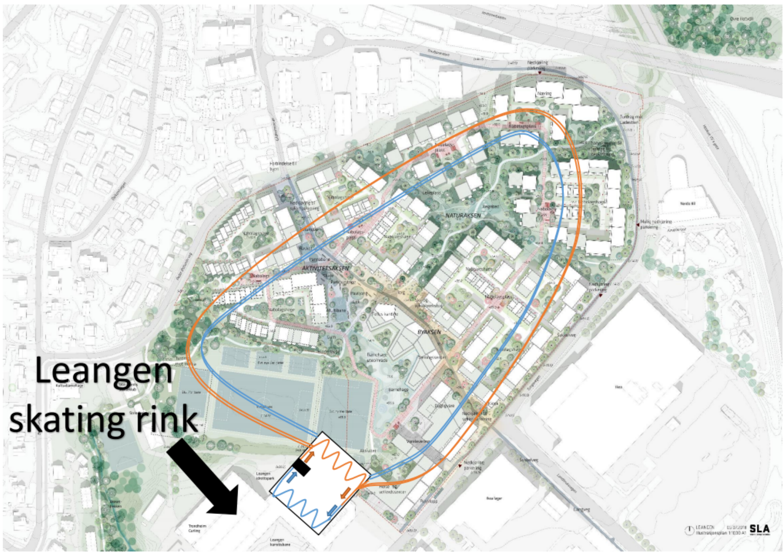

Figure 29: Illustration of low-temperature network at Leangen

Figure 29 shows a low-temperature DH system at Leangen in which waste heat is implemented. Thermal energy storage for both heat and cold can be implemented as well. Future data centers, food stores and warehouses can also be implemented in the local, thermal grid.

3.4 Models of the building interface

This chapter presents two different models of the building interface which utilizes the waste heat of the skate rink in two different ways. Model 1 utilizes ammonia or propane as working fluid in the heat pump and preheating of tap water while model 2 utilizes CO_2 and do not have preheating of tap water.

3.4.1 Construction of models

The heating and cooling supply system should be a *smart* low-temperature thermal grid. The models are designed with the ulterior motive of the system utilizing the heat from the ice skating rink in the best way possible. The models should not be too complicated due to costs. It should be a simple system with one parallel connection to the heating supply -and return lines. The space heating is done by underfloor heating, and the waste heat is reused after space heating as a heat source for the heat pump producing DHW. The models will also utilize the cold produced in the evaporator of the heat pump producing DHW for space cooling. The models differs in the way the domestic hot water is produced.

There are two models being made for further investigation which differs in the following way:

- Model 1 - Ammonia or propane as working fluid in heat pump with pre-heating of tap water from the low-temperature heating grid
- Model 2 - CO_2 as working fluid in heat pump with domestic hot water production performed entirely by the heat pump

The temperatures of the models is based on the assumption of the buildings being made of concrete. Space heating and cooling with water as heat carrier and an underfloor heating system requires a supply temperature of 35°C for space heating and for cooling the requirement is 15°C. We also assume that for each heat transfer or heat exchanger in the system there will be a temperature decrease in the range of 5-10K.

3.4.2 Model 1 - Ammonia or propane

The first model utilizes the waste heat from the ice skating rink for space heating in the building. Further on, the water is being directed to an equipment for pre-heating of tap water. This water is being directed to the inlet of the evaporator of a heat pump. The heat pump uses ammonia or propane as working fluid for production of domestic hot water, and the building will be cooled by the return water coming from the evaporator. The system sketch is presented in figure 30.

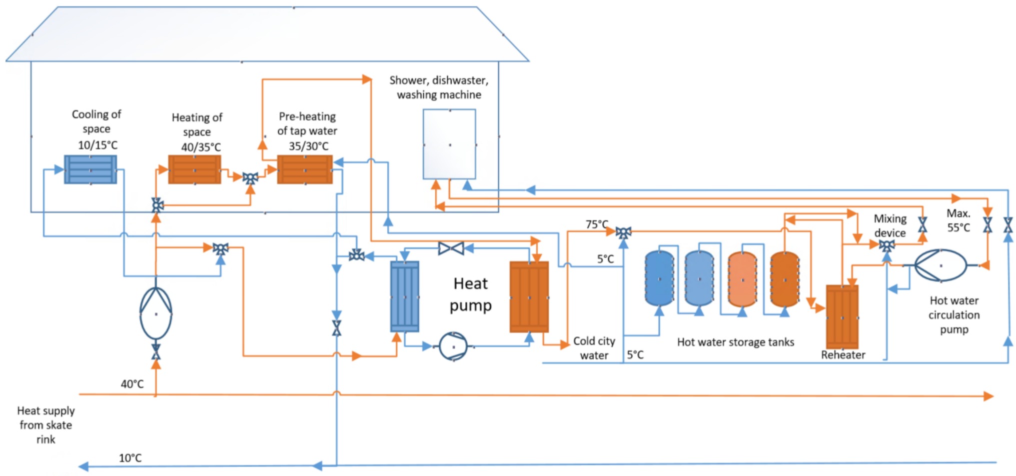

Figure 30: Model 1 with ammonia or propane

It is assumed that heat from the skate rink will heat the water to a temperature level of about 40°C. The evaporating inlet temperature is assumed to be found in the range of 30-35°C. We want to achieve a temperature level of 75°C for the domestic hot water tanks. The system for production of domestic hot water is shown in figure 31 which is a figure reused from the project thesis [1]. Hot water storage tanks can be utilized to store heat until demand occurs. When dealing with a large number of apartments, it is necessary to be prepared for the demands to come. The pre-heating should enable the domestic hot water temperature to reach the level of 75°C. he production of DHW should be in operation at least 22/24 hours a day.

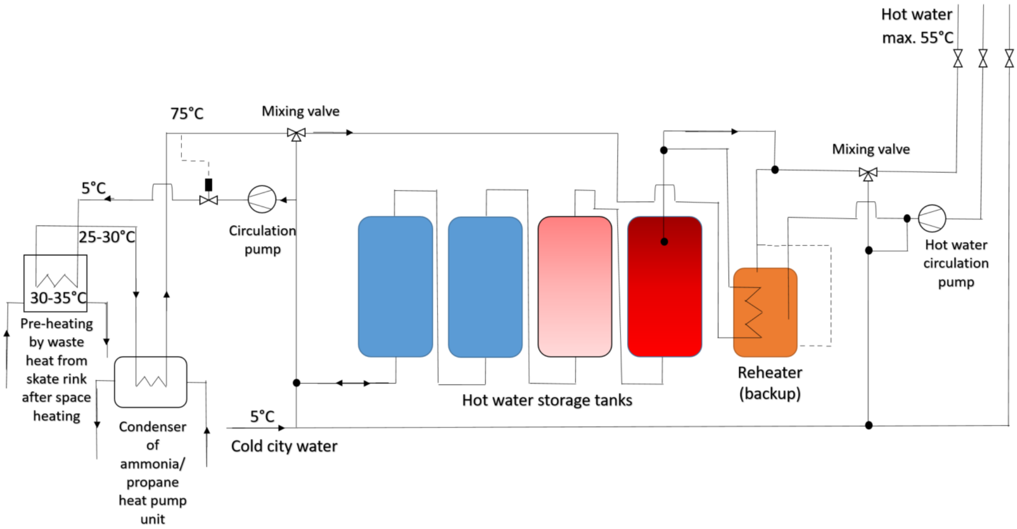

Figure 31: Model 1 - storage tanks

3.4.3 Model 2 - CO_2

The second model utilizes the waste heat from the ice skating rink in another way than model 1. The waste heat is being utilized as a input for space heating in the building. When the building requires space heating, the water flow of waste heat will be reused and goes through a bypass and into the evaporator as a heat source for the heat pump. When the building requires space cooling, and there is no source of heat available in the distribution system, the bypass valve is closed and we have a closed loop which utilizes the temperature coming from the return line of the cooling unit into the evaporator. The model 2 is shown in figure 32 made in Visio.

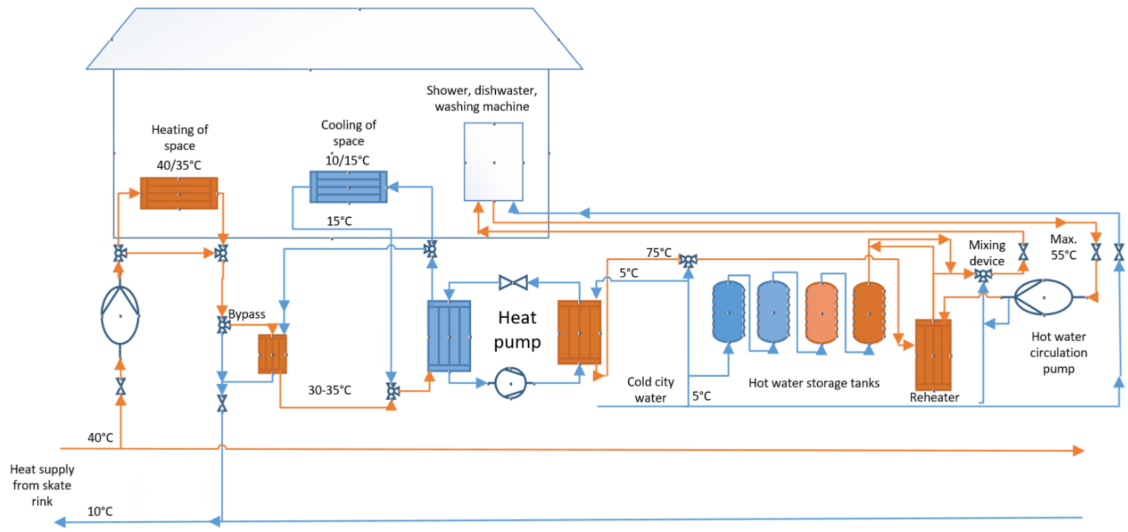

Figure 32: Model 2 with CO_2

It is assumed that heat from the ice skating rink will heat the water to a temperature level of about 40°C. Therefore, it is also assumed that the return line of the heat exchanger for space heating will have a temperature of 35-30°C. When the building requires space cooling it is assumed that the outlet temperature of the water in the evaporator can achieve a temperature of 15°C which can be used as a heat source for the heat pump. The desired temperature of DHW is 75°C which can be transferred to hot water storage tanks. Model 2 will use the thermal energy storage water tanks for the same reason as model 1; to prepare for the demand to come. The production of DHW should be in operation at least 22/24 hours a day. Model 2 do not deal with pre-heating as model 1 does and the CO_2 heat pump lifts the temperature of the cold city water all by itself. This concept is shown in figure 33 which is reused from the project thesis [1].

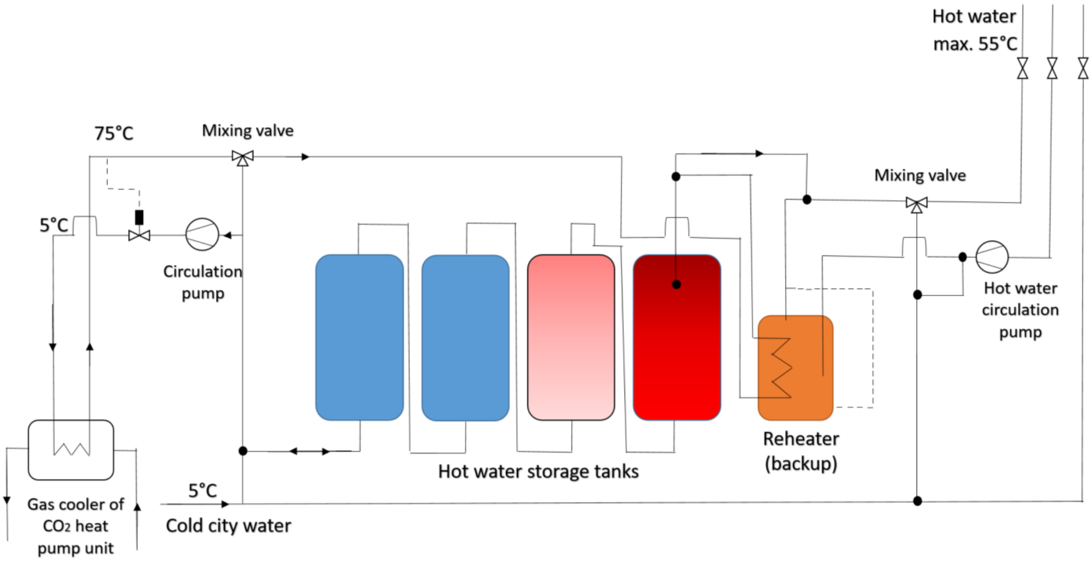

Figure 33: Model 2 - storage tanks

3.5 Alternative waste heat sources

Data from the municipality of Trondheim is given for evaluating the waste heat potential of the ice skating rink at the Leangen area. However, if this waste heat source can not cover the energy demand of the building area, other heat sources must be investigated. It is of interest to create a whole system which covers the total energy demand. The following solutions listed in this chapter can be implemented in the low-temperature local thermal gird if necessary.

3.5.1 Greywater implementation

In chapter 2.5.4 the implementation of greywater (GW) was discussed. Research confirms that 64% of the domestic hot water use is GW which can be harnessed after usage. We assume a temperature of GW in a range of 30-35°C. This is a temperature which can be fed into the heat pump for production of domestic hot water when the waste heat source do not deliver enough waste heat. There will be a constant input to this tank as the use of DHW is quite constant. This concept is shown in figure 34.

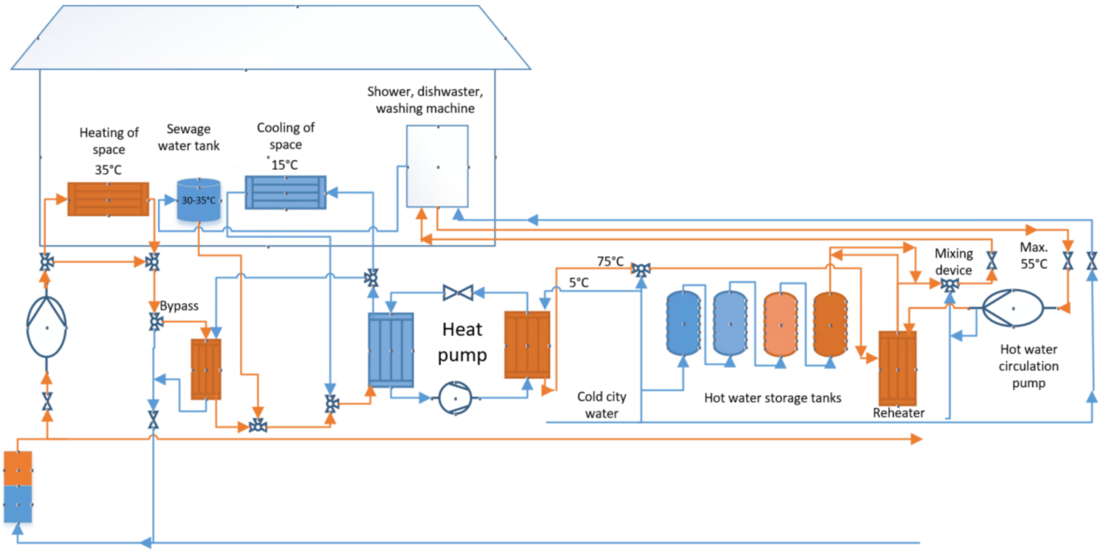

Figure 34: Model 2 with greywater implementation

Another way to implement the greywater solution is to connect all of the buildings to a common heat sink. This temperature can further on be lifted by using a heat pump and thereby achieve a temperature up to 40°C and send the waste heat into the low temperature network. This concept is shown in figure 35

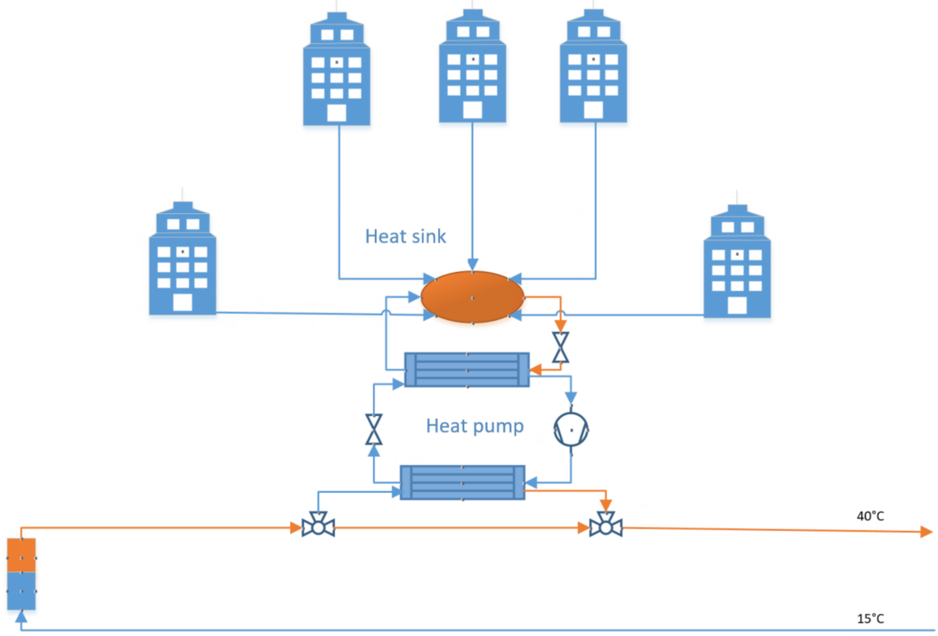

Figure 35: Centralized greywater implementation

3.5.2 Solar collectors combined with thermal storage

It is possible to store thermal energy in different ways as discussed in chapter 2.7. One possibility is to combine solar collectors and thermal storage. A brief explanation of solar collectors combined with thermal storage is listed below:

- The solar collector on the roof is used to charge the tank buried under the ground. The heat production from solar collector can be used directly if the produced temperature is high enough. Otherwise, it is used to charge the water tank.
- The hot water tank is buried deep under the ground, so the thermal loss is reduced due to not depending on out-side temperature variation. In this case, the storage is exposed to a constant ground temperature over the year.This storage is used directly for heating demand as long as the temperature is sufficient for direct usage.
- The heat pump is served by the hot water tank as a heat source when the temperature of storage is not enough to be used directly.

There may be possible to combine the this solution with waste heat from ice skating rink. The solution is shown in figure 36.

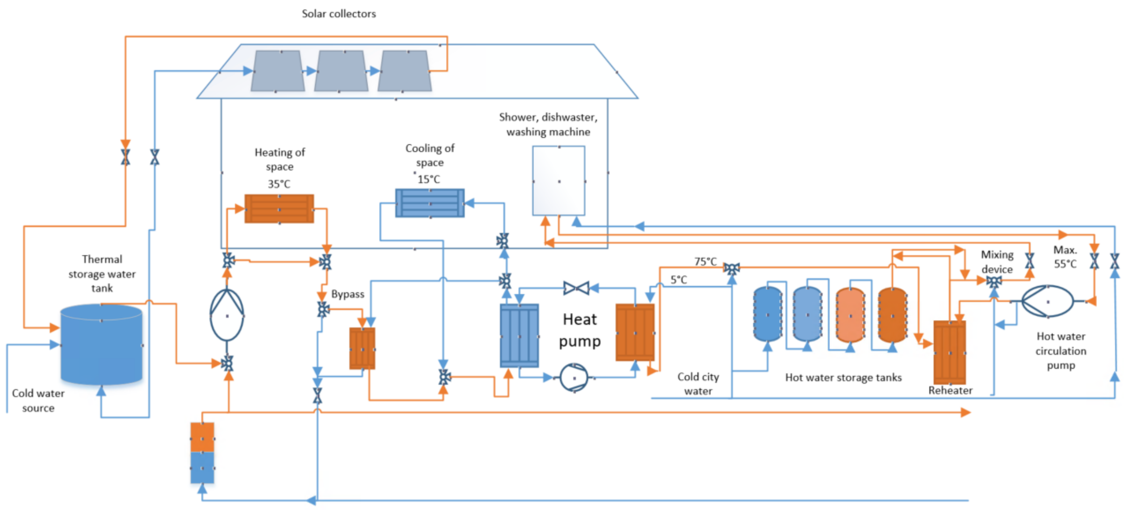

Figure 36: Model 2 with thermal storage and implementation of solar collectors

3.5.3 Future data centers and grocery stores

It is possible to use other waste heat sources than the sport center of the ice skating rink. The different solutions is shown in figure 37. The building area will be made in a period of 20 years. Therefore, other buildings which generates waste heat can be constructed in the same area during this time. It is possible to implement this waste heat and thereby cover the energy demand in the future. This can be done in the same way as the skate rink is implemented in a district heating system. This case is however difficult to investigate due to little knowledge of when and whether it will be built. Spas do use a lot of hot water which can be reused. As the area is constructed today, the high-temperature district heating system can be connected to the low-temperature district heating system as a security for energy supply for the buildings.

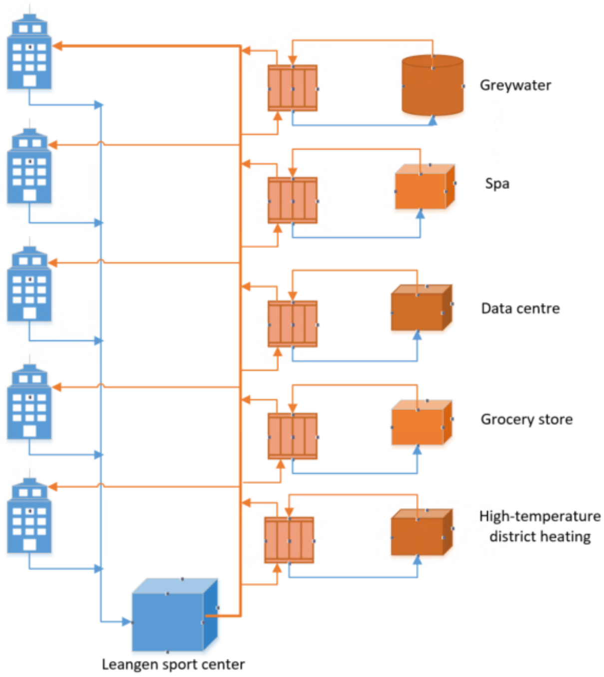

Figure 37: Low-temperature thermal grid connected to other sources

3.6 Calculations of the energy demand

This chapter contains information about a reference building made in Simien, a user profile given by Sintef Energi and relevant information about energy demand calculations for heat exchangers of space heating and cooling as well as sizing of DHW storage tanks.

3.6.1 Reference building in Simien

It is necessary to know which standard should be used when creating a reference building in Simien and evaluating the energy demand of the area. According to Asplan Viak AS and Lund Hagem Arkitekter the building area should consist of three different building stages with different specifications for construction as mentioned in chapter 3.2. The model in Simien will be developed according to the requirements of the TEK17 standard. It is desirable to make a building with a passive house standard. The building area at Leangen is going to be supplied with heat by a LTDH system which will result in water as a heat carrier in the reference building made in Simien. The average size of one apartment, amount of floors, as well as the average amount of buildings in total was given in the plan description made by Lund Hagem Arkitekter[93]. The specifications are given in table 12.

Table 12: Given data from Lund Hagem Arkitekter

Specifications	Values	Unit
Average size of one apartment	70	m^2
Average amount of floors	3-8	-
Average amount of apartments for area	1715	-

We have chosen to make a reference building consisting of following values presented in table 13 based on the specifications in table 12. The reference building made will result in 86 equal, square buildings in the area which contains 20 apartments each distributed over five floors. One apartment have a gross floor area of 70m^2.

Table 13: Size of reference building and area

Specifications	Values	Unit
Size of floor	280	m^2
Amount of apartments per floor	4	-
Amount of floors per building	5	-
Total number of buildings	85,75	-

According the specifications of TEK17, the specific energy demand for a building should be less than 95kWh/m^2. This requirement should be held when designing the building structure.

3.6.2 User profile of passive house building

It is of interest to investigate data from a user profile of a passive house building in Trondheim and compare the energy demand to the Simien file. The user profile was given by Sintef Energi and shows specific space heating and DHW demand per hour given in W/m^2. The energy demand per month is shown in table 14.

Table 14: Energy demand for passive house user profile

Month	Qsh [kWh]	Qdhw [kWh]	Qtot
January	844983	389268	1234251
February	845599	351150	1196749
March	834638	387764	1222402
April	500525	376679	877204
May	192401	389040	581440
June	67351	375513	442864
July	26816	389340	416156
August	40541	388844	429385
September	114751	375840	490591
October	370562	389023	759585
November	600227	375676	975903
December	643725	382320	1026045

The specific energy demand is multiplied with the size of the reference building. The building have a specific space heating demand of 26,2 kWh/m^2 and a domestic hot water demand of 30,6kWh/m^2. The space cooling demand is not measured. This is data which is important to investigate when evaluating the reference building simulated in Simien. By utilizing the MAX function in excel we find the biggest specific capacity for both space heating and hot tap water demand which is 15,332 W/m^2 and 4,222W/m^2.

3.6.3 Heat exchangers for space heating and cooling

There are some parameters which should be investigated when using heat exchangers for space heating and cooling in Dymola. We need to calculate the overall heat transfer coefficient (U) and mass flows of water.

The temperatures needed for the heating and cooling is about 35°C and 15°C. We assume that the supply temperature from the building itself is respectively 25°C when heating up the building and 35°C when we want to cool it down. The values of both heating and cooling are listed in table 15. The temperature in the building is set to 25°C. This is not a constant value throughout the year. The ambient outdoor temperature is influencing the temperature of the water in the building and accordingly the heating and cooling demand varies to obtain a certain temperature level. However, we make a set point temperature so we can investigate the heat exchangers.

Table 15: Temperature values for heating and space cooling

Parameter	Temperatures for space heating	Temperatures for space cooling
T_{h1}[°C]	40	25
T_{h2}[°C]	35	15
T_{c1}[°C]	25	10
T_{c2}[°C]	35	15
C_{pwater} [J/kgK]	4200	4200

It is possible to calculate the UA value and mass flow values for the heat exchanger with water on both sides by using equation 2 which was presented in chapter 2.2.1. For this purpose, the capacity is needed, and the capacity for space heating for the reference building can be calculated by using the specific capacity value given by the user profile as 15,332W/m^2 which is equal to 21,464 kW of space heating demand per building. The value of 15,332 W/m^2 is the highest specific capacity value for space heating. The theoretic value mentioned in chapter 2.3.1 is 10W/m^2 which is way less than reality. Therefore we use the realistic value to be sure of enough heat supply. The specific capacity of space cooling is hard to find for low-energy consumption or passive house buildings. Therefore, we need to find a method for estimating a value. The cooling capacity is estimated by looking at the relationship between heating and cooling demand generated by Simien later on.

$$\frac{Q_{cooling}[kWh/m^2]}{Q_{heating}[kWh/m^2]} * 15,332W/m^2 = Q_{cooling}[W/m^2]$$

3.6.4 Sizing of storage tanks

It will be implemented storage tanks connected to the heat pump producing domestic hot water. The storage tanks will cover the demand during the peak hours, and will prevent unnecessary on -and off operation of the compressor. It is difficult to calculate the mass flows and UA values of CO_2 due to the fact that specific heat capacity of CO_2 changes rapidly with temperature changes. As the user profile suggests, 4,222 W/m^2 is the highest specific capacity value for domestic hot water production. In this case the theoretic value of 5,1 W/m^2 is higher and is therefore used in further calculations to be sure of enough domestic hot water being produced. The storage tanks should have a capacity of about 7,140 kW which can be stored as high temperature DHW.

When designing the tanks, the demand during a day must be evaluated. According to Geir Eggen [99] a possible method for calculating the size of the tanks when producing water all hours a day is as follows:

- Measure the amount of hot water demand for a period
- Look at the day with the highest amount of accumulated flow
- Compare the demand curve with heat production of the CO_2 heat pump
- Evaluate the biggest vertical distance between the consumption and production curve to find the volume given in m^3 for the size of the tank

An example of this method and its result is shown in figure 38, in which the distance between the slope from start point to end point indicates the production of domestic hot water which will be constant during the day. The user profile will be investigated on the day with the highest demand to be sure of the tanks being sized big enough.

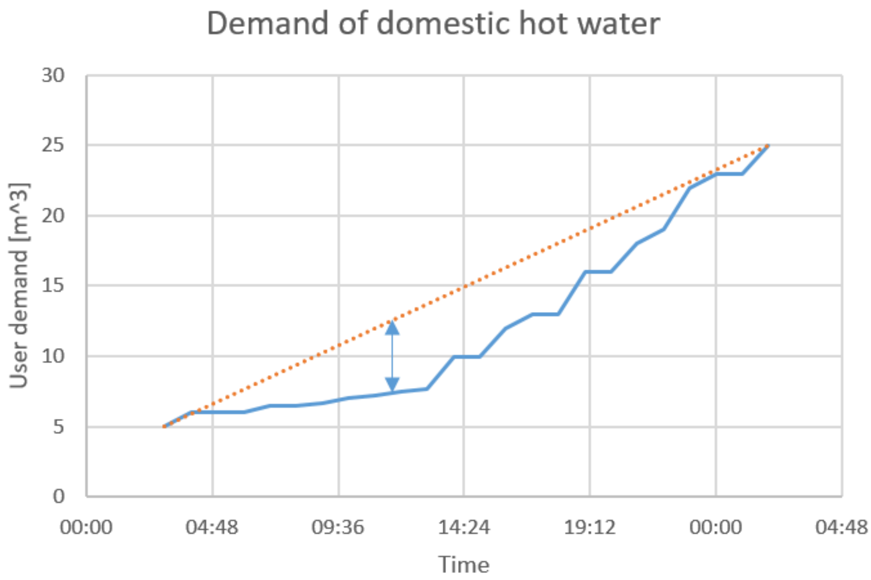

Figure 38: Method for sizing water tanks

We want to make a graph which shows the user demand given in m^3 on the y-axis and hours on the x-axis. It is also desirable to implement a trend-line on this graph. In this way it is possible to read how big the tanks should be to cover the demand of DHW. It is desirable to investigate the user profile at the day with highest DHW demand as well as theoretic values of DHW demand and further on design a storage tank. Data from experiences of sizing a DHW tank will also be of interest. Information from personal communication with Harald Walnum from SINTEF Community was given regarding sizing of water storage tanks in which an apartment of $70m^2$ will have a demand of 100L per day, but the demand varies from 80-120L. To maintain safety one should design the tanks with a demand of 150L. This was a requirement regarding a temperature difference of 50K in the tanks when water is heated from 10-60°C.

3.7 Heat pump cycles in Coolpack

For the comparison of the two models using different refrigerants for the heat pump process, heat pump cycles must be made. Coolpack is being used for the construction of the cycles and for decision of which working fluid to use and thereby which model to use. We investigate ammonia, propane and CO_2 and the temperature levels will be equal for all refrigerants to investigate which one can achieve the highest COP value.

3.7.1 General data for all refrigerants

It is necessary to have a capacity of the storage tanks of 7,140 kW to cover the hot water demand when having a specific hot tap water demand of 5,10W/m^2. We assume a capacity of 7,140 kW for the condenser/gas cooler. It is desirable to achieve a temperature level of 75°C for the domestic hot water storage tanks. The loss when transferring heat have a range of 5-10K, and we chose a discharge temperature of 90° to be sure of enough heat being transferred. The ammonia, propane and CO_2 heat pump cycle will have the following fixed parameters listed in table 16.

Table 16: Data for ammonia, propane and CO_2 heat pump cycle

Parameter	Value
Discharge temperature [°C]	90
Superheat [K]	5
Isentropic efficiency	0,75
Q_c/Q_{gc} [kW]	7,14

3.7.2 Model 1 - Ammonia or propane

Ammonia is chosen as a working fluid due to its high critical temperature and pressure, as well as its high values of specific enthalpy of evaporating. It is also considered as a natural working fluid with a ODP and GWP of zero. However, this working fluid will require some safety measures due to the working fluid being toxic. Propane is also a natural refrigerant with zero ODP and GWP. However, a hydrocarbon like propane is flammable, and there must be implemented safety measures.

Ammonia heat pump cycle
We want to investigate how the evaporating temperature affect the system and its COP. The different attempts for evaporating temperatures is shown in table 17.

Table 17: Ammonia heat pump cycle with different evaporating temperatures

Parameter	Attempt 1	Attempt 2	Attempt 3	Attempt 4
Evaporating temperature [°C]	5	10	15	20
Q_e [kW]	3,194	3,439	3,684	3,931
m [kg/s]	0,033	0,033	0,034	0,034
V [m^3/h]	10,096	8,9391	7,938	7,067
W [kW]	3,946	3,701	3,456	3,209

Propane heat pump
We want to investigate how the evaporating temperature affect the system and its COP. The different attempts for evaporating temperatures is shown in table 18.

3.7.3 Model 2 - CO_2

CO_2 is chosen for further investigation due to its low critical temperature compared to other working fluids. Due to its low critical temperature, CO2 can be operated in a transcritical stage in which the condenser is no longer a condenser but a gas cooler. The heat delivery temperatures are no longer limited by the critical temperature. CO2 has a GWP and a ODP of zero, and is considered as a natural refrigerant.

Table 18: Propane heat pump cycle with different evaporating temperatures

Parameter	Attempt 1	Attempt 2	Attempt 3	Attempt 4
Evaporating temperature [°C]	5	10	15	20
Q_E [kW]	3,194	3,439	3,684	3,931
m [kg/s]	0,033	0,033	0,034	0,034
V [m^3/h]	10,096	8,939	7,938	7,067
W [kW]	3,946	3,701	3,456	3,209

The system needs equipment which can handle high pressures. The disadvantage of CO_2 as a working fluid is that big concentrations of CO2 can however choke a human being. Different values is being performed for the construction of cycles.

Table 19: CO_2 heat pump cycle with different pressures

Parameter	Attempt 1	Attempt 2	Attempt 3	Attempt 4
Pressure [bar]	75	90	105	120
Evaporating temperature [°C]	10	10	10	10
Outlet temperature (T_4) [°C]	30	30	30	30

The CO_2 cycle uses the parameters as listed in table 16. We want to investigate how different pressures in the system affect the COP. We also want to know how the evaporating temperature and the outlet temperature (T_4) affect the system. The different attempts for different pressures is shown in table 19. We still want a capacity of 7,14 kW for the gas cooler. The attempts for different evaporating temperatures is shown in table 20.

Table 20: CO_2 heat pump cycle with different evaporating temperatures

Parameter	Attempt 1	Attempt 2	Attempt 3	Attempt 4
Evaporating temperature [°C]	0	5	10	15
Pressure [bar]	95	95	95	95
Outlet temperature (T_4) [°C]	30	30	30	30

The attempts for different outlet temperatures (T_4) is shown in table 21 and the attempts considers temperatures in the range of 15-45°C.

Table 21: CO_2 heat pump cycle with different outlet temperatures (T_4)

Parameter	Attempt 1	Attempt 2	Attempt 3	Attempt 4
Outlet temperature (T_4) [°C]	15	25	35	45
Pressure [bar]	95	95	95	95
Evaporating temperature [°C]	10	10	10	10

3.8 Developing a model in Dymola

This chapter will contain information about how Dymola is being used to investigate the heat pump cycle and heat exchangers. One of the models will be constructed in Dymola for further technical considerations. The choice of model will depend on the system solution and the characteristics of the refrigerant used in the heat pump. The heat pump will be investigated in connection to the space heating circuit and the space cooling circuit. Following components and heat pump solutions will be investigated in Dymola:

1. Simple heat pump for DHW production with PI-regulator
2. Heat pump with superheat and internal heat exchanger
3. Heat exchanger for space heating
4. Heat exchanger for space cooling

The heat pump which will be simulated in Dymola for our reference building is assumed to have a gas cooler capacity of 7140W due to theoretic values of DHW demand of 5,1W/m^2. This results in a mass flow of 0,024kg/s on the water side of the gas cooler according to equation 1. Equation 1 defines the demand which must be stored in the storage tanks.

3.9 Cost analysis of system

It is of interest to compare the costs of a conventional DH system and a smart, local thermal grid. This chapter will give a brief overview of how to evaluate the cost saving potential with our new system solution.

3.9.1 Electricity versus district heating and cooling

According to Statkraft Varme[90] the energy price is the price of the actual amount of energy used, with a price consisting of the monthly average spot price in the market area. In the years of 2007-2017 the annual value of the average spot price in EUR/MWh for the Nordic/Baltic system varied between 22-53 EUR/MWh according to a report by Nord Pool[97]. The values is listed in table 22.

Table 22: Nordic spot prices from 2007-2017

Year	Nordic spot price	Unit
2007	26	[EUR/MWh]
2008	45	[EUR/MWh]
2009	35	[EUR/MWh]
2010	53	[EUR/MWh]
2011	48	[EUR/MWh]
2012	32	[EUR/MWh]
2013	38	[EUR/MWh]
2014	27	[EUR/MWh]
2015	22	[EUR/MWh]
2016	27,5	[EUR/MWh]
2017	29	[EUR/MWh]

The lowest and highest spot price will be used to make a cost saving potential when the spot prices is low and high. The lowest and highest spot prices is 22 and 53 EUR/MWh. For further evaluation the value of 9,57 NOK/EUR for the date of 21.04.19 is used.

3.9.2 Production of domestic hot water

By implementing a heat pump for production of tap water, it is possible to reduce the costs by following equations listed below. These equations is being used further when evaluating the cost saving potential of implementing heat pumps producing DHW.

$$EL_{HP}[kWh] = \frac{EL_{boiler}[kWh]}{COP_{HP}}$$

$$Reduction_{energy}[kWh] = EL_{boiler}[kWh] - EL_{HP}[kWh]$$

$$Reduction_{costs}[NOK] = Reduction_{energy}[kWh] * spotprice[NOK/kWh]$$

4 Results

This chapter deals with results which will reflect the energy saving potential of implementing a smart, low-temperature thermal grid at Leangen. The chapter includes information about the energy demand of the reference building made in Simien and an investigation of potential waste heat from ice skating rink and greywater. In addition, heat pump cycles for ammonia, propane and CO_2 as working fluids is investigated considering evaporating temperatures, pressure and outlet temperature out of gas cooler. A CO_2 heat pump is further on investigated in Dymola with the given temperatures in our system to gain knowledge of the real process. In addition, the heat exchangers for space heating and cooling is investigated according to the capacity needed for the reference building. A demonstration of high temperatures in storage tanks for DHW and how the volume of the tank is affected by the temperature difference is presented, as well as sizing of the tanks. The chapter will end with an analysis of the whole system and the energy saving potential together with a cost analysis which will describe the potential by shifting from a conventional energy system based on electricity or high-temperature district heating to a local, smart low-temperature thermal grid.

4.1 Energy -and power demand for reference building

The thermal load of a building connected to the energy distribution system is essential when analyzing the efficiency of the low-temperature thermal grid. It will appear different thermal loads according to different user behavior. Some customers do have sporadically power peaks, while others do have a more uniform power demand. This chapter presents the energy -and power demand of space heating and cooling, as well as production of DHW for a residential building connected to the smart, thermal grid at Leangen.

4.1.1 Heating, cooling and hot tap water demand

The monthly energy demand of the reference building is shown in figure 39. This is equal to a yearly net energy demand of 129080kWh and a specific energy demand of 92,2kWh/m^2. The area consists of 86 buildings which constitutes a yearly energy demand of 11100880kWh for the building area. The different colors symbolize the energy demand in following areas:

- Red - space heating
- Dark blue - domestic hot water
- Yellow - fans
- Medium blue - pumps
- Green - lightning
- Grey - technical equipment
- light blue - space cooling

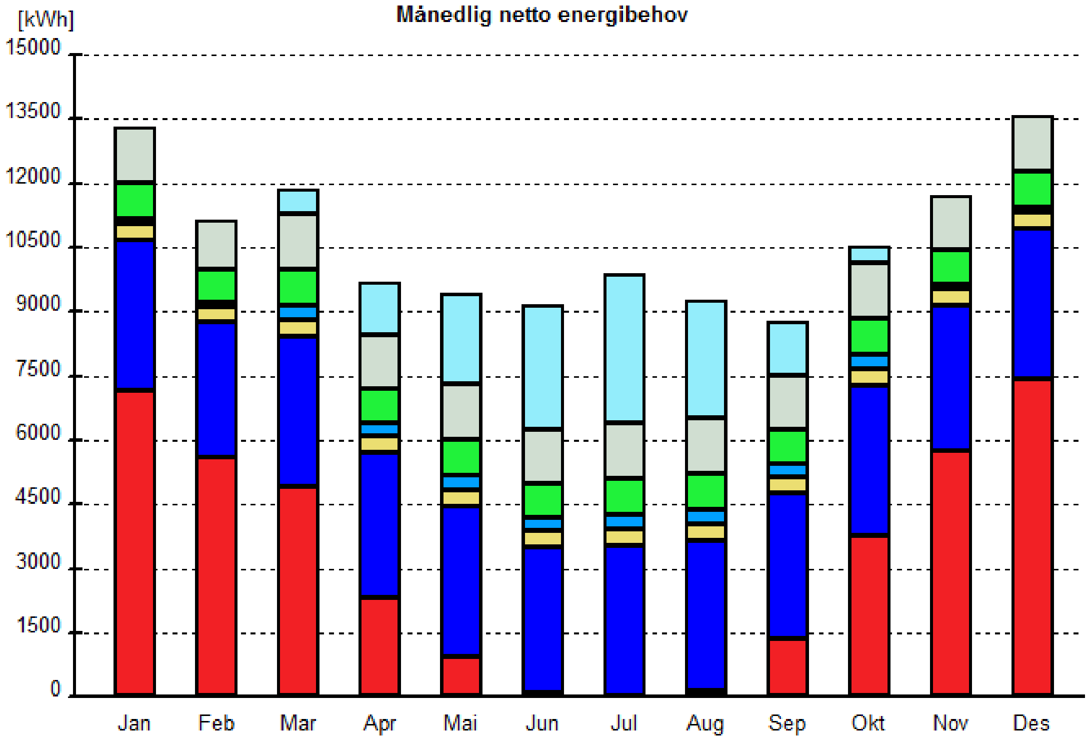

Figure 39: Monthly net energy demand for reference building

The monthly energy demand for the area is reflected in table 23 and given as kWh. In total the space heating demand for one building will constitute 39313kWh and 28,1kWh/m^2, the space cooling demand will constitute 14791kWh and 10,6kWh/m^2, and the production of DHW will be 41711kWh and 29,8kWh/m^2. Figure 40 reflects the energy demand presented in a pie chart.

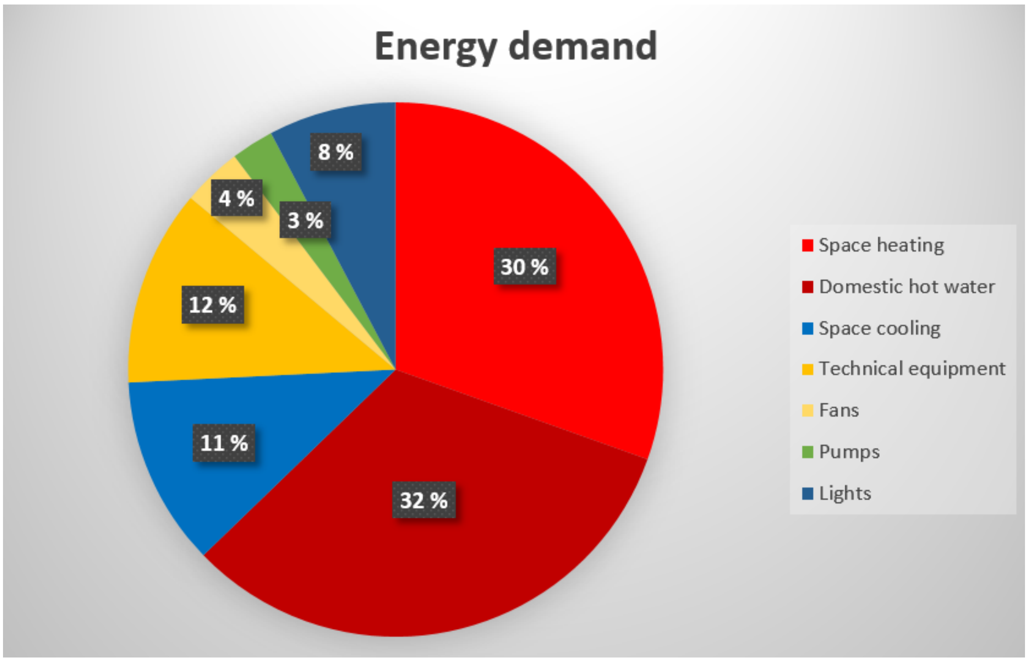

Figure 40: Energy demand of reference building

Table 23: Energy demand for one building

Month	space heating [kWh]	space cooling [kWh]	DHW demand [kWh]
January	7109	0	3401
February	5609	0	3250
March	4934	583	3401
April	2309	1333	3401
May	809	2108	3701
June	0	2934	3551
July	0	3384	3551
August	0	2783	3701
September	1409	1283	3401
October	3809	383	3401
November	5759	0	3401
December	7566	0	3551

Figure 41 reflects the use of different energy sources. Electricity constitutes 33%, electricity for heat pump constitutes 13% and the district heating will constitute 54% of the energy delivered to the building. Each building will be of the same size and have the same square geometry. Therefore the energy demand will be the same. The use of DHW and the space heating for apartment buildings have a longer uptime than for office buildings. For office buildings the uptime is often between 08:00-18:00 weekly, and off or active at a low level when people are at home. The uptime for space heating is long for this reference buildings, and there is only a few hours at the night in which the heating is switched off. The space cooling is activated a few hours in the middle of the day from March to October.

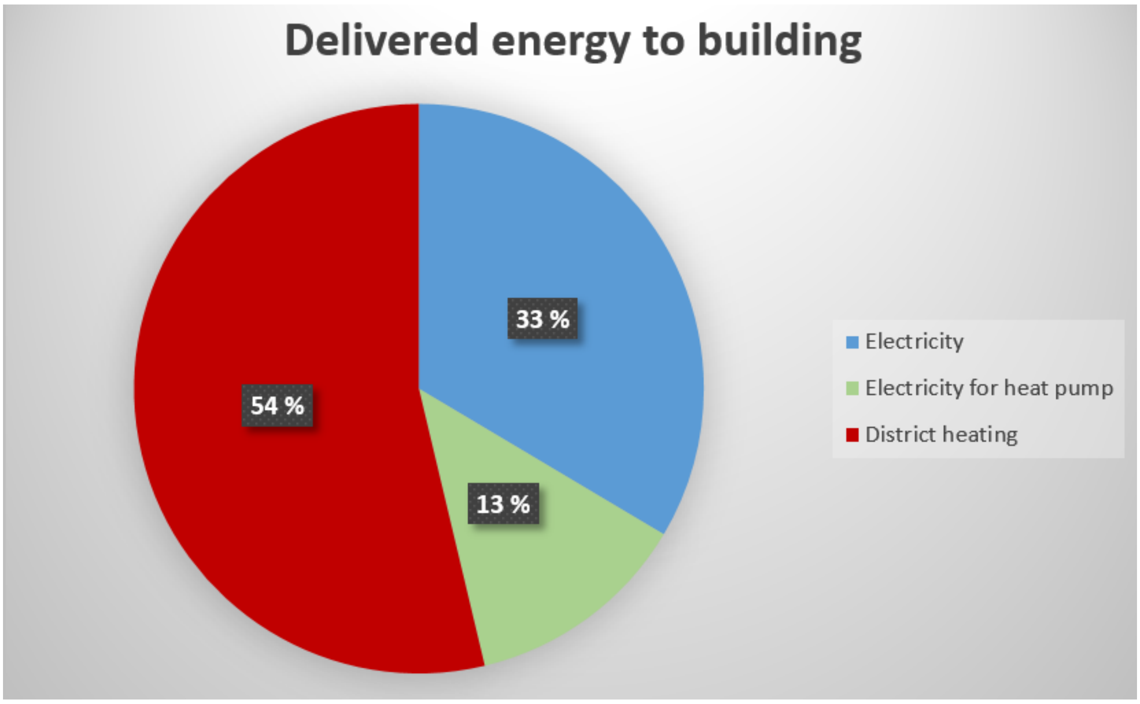

Figure 41: Delivered energy to reference building

4.1.2 Duration curves and thermal loads

The use of power for one building is presented in a power duration curve shown in figure 42. This curve is based on variations of the power demand for one building during a year and is sorted by high to low power demand. The red line represents the space heating demand while the green line represents the space cooling demand. The area under the curve defines the energy demand, and the power demand is put in order from high demand to low demand. The duration curve will be different for different years due to variations of the weather conditions.

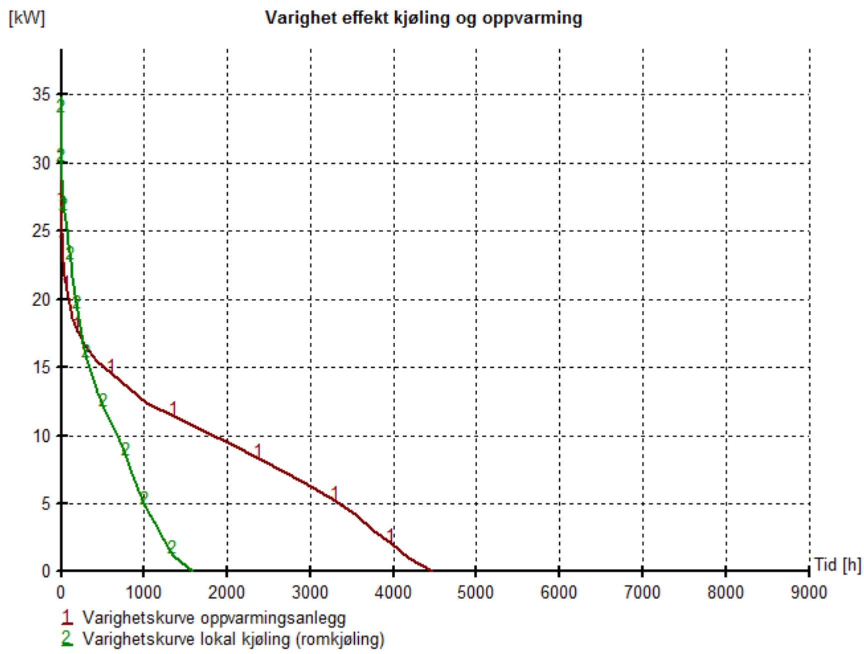

Figure 42: Duration curve for reference building

Conventionally, such a curve should have been more pulled to the right, especially considering the heating demand. However, the reference building is a very energy efficient building, and this duration curve only shows the heating and cooling of the building. Domestic hot water is not being considered. In many homes, heating of domestic hot water is done by using boilers which will contribute to a larger power demand.

4.1.3 Energy balance and heat losses

Figure 43 shows the components which contributes to the energy consumption of both space heating and cooling. On the upper side we recognize that the sun (yellow), the heat recovery of ventilation (light pink) and the space heating (red) equals the generated energy. On the downside we recognize that the ventilation (dark red), transmission of building structure (turquoise) and the space cooling (purple) equals the lost energy. By looking at this energy balance it is possible to decide the optimal energy demand of the building.

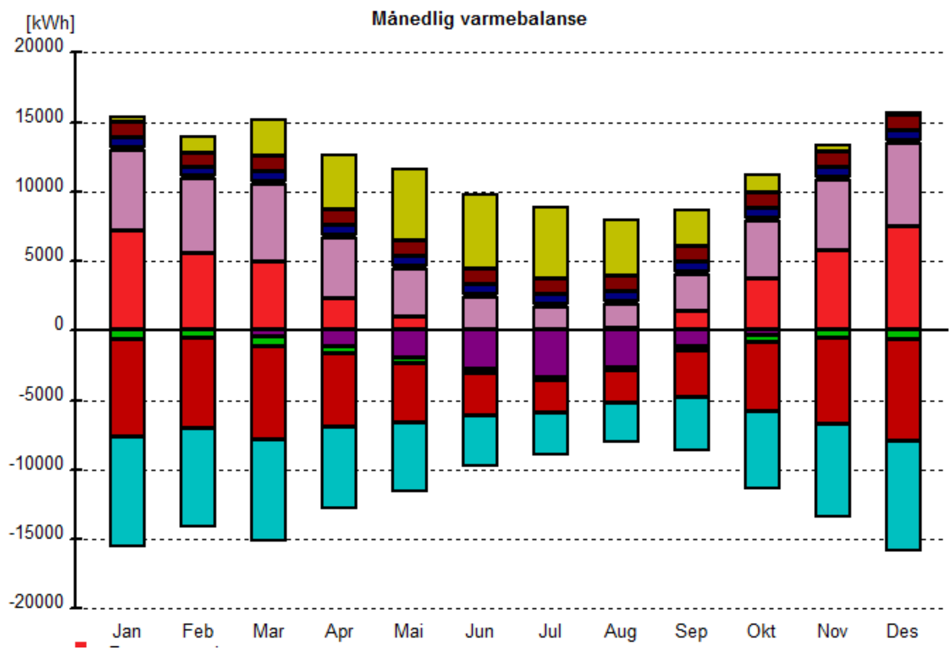

Figure 43: Energy balance for reference building

The heat losses of the building is due to several factors. Heat loss occurs through walls, roof, floor, windows and doors. Other sources for heat losses is due to the ventilation of air, infiltration and thermal bridges. The heat losses is distributed as shown in figure 44.

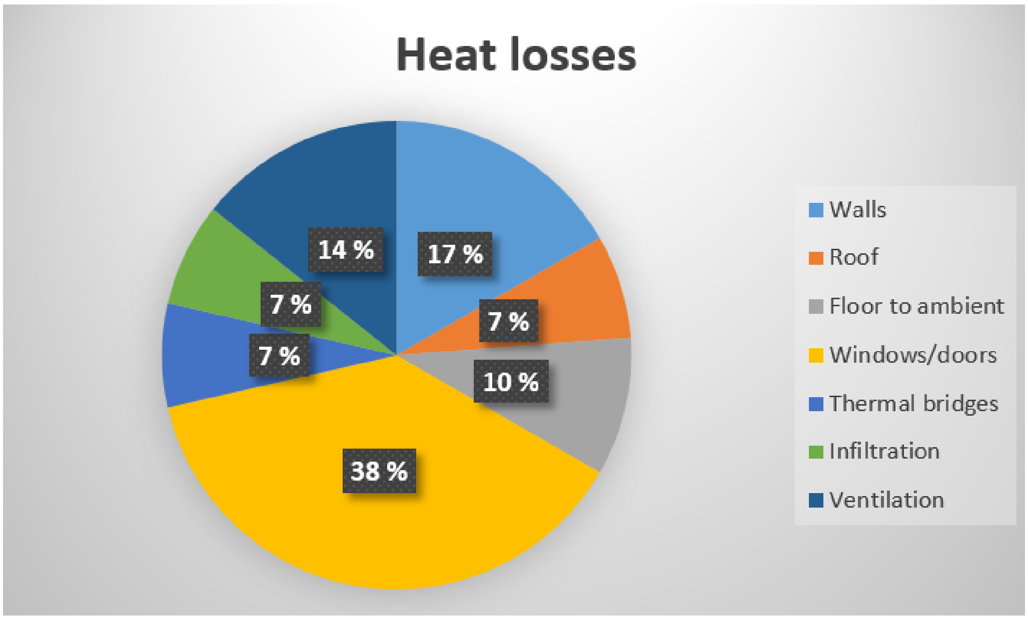

Figure 44: Energy loss for reference building

The user profile of a passive house building given by Sintef Energi indicates a yearly specific demand of 26,2 kWh/m^2 for space heating, and 30,6 kWh/m^2 of DHW demand. This indicates that our reference building in Simien have a bigger energy use which is 28,1kWh/m^2 for space heating and a lower energy demand for DHW production of 29,7kWh/m^2. Further on, we will investigate how waste heat can cover our simulated reference building in Simien.

4.2 Potential of waste heat sources

This chapter will contain information about how the waste heat coming from the ice skating rink and greywater can cover the energy demand of the reference building made in Simien. This will give a brief view of the potential of the waste heat at the Leangen area and the possible energy saving potential by implementing waste heat in a building area consisting of low-energy consumption buildings.

4.2.1 Waste heat potential of ice skating rink

In table 24 the available waste heat from the refrigeration system at the ice skating rink nearby is given in kWh. A system sketch of the refrigeration system is shown in appendix G. It is of interest to investigate the energy potential of the waste heat from both condensers and further on demonstrate how the waste heat can cover the energy demand for the reference building.

Table 24: Waste heat from ice skating rink

Month	Youth hall [kWh]	Arena [kWh]	Total waste heat [kWh]
January	152375	312182	464557
February	138011	324388	462398
March	156585	229857	386442
April	156246	55313	211560
May	154010	0	154010
June	140228	8	140236
July	0	51	51
August	135448	125098	260546
September	158921	108428	267350
October	175709	74366	250074
November	173866	629286	803152
December	161524	590218	751742

Figure 45 illustrates the total amount of waste heat, and it is possible to recognize that there is a lot of waste heat available as shown by the energy meters placed by condensers of the ammonia refrigeration system.

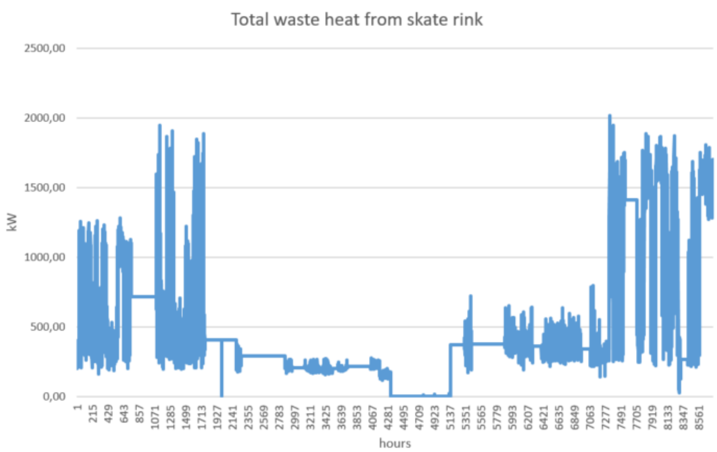

Figure 45: Total waste heat from both condensers of ice skating rink

The total waste heat of the year of 2018 was 4152292 kWh. Separate graphs of the condensers is shown in figure 122 and 123 which can be found in appendix A. The amount of waste heat from the ice skating rink together with the information of energy demand for the reference building is further illustrated. It is of interest to see how much of the space heating demand the waste heat can cover. We do also want to investigate how much the waste heat can cover the energy demand of both space heating and DHW production. We assume 86 buildings at the area of Leangen.

Space heating
It is of interest to consider how much of space heating demand the waste heat from the ice skating rink is able to cover of the area. Table 25 shows the coverage of the energy demand for space heating.

Table 25: Waste heat from ice skating rink and coverage of space heating demand

Month	Waste heat [kWh]	Space heating [kWh]	Percentage of buildings covered [%]
January	464557	611374	76,2
February	462398	482374	95,2
March	386442	424324	91,1
April	211560	198574	106,4
May	154010	69574	221,4
June	140236	0	-
July	51	0	-
August	260546	0	-
September	267350	121174	220,6
October	250074	327574	76,3
November	803152	495274	162,2
December	751742	650676	115,5

As figure 46 indicates, the energy demand for space heating is covered for almost all months for the entire building area.

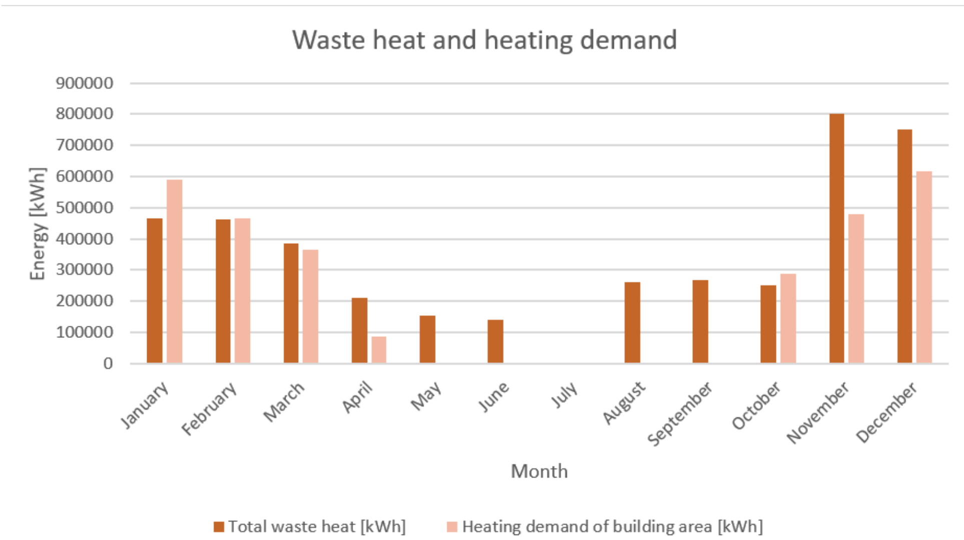

Figure 46: Waste heat potential and space heating demand

The exception is January, February, March, and October in which the waste heat covers 76,2%, 95,7%, 91,1 and 76,3%. Nevertheless the percentage of coverage is high, and it exists a big potential of using waste heat as a heat source. In the months from May to September there is no or little demand for space heating. The yearly average coverage of the building area is 122,8%. From May to September it will be no need for space heating due to warmer temperatures and good isolation in new buildings.

Production of domestic hot water
Table 26 shows the coverage of the energy demand for production of DHW. In this case, we consider the waste heat as an energy source for the heat pump producing DHW. We assume a COP of 4 of the heat pump, and due to the high COP value the heat pump will have a low electricity demand. The heat needed as a heat source is the value of demand per month minus the demand per month divided by the COP value of the heat pump. The energy needed as a heat source for the heat pump is denoted as X.

$$X = 3587146 - 3587146/COP$$

Table 26: Waste heat from ice skating rink and coverage of DHW demand

Month	Waste heat [kWh]	Heat demand HP [kWh]	Percentage of buildings covered [%]
January	464557	219365	212,2
February	462398	209625	220,3
March	386442	219365	176,1
April	211560	219365	96,4
May	154010	238715	64,5
June	140236	229040	61,1
July	51	229040	0,0
August	260546	238715	109,3
September	267350	219365	121,9
October	250074	219365	114,0
November	803152	219365	366,1
December	751742	229040	328,2

Table 26 illustrates the potential of implementing waste heat from the ice skating rink to cover the DHW demand considering heat source for the heat pump. This is reflected in figure 47. It exists a huge potential of using waste heat. The yearly average percentage of covered building DHW demand is 154,3%.

Space heating and domestic hot water
Table 27 shows the coverage of the energy demand for both space heating and the heat needed for the heat pump. The two different demands is denoted as *Energy demand*.

Table 27 illustrates the potential of implementing waste heat from the ice skating rink. As figure 48 indicates, the waste heat can not cover the energy demand of both space heating and production of domestic hot water for the months of August and November with a coverage of 109,3% and 112,4. It is still a huge potential of using waste heat and the waste heat covers over 50% for all months with the exception being June and October when it only covers 0,0% and 45,7%.

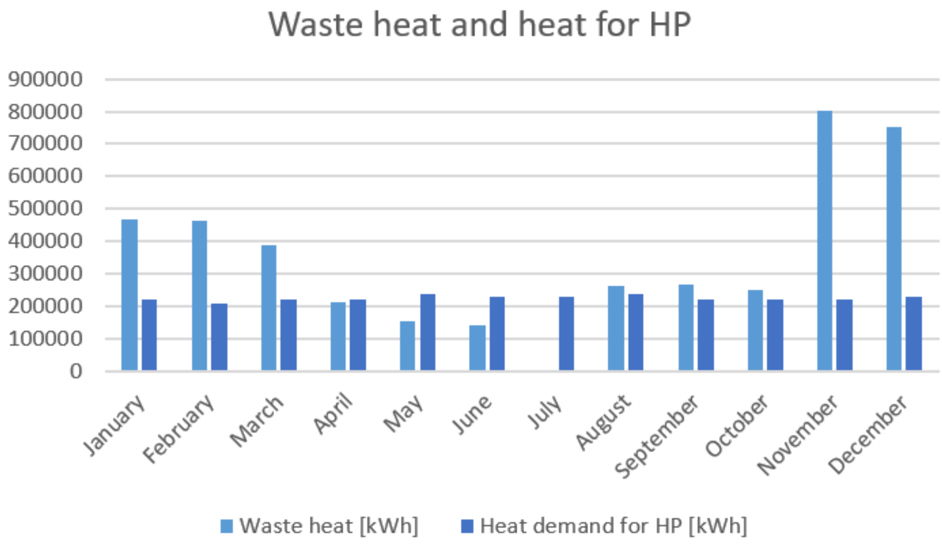

Figure 47: Waste heat potential and DHW demand

Table 27: Waste heat from ice skating rink and coverage of space heating and DHW demand

Month	Waste heat [kWh]	Energy demand [kWh]	Percentage of buildings covered [%]
January	464557	830739	56,0
February	462398	691999	66,7
March	386442	643689	60,0
April	211560	417939	50,6
May	154010	308289	50,0
June	140236	229040	61,1
July	51	229040	0,0
August	260546	238715	109,3
September	267350	340539	78,5
October	250074	546939	45,7
November	803152	714639	112,4
December	751742	879716	85,5

4.2.2 Potential of waste heat from greywater

Every person produces a lot of greywater during a day. About 64% of household's water consumption corresponds to greywater which can be used for heat harnessing and implemented in our system solution. The appliances in households for heat are:

- Showers(baths
- WC basins
- Dishwashers
- Kitchen sinks
- Washing machines

Every building have 20 apartments and we do have 86 buildings in the area. The amount of greywater will vary according to the amount of days in the month. We want to investigate how this internal waste heat source can cover the different energy demands of only space

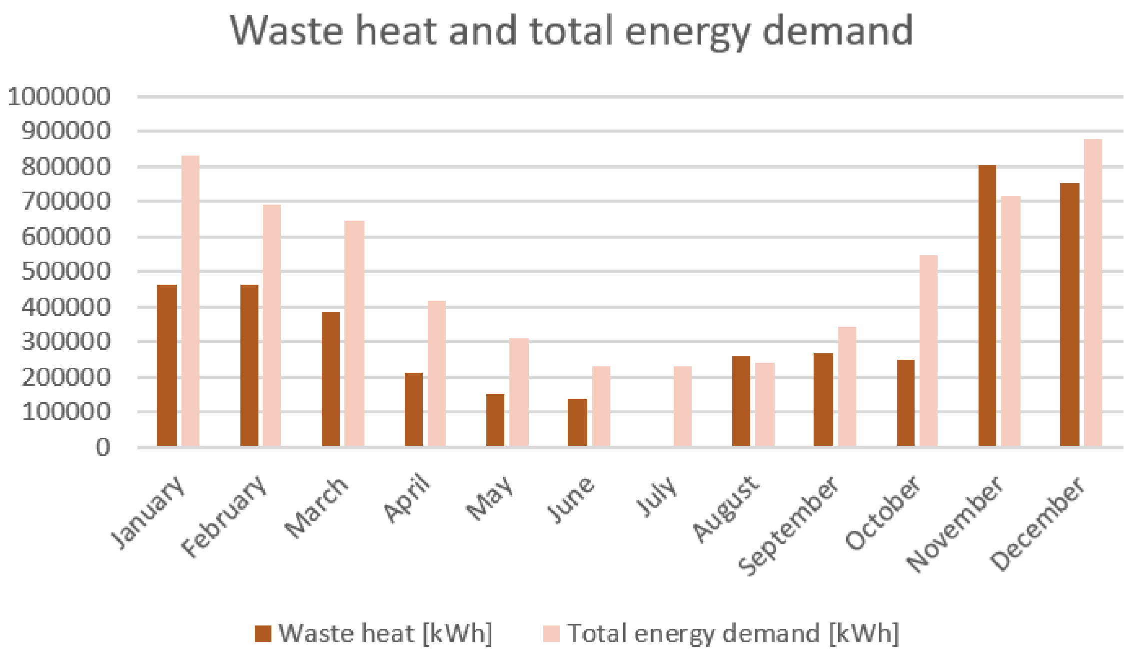

Figure 48: Waste heat potential and total energy demand

heating, as a heat source for the heat pump and the total energy demand of both space heating and heat for heat pump. The waste heat from the greywater will be investigated for the reference building made in Simien. Table 28 shows how this amount varies during a year. The amount of greywater used in this subsection is that 64% of hot tap water can be used for heat harnessing.

Table 28: Waste heat from greywater and coverage of space heating and DHW demand

Month	Q_{GW} [kWh]	Coverage SH [%]	Coverage heat for HP [%]	Coverage total [%]
January	187191	30,6	85,3	22,5
February	178880	37,1	85,3	25,8
March	187191	44,1	85,3	29,1
April	187191	94,3	85,3	44,8
May	203703	292,8	85,3	66,1
June	195447	-	85,3	85,3
July	195447	-	85,3	85,3
August	203703	-	85,3	85,3
September	187191	154,5	85,3	55,0
October	187191	57,1	85,3	34,2
November	187191	37,8	85,3	26,2
December	195447	30,0	85,3	22,2

The greywater does not cover the heating demand for all months. However, as figure 49 indicates, the heat of the greywater can cover the entire space heating demand in March, April and October. In the months from May to September there is no need for heating. The yearly average percentage of covered demand is 67,9%.

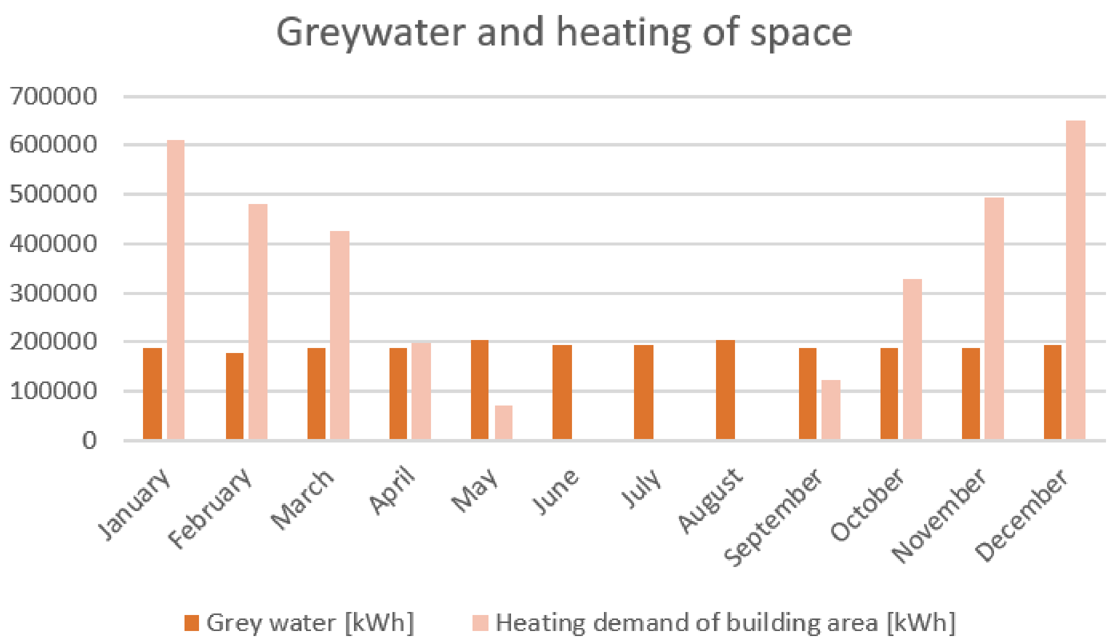

Figure 49: Greywater waste heat and space heating demand

As figure 50 indicates, the heat of the greywater can cover all of the DHW demand during the different months with an average percent of covered demand of 85,3%. This indicates that the system of production of DHW can not be self-driven when implementing greywater.

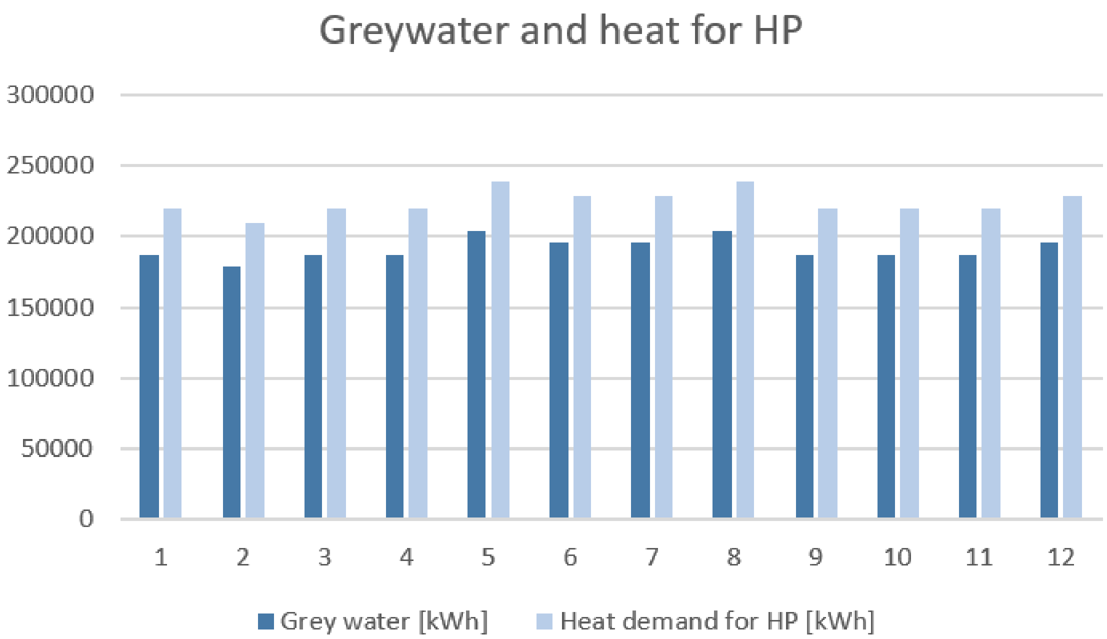

Figure 50: Greywater waste heat and DHW demand

Figure 51 shows how the waste heat coming from greywater can cover both space heating demand and demand of DHW production. The heat from the greywater can cover the entire demand in the months of May to September. The greywater is able to cover this energy demand with a yearly average percentage of 37,8%.

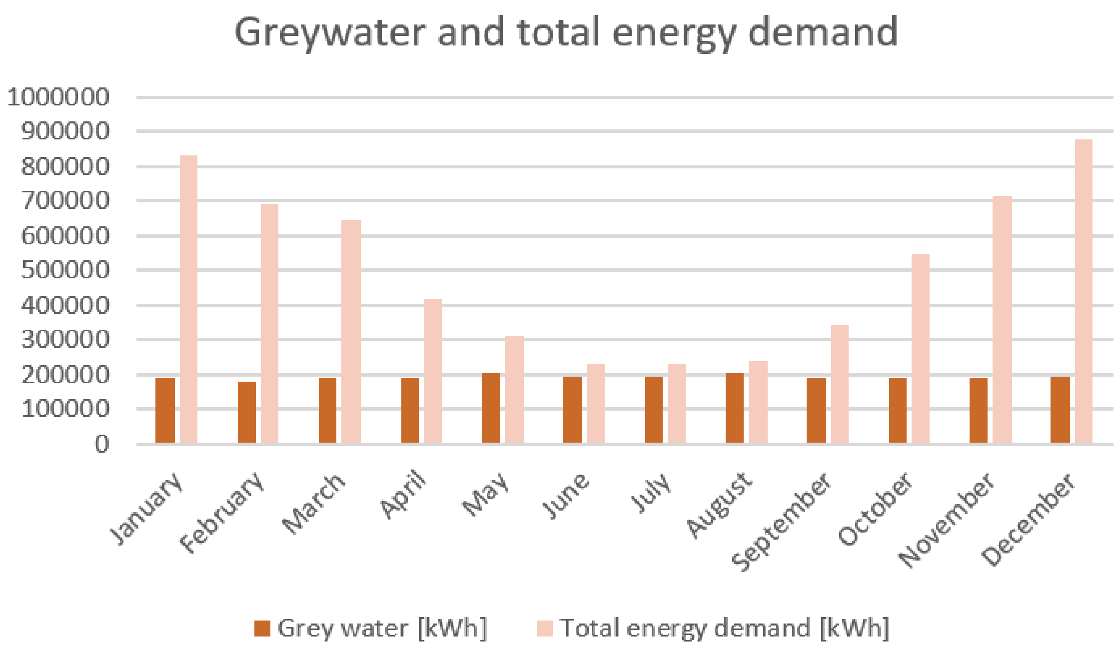

Figure 51: Greywater waste heat potential and total energy demand

4.2.3 Total waste heat potential

We add the waste heat of the ice skating rink and the internal waste heat of greywater inside the building. It is desirable to consider the total potential of those two sources for the reference building simulated in Simien. Table 29 shows the coverage of the energy demand for space heating, DHW demand as well as the coverage of the total energy demand for the reference building made in Simien.

Table 29: Total waste heat and coverage of space heating and DHW demand

Month	Waste heat [kWh]	Coverage SH (%)	Coverage heat for HP (%)	Coverage total (%)
January	652783	106,8	297,6	78,6
February	640681	132,8	305,6	92,6
March	573600	135,2	261,5	89,1
April	398564	200,7	181,7	95,4
May	357710	514,1	149,8	116,0
June	335463	-	146,5	146,5
July	195498	-	85,4	85,4
August	464629	-	194,6	196,6
September	454541	375,1	207,2	133,5
October	437265	133,5	199,3	79,9
November	990343	200,0	451,5	138,6
December	947189	145,6	413,5	107,7

The yearly, average percentage of coverage is 190,7% for space heating, 239,7% for DHW and for both the yearly average coverage is of 106,2%. As Figure 52 indicates, the two waste heat sources submerged together is able to cover the entire energy demand with over 100% for 6 months, the months being May, June, August, September, November and December.

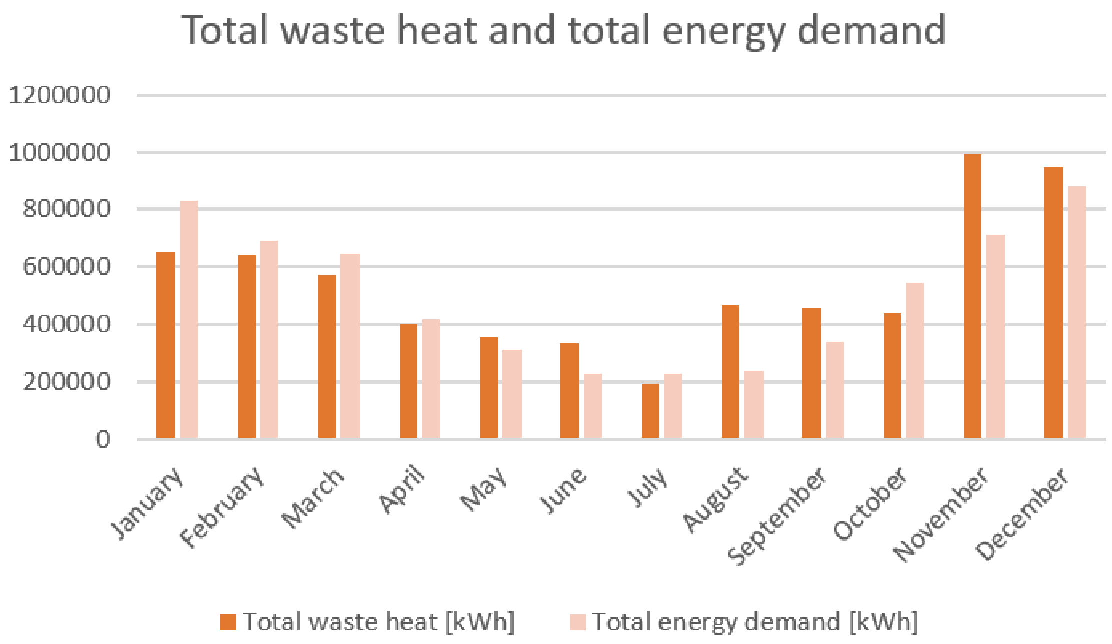

Figure 52: Total waste heat and total energy demand coverage for reference building

4.3 Building stages and energy coverage

It is of interest to analyze the different building stages. The building area will be built in three different stages with an equal amount of buildings being built. We want to analyze how the waste heat from the ice skating rink can cover the energy demand of those three stages. It is also of interest to see how both waste heat from the ice skating rink and greywater together can covert he energy demand.

4.3.1 Waste heat from ice skating rink and total energy demand

It is of interest to analyze how the total energy demand of space heating and DHW demand can be covered by only utilizing the waste heat from the ice skating rink.

Table 30: Waste heat from ice skating rink and percentage of covered total energy demand of three building stages

Month	Stage 1 coverage [%]	Stage 2 coverage [%]	Stage 3 coverage [%]
January	168,1	84,1	56,0
February	200,2	100,1	66,7
March	180,1	90,0	60,0
April	151,7	75,9	50,6
May	149,9	74,9	50,0
June	183,4	91,7	61,1
July	0,1	0,0	0,0
August	327,9	164,0	109,3
September	235,5	117,8	78,5
October	137,2	68,6	45,7
November	337,2	168,6	112,4
December	256,4	128,2	85,5

Table 30 shows the percentage of covered total energy demand when utilizing the total waste heat. The yearly, average value for coverage of stage 1 will be 205,2%. Figure 53 shows the coverage of the energy demand of building stage 1.

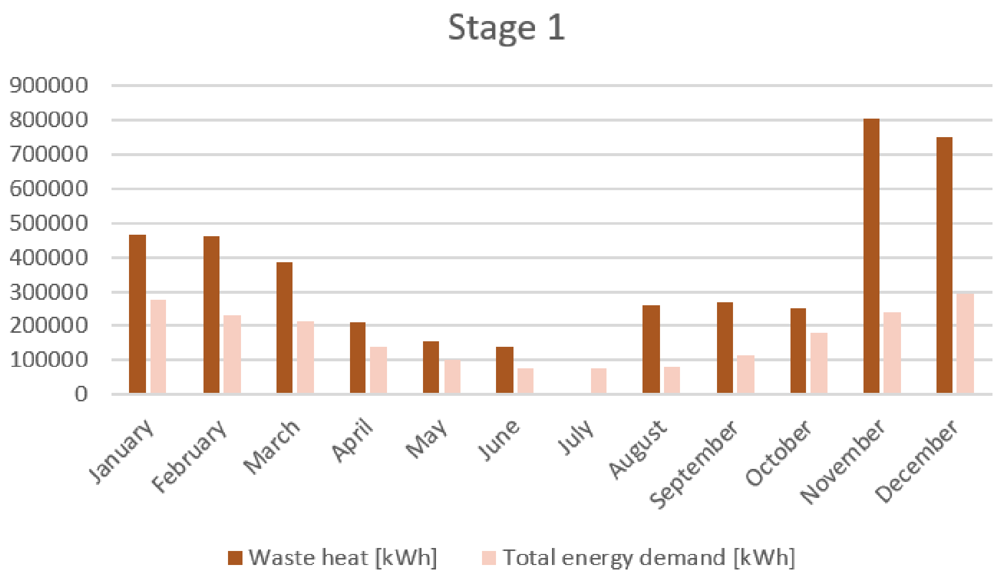

Figure 53: Stage 1 - Waste heat and total energy demand

Figure 53 indicates that the energy demand will be covered entirely after building stage 1 with the exception being July with a coverage of 0,1%. The total waste heat available is too high for every month. Figure 54 shows the total waste heat and total energy demand during building stage 2. The average yearly value for coverage of stage 2 will be 102,6%. During this building stage the amount of total waste heat available do cover the entire energy demand with more than 100% for February, August, September, November and December.

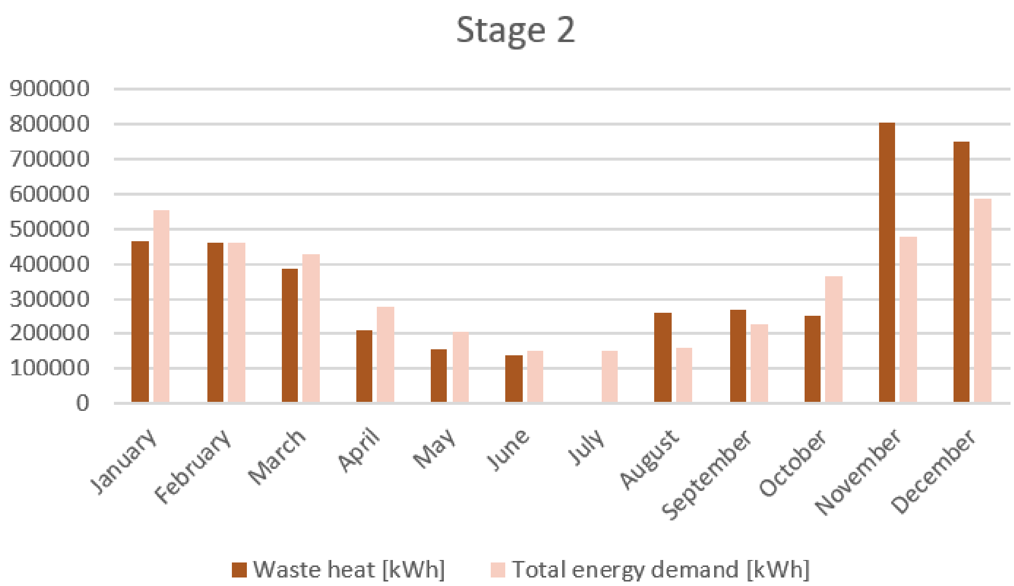

Figure 54: Stage 2 - Waste heat and total energy demand

Figure 55 shows the coverage of energy after building stage 3. The average yearly value for coverage of stage 3 will be 68,4%. This building stage varies quite a lot considering coverage. The waste heat can cover the total energy demand of 5 months. The months which is covered with more than 100% is August and November.

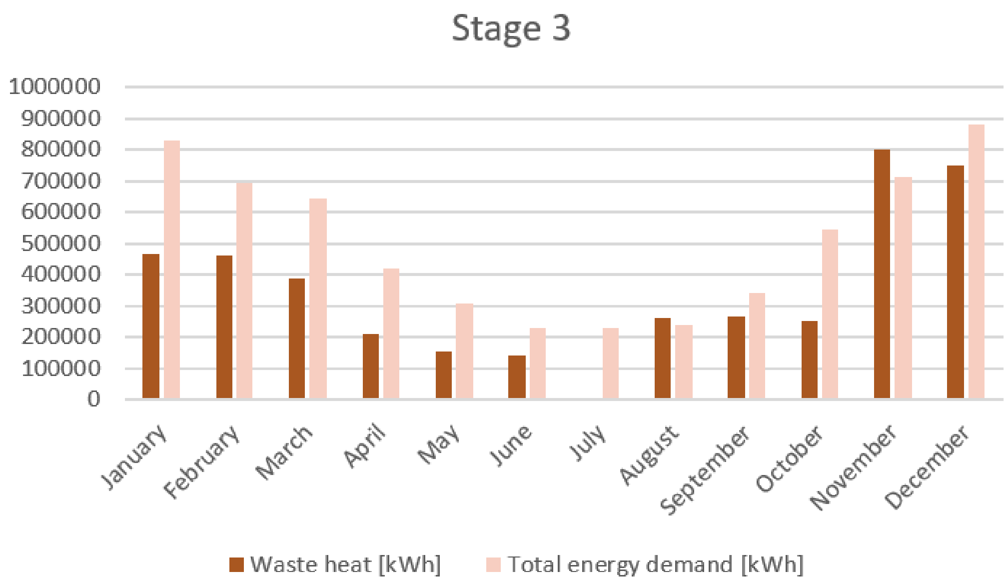

Figure 55: Stage 3 - Waste heat and total energy demand

4.3.2 Total waste heat and total energy demand

We want to analyze how the total energy demand of space heating and DHW can be covered by using the waste heat from the ice skating rink together with the waste heat from greywater. Table 31 shows the percentage of covered total energy demand when utilizing two waste heat sources.

Table 31: Total waste heat and percentage of covered total energy demand of three building stages

Month	Stage 1 coverage [%]	Stage 2 coverage [%]	Stage 3 coverage [%]
January	190,7	106,6	78,6
February	226,1	126,0	92,6
March	209,2	119,1	89,1
April	196,5	120,7	95,4
May	215,9	141,0	116,0
June	268,7	177,0	146,5
July	85,4	85,4	85,4
August	413,2	249,3	194,6
September	290,5	172,7	133,5
October	171,4	102,8	79,9
November	363,4	194,8	138,6
December	243,0	140,4	106,2

The average, yearly value of coverage for stage 1 is 243,0%. Figure 56 shows the coverage of the energy demand after building stage 1 and indicates that the energy demand will be covered entirely after building stage 1. The energy demand is not covered for July.

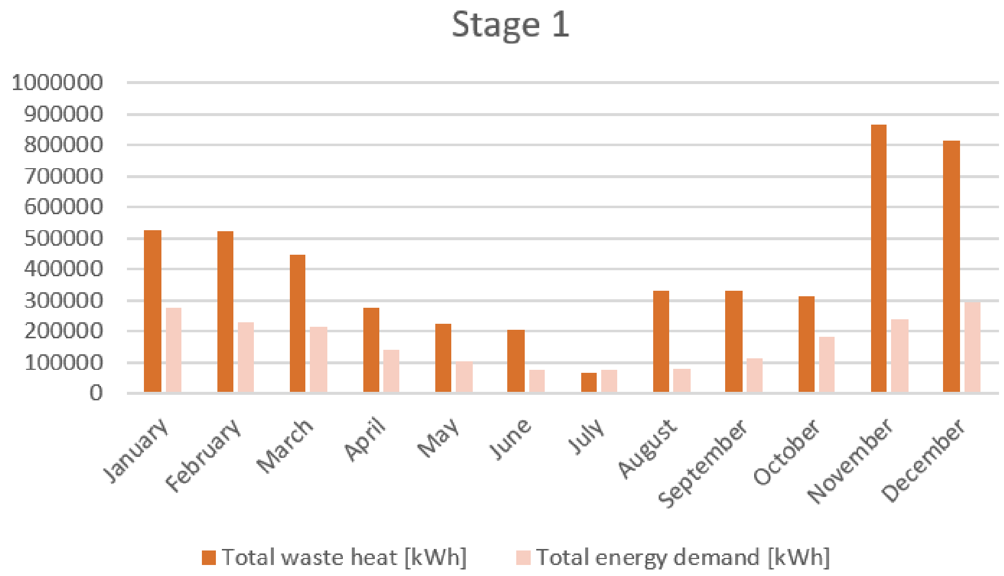

Figure 56: Stage 1 - Total waste heat and total energy demand

Figure 57 shows the total waste heat and total energy demand during building stage 2. The average, yearly value of coverage for stage 2 is 140,4%. The energy demand is not covered for July.

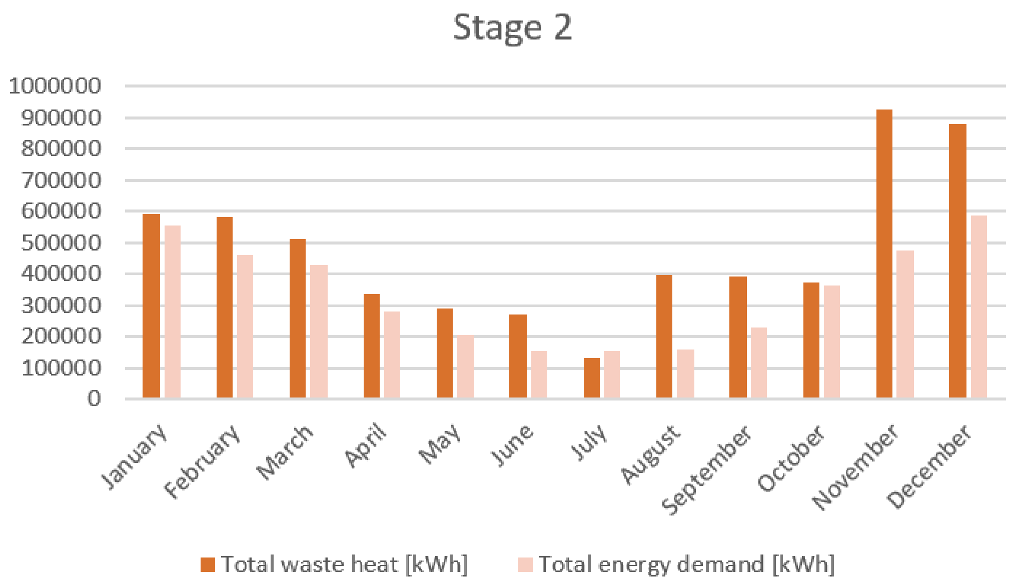

Figure 57: Stage 2 - Total waste heat and total energy demand

Figure 58 shows the coverage of energy after building stage 3. The average, yearly value of coverage for stage 3 is 106,2%. The energy demand is only covered for May, June, August, September, November and December. The energy demand for the other months is covered by over 78%.

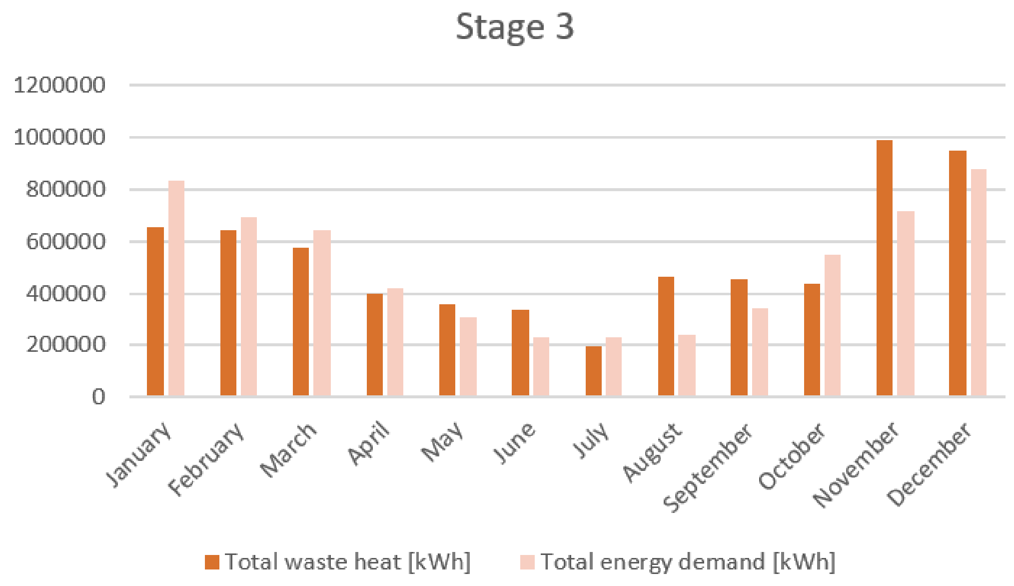

Figure 58: Stage 3 - Total waste heat and total energy demand

4.4 Analysis of working fluids in Coolpack

It is of interest to investigate which working fluid that is the best alternative for the heat pump producing domestic hot water. Coolpack was used to investigate the different working fluid. The working fluid with the highest COP will be used further on in Dymola. The temperatures in the heat pump cycle is set to the same levels. The subsection contains information regarding heat pump cycles which will heat up water from 5-75°C.

4.4.1 Ammonia heat pump

The values of table 17 and 16 will give the following results listed in table 32. The heat pump cycles with ammonia as working fluid is shown in figure 124, 125, 126 and 127 given in appendix B.

Table 32: Ammonia heat pump cycle results for different evaporating temperatures

Calculated	Attempt 1	Attempt 2	Attempt 3	Attempt 4
Evaporating temperature [°C]	5	10	15	20
Q_e [kJ/kg]	833,074	838,179	842,896	847,210
Q_c [kJ/kg]	1341,454	1301,381	1263,497	1227,593
COP	1,64	1,81	2,00	2,23
W [kJ/kg]	508,380	463,203	420,601	380,383
Pressure ratio	9,916	8,315	7,020	5,965

The figures in appendix B represents the log p-h diagram of the four different attempts with different evaporating temperature. Figure 59 shows how the COP changes when the evaporating temperature increases in a temperature range of 0-20°C. It is noticeable that by increasing the evaporating temperature the COP of the heat pump cycle increases.

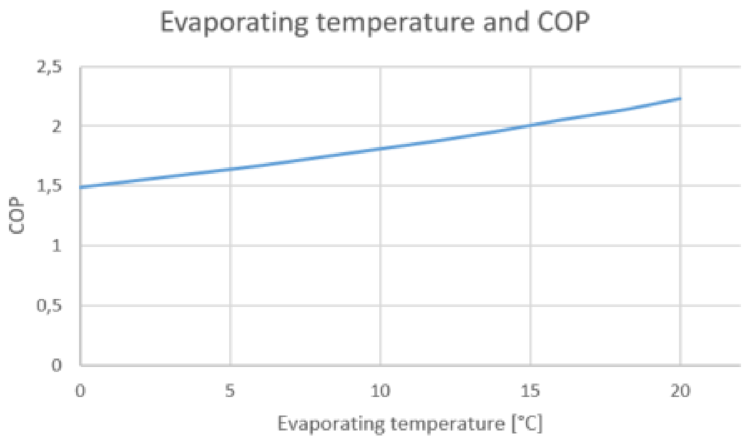

Figure 59: Relationship of evaporating temperature and COP for ammonia

4.4.2 Propane heat pump

The values of table 18 and 16 will give the following results listed in table 33. Figure 128, 129, 130 and 131 in appendix C represents the log p-h diagram of the four different attempts with different evaporating temperature.

Table 33: Propane heat pump cycle results for different evaporating temperatures

Calculated	Attempt 1	Attempt 2	Attempt 3	Attempt 4
Evaporating temperature [°C]	5	10	15	20
Q_e [kJ/kg]	97,829	103,303	108,681	113,946
Q_c [kJ/kg]	218,680	214,497	210,611	206,979
COP	0,81	0,93	1,07	1,22
W [kJ/kg]	120,851	111,194	101,930	93,033
Pressure ratio	6,893	5,964	5,187	4,534

Figure 60 shows how the COP changes when the evaporating temperature increases in a temperature range of 0-20°C. It is noticeable that the COP of the system increases with increasing value of evaporating temperature for propane as well as ammonia.

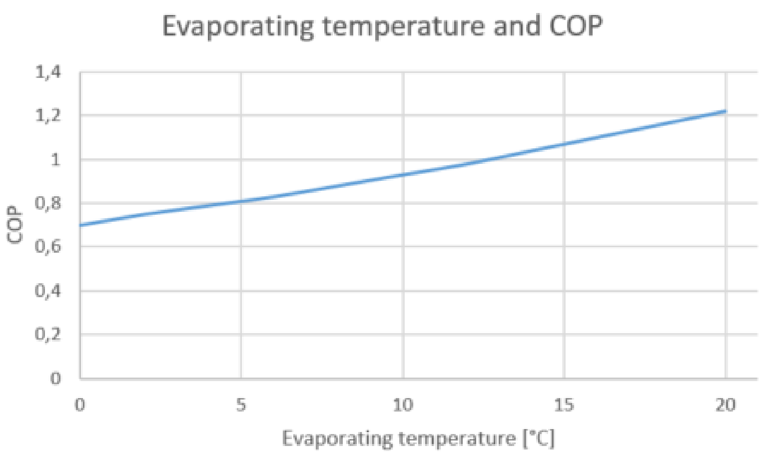

Figure 60: Relationship of evaporating temperature and COP for propane

4.4.3 CO_2 heat pump

The transcritical heat pump cycle for CO_2 is easier to investigate in Coolpack. We investigate the effect of increasing the pressure, increasing the evaporating temperature and increased outlet temperature out of the gas cooler. The values of table 19 is presented in appendix D as figure 132, 133, 134 and 135.

Table 34: CO_2 heat pump cycle results for different pressures

Calculated	Attempt 1	Attempt 2	Attempt 3	Attempt 4
Pressure [kW]	75	90	105	120
Q_E [kW]	4,657	5,097	5,489	5,918
W [kW]	0,848	1,167	1,507	1,868
m [kg/s]	0,033	0,032	0,033	0,035
V_s [m^3/h]	0,940	0,932	0,965	1,014
COP	5,495	4,368	3,644	3,168

The relationship between increasing pressure and the COP is shown in figure 61. This graph is being made for pressure values in the range of 75-130 bar with an interval of 5 bars. The pressure has to be a pressure over the critical pressure of 73,77bar.

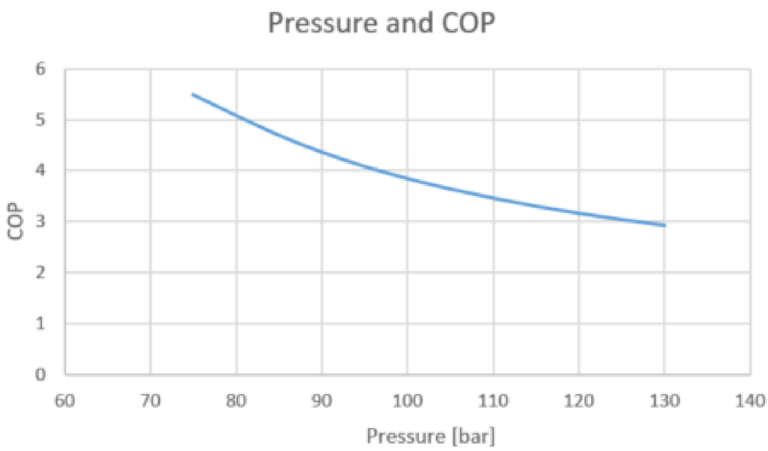

Figure 61: Relationship of pressure versus COP for CO_2

Figure 61 shows that the COP increases when the pressure decreases. Table 34 shows how the evaporating temperature affect the COP and other parameters of the heat pump cycle. The outlet temperature is set to 30°C and the pressure is set to 95bar. The values of table 35 is presented in appendix E as figure 136, 137, 138 and 139.

Table 35: CO_2 heat pump cycle results for different evaporating temperatures

Calculated	Attempt 1	Attempt 2	Attempt 3	Attempt 4
Evaporating temperature [°C]	0	5	10	15
Q_E [kW]	5,403	5,327	5,226	5,095
W [kW]	1,846	1,553	1,278	1,023
m [kg/s]	0,033	0,033	0,033	0,033
V_s [m^3/h]	1,274	1,095	0,941	0,806
COP	2,926	3,431	4,088	4,979

Figure 62 shows the relationship between evaporating temperature and COP value. The values is calculated by using Coolpack with an evaporating temperature in the range of 0-10°C with a pressure of 95bar and an outlet temperature of 30°C. The values of table 36 is presented in appendix E as figure 140, 141, 142 and 143.

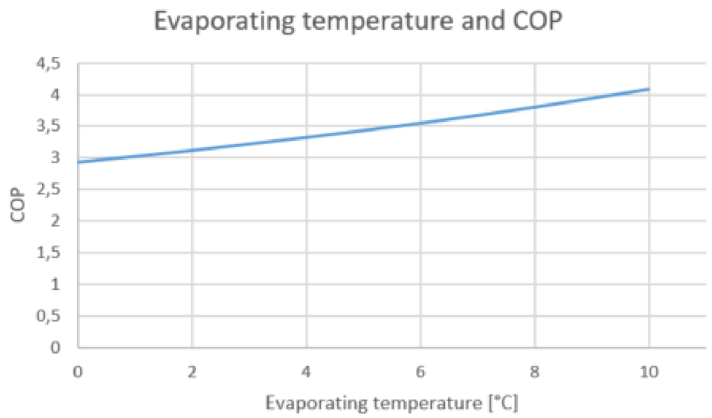

Figure 62: Relationship of evaporating temperature versus COP for CO_2

Table 36 presents the results of trying different outlet temperatures for CO_2. The pressure is set to 95bar and the evaporating temperature is set to 10°C.

Table 36: CO_2 heat pump cycle results for different outlet temperatures (T_4)

Calculated	Attempt 1	Attempt 2	Attempt 3	Attempt 4
Outlet temperature (T_4) [°C]	15	25	35	45
Q_E [kW]	5,540	5,356	5,039	3,757
W [kW]	1,069	1,192	1,403	2,259
m [kg/s]	0,027	0,030	0,036	0,058
V_s [m^3/h]	0,786	0,877	1,033	1,663
COP	5,184	4,495	3,590	1,663

Figure 63 shows the relationship between outlet temperature (T_4) and COP value. The values is calculated by using Coolpack in a range of an outlet temperature 15 to 45°C. Figure 63 indicates that the COP increases with decreasing outlet temperature of the gas cooler.

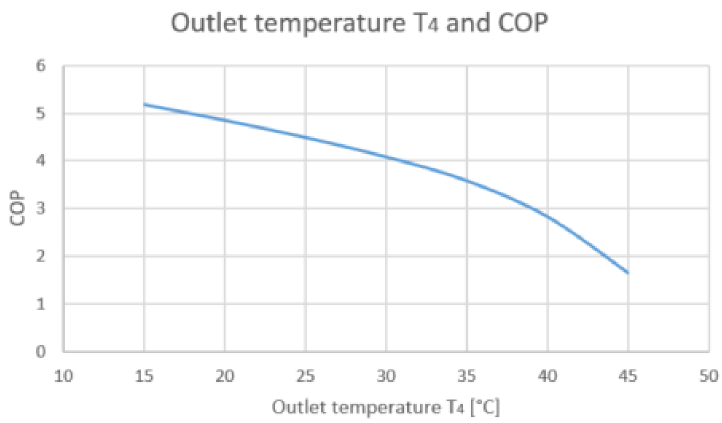

Figure 63: Relationship of outlet temperature (T_4) and COP for CO_2

4.5 Analysis of system in Dymola

This chapter includes information about the models constructed in Dymola and the results generated by the simulations as well as equations from chapter 2.2.1. The simulations made in Dymola will give a more realistic impression of the system than Coolpack which only considers the heat pump cycle of different fluids. We investigate model 2 which includes the CO_2 heat pump due to a higher COP value which was investigated in chapter 4.6. Firstly, the heat pump itself is investigated in heating mode and cooling mode, and further on the heat exchangers for space heating and cooling are investigated.

4.5.1 Geometry of heat exchangers for heat pump

The geometry of the gas cooler and evaporator will be the same for every case investigated. The size is the same for the heating mode and the cooling mode. The geometry and heat transfer coefficient is chosen so that the temperatures and pressures of the system stabilizes. Table 37 shows the geometry of the evaporator. The evaporator is a plate heat exchanger.

Table 37: Geometry of evaporator for CO_2 heat pump

Parameter	Evaporator values
Number of plates	50
Plate length [m]	0,3
Plate width [m]	0,1
Pattern angle in deg	35
Wall thickness [m]	0.75e-3
Pattern amplitude [m]	2e-3
Pattern wave length [m]	12.6e-3
nCells	20
Constant alphaA [W/K]	2100

Table 38 shows the geometry of the gas cooler which gives the most stable conditions. The gas cooler is a tube and tube heat exchanger.

Table 38: Geometry of gas cooler for CO_2 heat pump

Parameter	Tube a	Tube b
Thermal resistance [K/W]	6e-5	6e-5
Inner Diameter [m]	0,01	0,01
nParallel tubes	1	1
Length of tube [m]	20	20
nCells	20	20
Constant alphaA [W/K]	2400	2400

The number of cells defines in how many parts the total length is separated. The more cells you use, the more accurate the calculation will be for each subset. 10 cells should be sufficient in the beginning of making the model, but was regulated to 20 cells to drop from a few bars. An increase of cells will result in an increase of the simulation time. When implementing these values as well as an PI-regulator attached to the compressor, it is possible to achieve the desired temperatures of the system at a stable level. Figure 65 shows the temperatures within the gas cooler.

4.5.2 PI-regulator settings

The PI-regulator is chosen with values so that the compressor stabilizes at a nice level. The necessary values for the PI-regulator attached to the compressor is plotted by the method mentioned in chapter 2.2.4. the values of K_p and T_i for the PI-regulator was set to 5e-6 and 20s for every case since we want to try a system solution with 100bar at the high-pressure side. The values of u_s and u_m is the connector of set point input signal and the connector of measurement input signal. If these values meet, the PI-regulator is working. Both the PI-regulator and correct start values are important when considering how fast the system will stabilize itself. We control a compressor with isentropic efficiency of 0,75.

4.5.3 Heat pump connected to heating circuit

We assume an inlet temperature of water coming into the gas cooler of 5°C, while the desired temperature of domestic hot water is 75°C for the outlet of gas cooler on the water side. Ideally the pressure on the high pressure side should be about 85-95 bar when heating up the water to 80°C, but it is possible to have a pressure in a CO_2 heat pump up to 130bar. We assume that after space heating there will exist a temperature of 35°C which will be fed into the evaporator. When heating the building the temperature of the water out of the evaporator does not have to be specifically 10°C.

The simple heat pump
The model of the heat pump which will be attached to a heat exchanger for space heating is shown in figure 64.

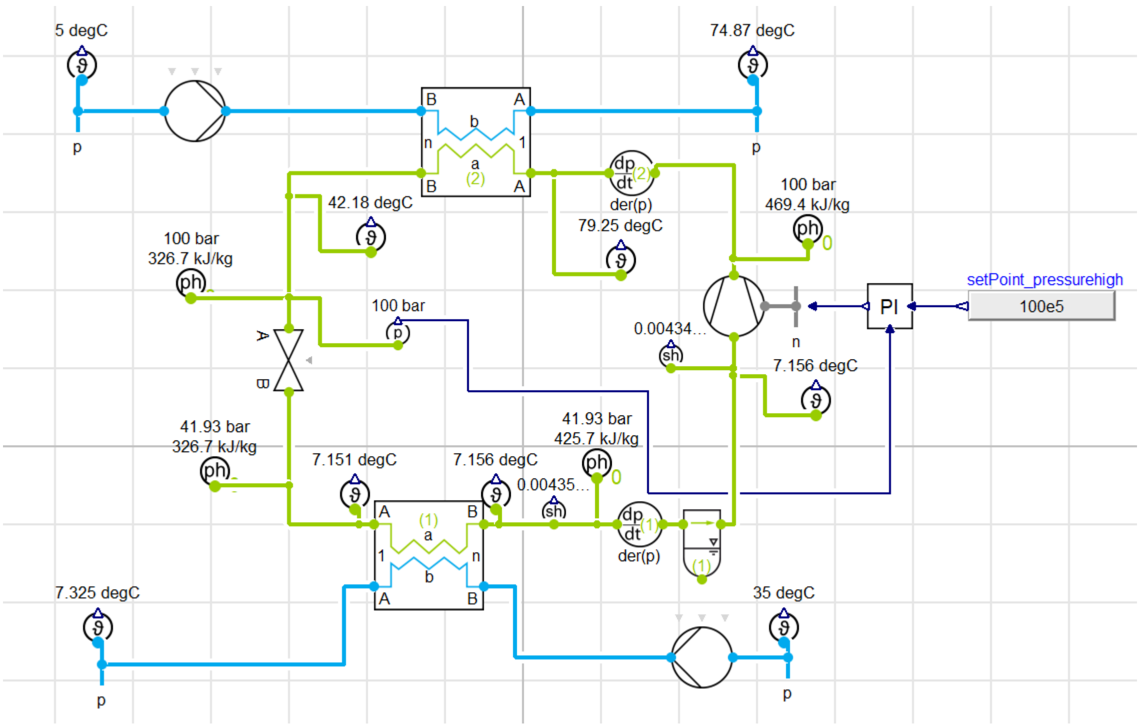

Figure 64: Model of a simple CO_2 heat pump

Hot water is flowing from the ice skating rink to the heat exchanger for space heating, and further on flowing into the evaporator of the heat pump. This flow is set to 0.042 kg/s. The water outlet temperature of the evaporator have a temperature of about 7,3°C and goes into the return line of the low-temperature district heating system. The mass flow of cold city water is set to 0.024kg/s, which is the mass flow of a heat exchanger with a capacity of 7,140kW and a heat difference of water of 70 K. The size of the valve is set to 0.6e-6m^2, the volume of the liquid separator is 6e-3m^3 and the displacement of the compressor is set to 12e-6 m^3.

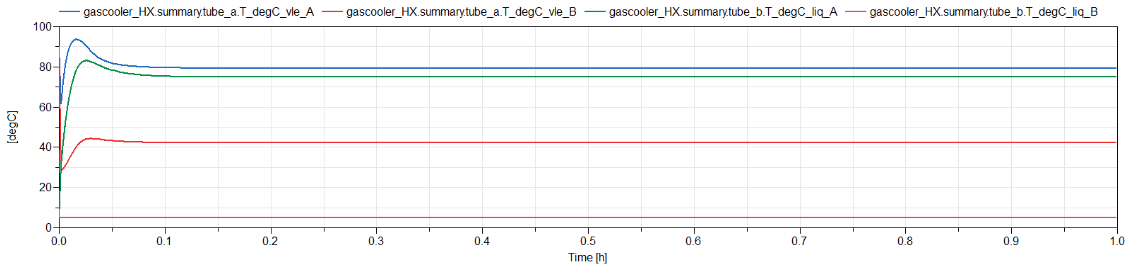

Figure 65: Simple CO_2 heat pump - Gas cooler

In figure 65 the pink line represents the cold city water with a constant temperature of 5°C, while the green line represents the heated city water of 74,9°C. The red line represents the cooled CO_2 with a high temperature of 42,2°C, while the blue line represents the hot CO_2 with a temperature of 79,3°C. Figure 66 shows the temperatures within the evaporator.

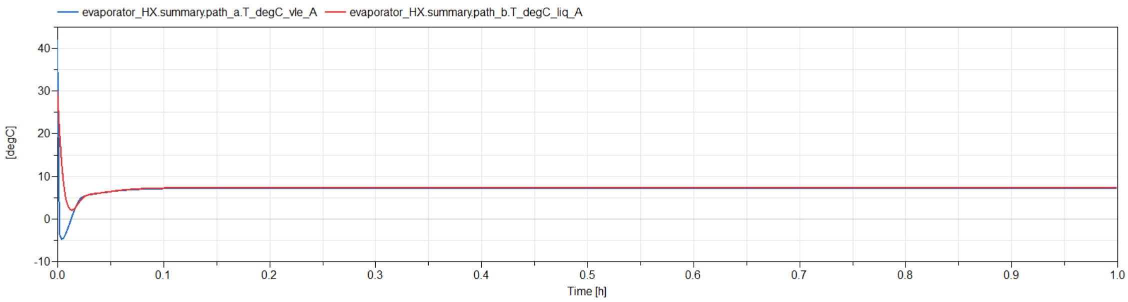

Figure 66: Simple CO_2 heat pump - Evaporator

We assume an inlet temperature of water of 35°C coming from the outlet of heat exchanger for space heating. In figure 66 the red line represents the water at the outlet of the evaporator with a temperature of 7,3°C. The blue line represents the temperature of CO_2 which is 7,2°C. The pressure on the low and high side is equal to 41,9 bar and 100 bar and the graph of the pressures is shown in figure 67. dumbo

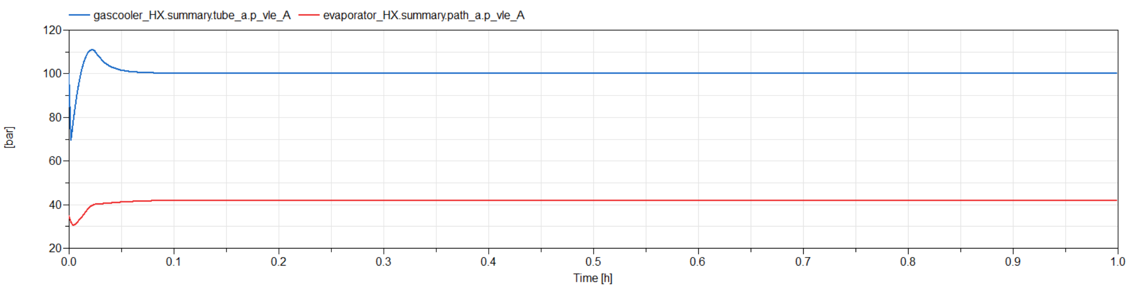

Figure 67: Simple CO_2 heat pump - pressures

The heat pump cycle made in DaVE is shown in figure 68. When using equation 2 from chapter 2.2.1 the enthalpy values of this cycle gives a COP of 3,262. The work of the compressor is 43,77 kJ/kg.

$$COP_{heating} = \frac{469,45 - 326,68}{469,45 - 425,68} = 3,262$$

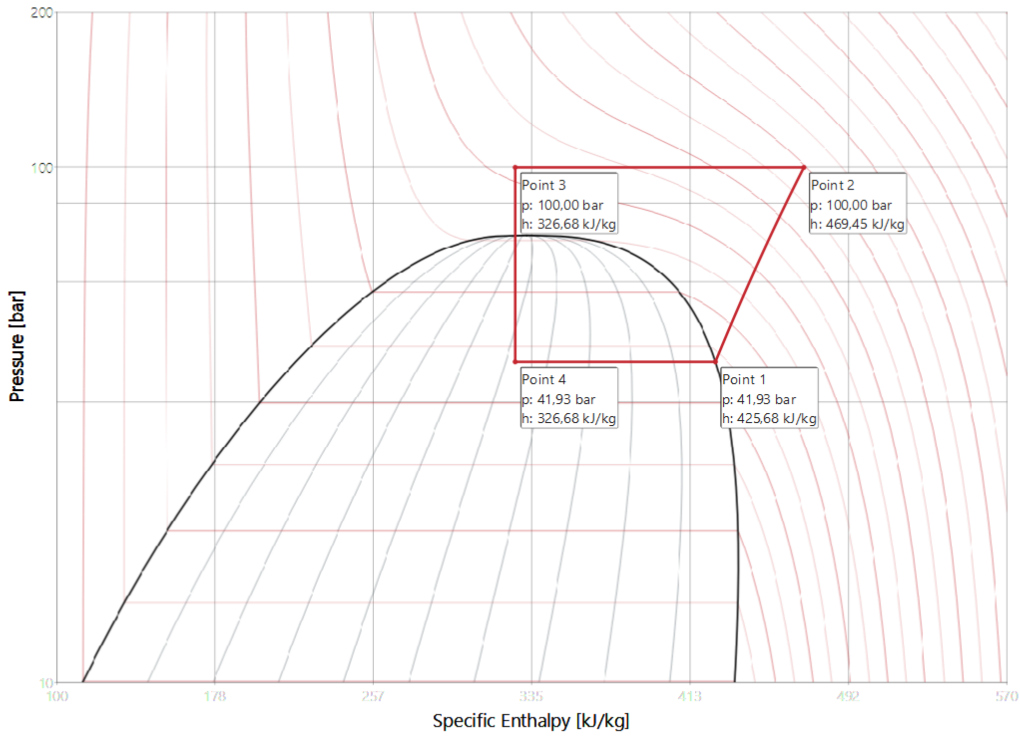

Figure 68: Simple CO_2 heat pump - heat pump cycle

The PI-regulator signal is shown in figure 69. The PI-regulator regulates the behavior of the compressor. As shown in figure 70 the compressor will stabilize itself at a stable speed level of 44,41 Hz.

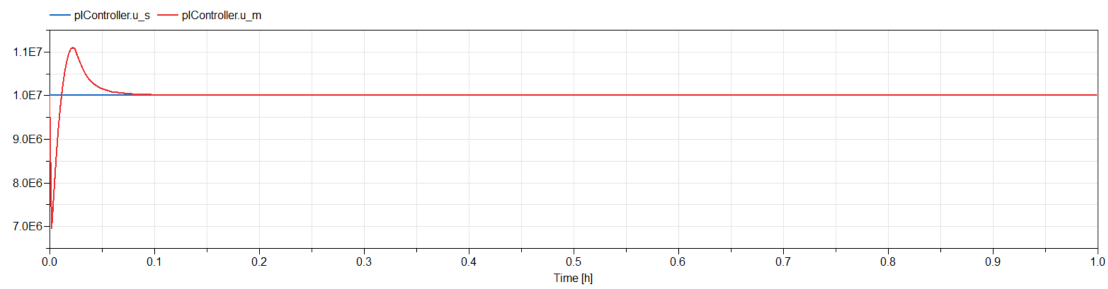

Figure 69: Simple CO_2 heat pump - PI-regulator

It is noticeable that the temperature out of the CO_2 out of the gas cooler is quite high and the temperature into the compressor is quite low. The next step will be to allow an evaporator superheat of about 5K.

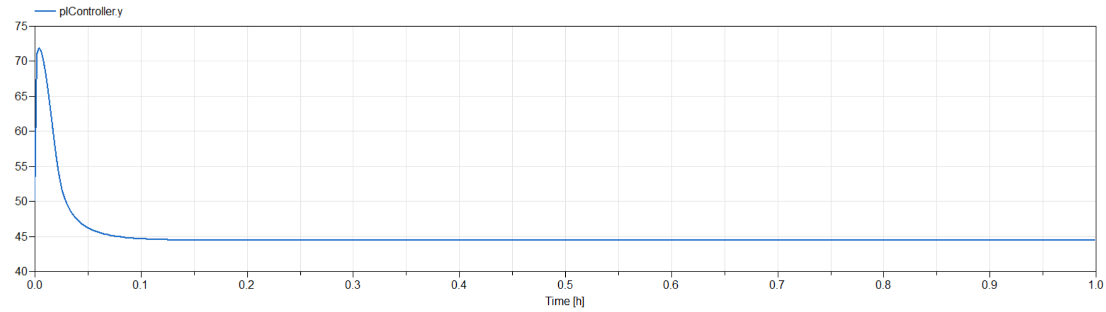

Figure 70: Simple CO_2 heat pump - y-signal of compressor

Simple heat pump with superheat

The model of the simple heat pump with evaporator superheat is shown in figure 71. We will try to create an evaporator superheat by adjusting the size of the liquid receiver. We want to achieve a higher temperature into the compressor to be sure of the liquid being fully vaporized. Hot water is flowing from the ice skating rink to the heat exchanger for space heating, and further on flowing into the evaporator of the heat pump. This flow is set to 0.05 kg/s.

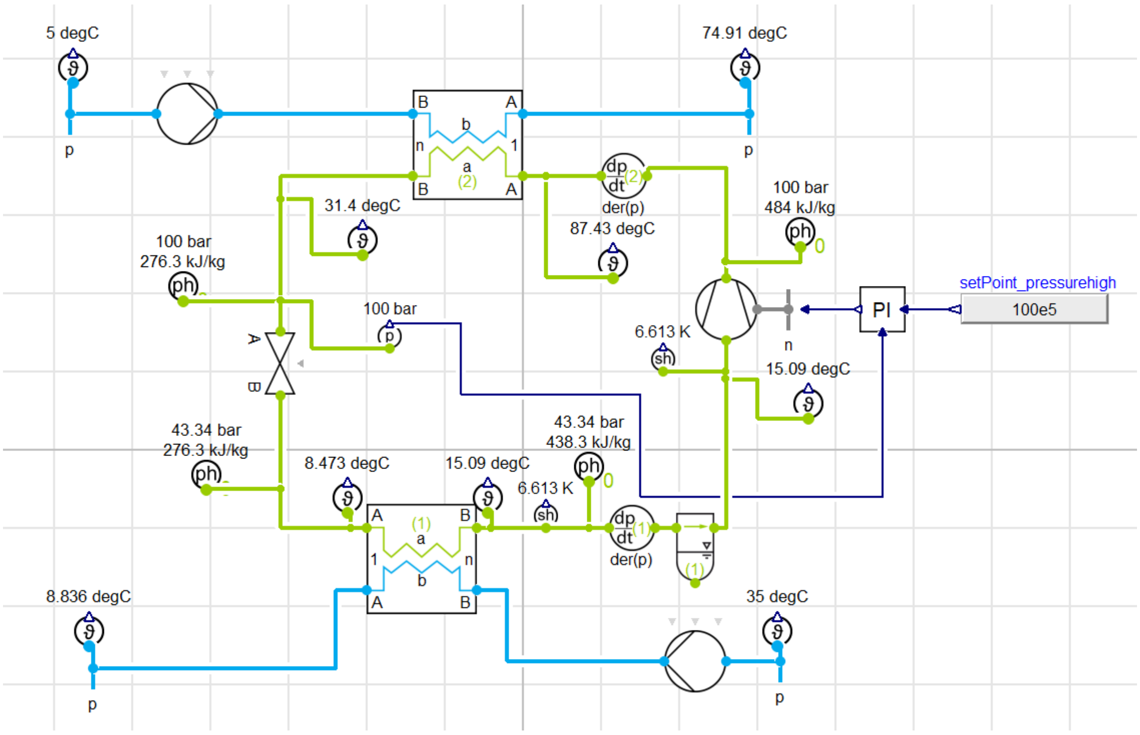

Figure 71: Model of simple CO_2 heat pump with evaporator superheat

The water outlet temperature of the evaporator have a temperature of 8,8°C and goes into the return line of the low-temperature district heating system. The mass flow of cold city water is set to 0.024kg/s, which is the mass flow of a heat exchanger with a capacity of 7,140kW and a heat difference of water of 70 K. The size of the valve is set to 0.365e-6m^2, the volume of the liquid separator is 1.91e-3m^3 and the displacement of the compressor is set to 12e-6 m^3. In figure 72 the pink line represents the cold city water with a constant temperature of 5°C, while the green line represents the heated city water of 74,9°C.

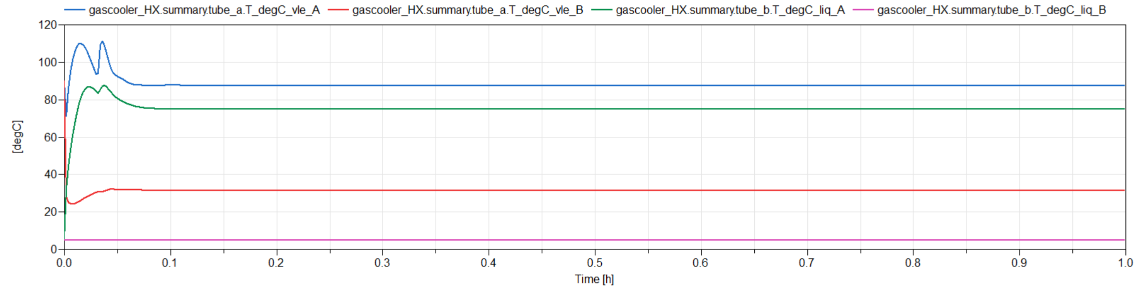

Figure 72: Simple CO_2 heat pump with evaporator superheat - Gas cooler

The red line in figure 72 represents the cooled CO_2 with a high temperature of 31,4°C, while the blue line represents the hot CO_2 with a temperature of 87,4°C. Figure 73 shows the temperatures within the evaporator.

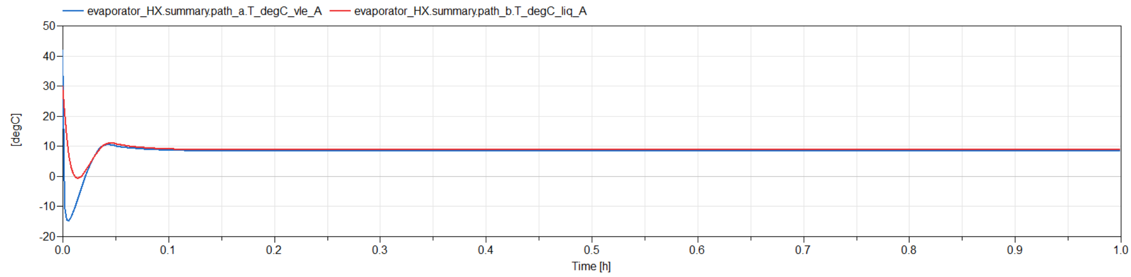

Figure 73: Simple CO_2 heat pump with evaporator superheat - Evaporator

We assume an inlet temperature of water of 35°C coming from the outlet of heat exchanger for space heating. In figure 73 the red line represents the water at the outlet of the evaporator with a temperature of 8,8°C. The blue line represents the inlet temperature of CO_2 which is 8,5°C, while the outlet temperature is 15,1°C. The pressure on the low and high side is equal to 43,3 bar and 100 bar and the graph of the pressures is shown in figure 74.

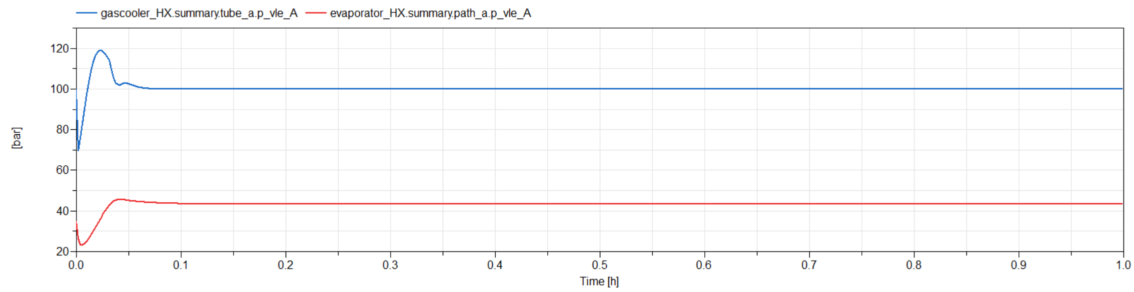

Figure 74: Simple CO_2 heat pump with evaporator superheat - pressures

The heat pump cycle made in DaVE is shown in figure 75. When using equation 2 from chapter 2.2.1 the enthalpy values of this cycle gives a COP of 4,543. The work of the compressor is 45,72 kJ/kg.

$$COP_{heating} = \frac{484,00 - 276,30}{484,00 - 438,28} = 4,543$$

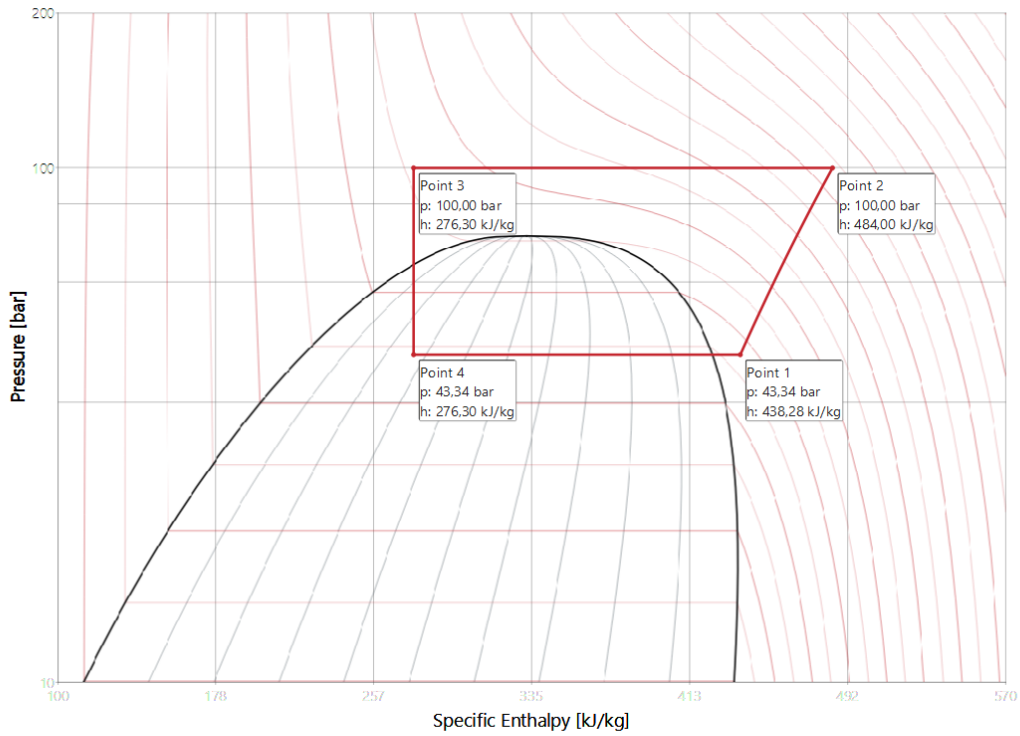

Figure 75: Simple CO_2 heat pump with evaporator superheat - heat pump cycle

By allowing 6,6K of superheat in our heat pump, this results in a rise of the COP value of 1,281. The PI-regulator signal is shown in figure 76.

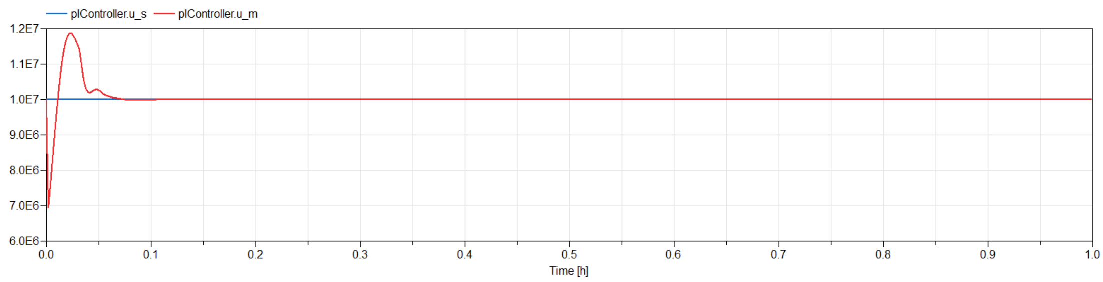

Figure 76: Simple CO_2 heat pump with evaporator superheat - PI-regulator

As shown in figure 77 the compressor will stabilize itself at a stable level of 32,24. The compressor do not run as fast as for the model without superheat which used 44,1 Hz.

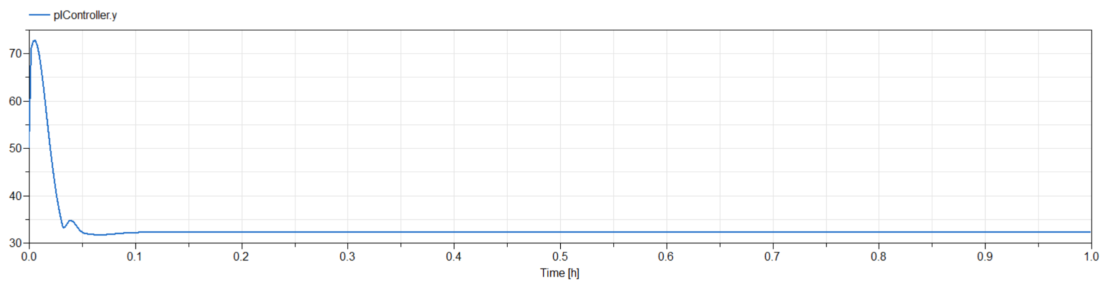

Figure 77: Simple CO_2 heat pump with evaporator superheat - y-signal of compressor

It noticeable that the temperature out of the CO_2 out of the gas cooler is lower when including superheat of 6,6K in our simple heat pump. The temperature into the compressor is higher with a value of 15,1°C. It is desirable to lower the temperature coming out of the gas cooler further and increase the temperature into the compressor even more. We implement an internal heat exchanger which will rise the amount of system superheat.

Heat pump with internal heat exchanger and superheat
The implementation of an internal heat exchanger is shown in figure 78. This implementation will contribute to a lower temperature level of the CO_2 at the outlet of the gas cooler and a higher temperature into the compressor.

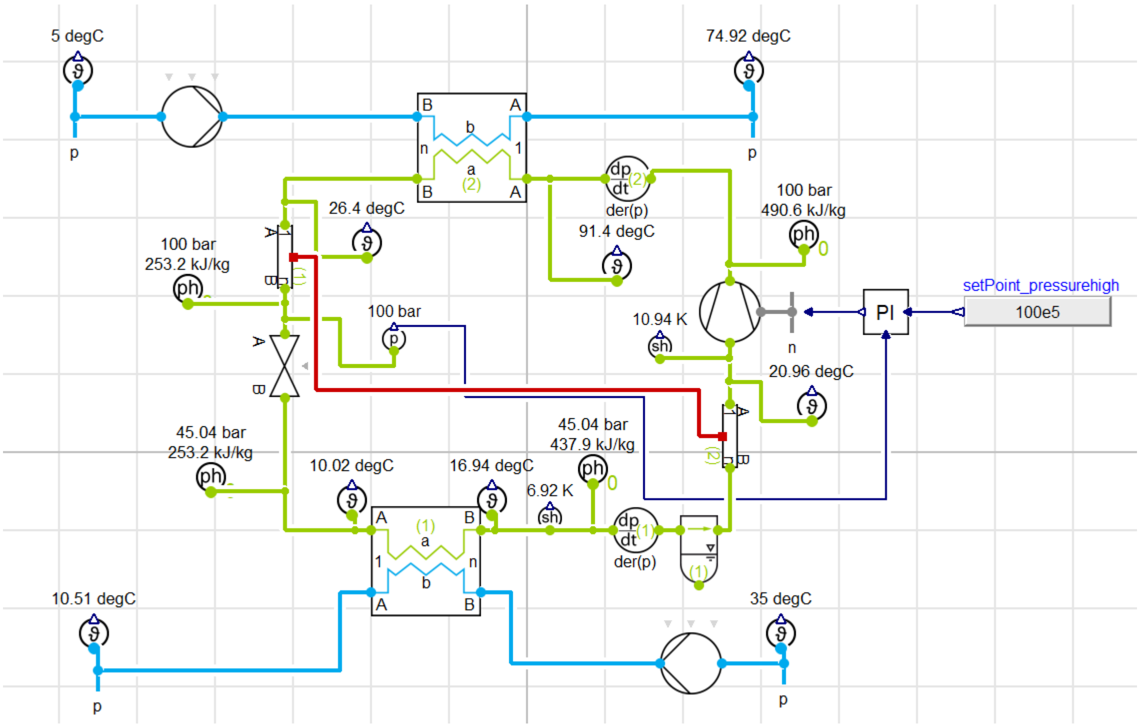

Figure 78: Model of heat pump with internal heat exchanger and superheat

Hot water is flowing from the ice skating rink to the heat exchanger for space heating is set to 0.055 kg/s. The water outlet temperature of the evaporator have a temperature of about 10,5°C and goes into the return line of the low-temperature district heating system. The mass flow of cold city water is still the same and set to 0.024kg/s. The size of the valve is still 0.32e-6m^2 and the volume of the liquid separator is the same and set to 2.6e-3m^3.

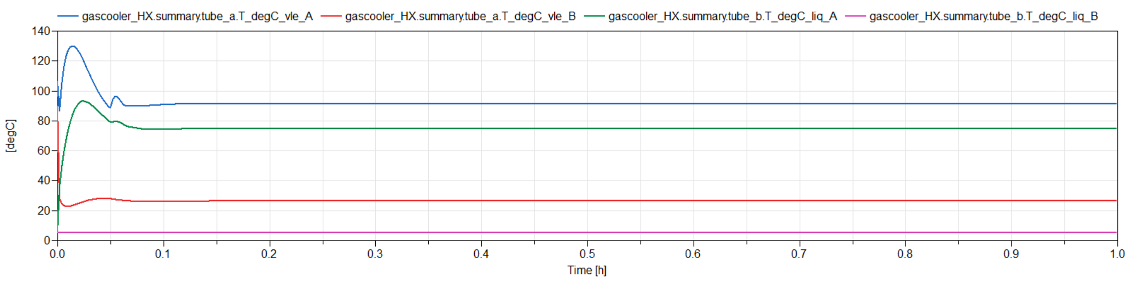

Figure 79: CO_2 heat pump with internal heat exchanger and superheat - Gas cooler

In figure 79 the pink line represents the cold city water with a temperature of 5°C, while the green line represents the heated city water of 74,9°C. The red line represents the cooled CO_2 with a temperature of 26,4°C, and the blue line represents the hot CO_2 with a temperature of 91,4°C. Figure 80 shows the temperatures within the evaporator. We assume an inlet temperature of water of 35°C coming from the outlet of heat exchanger for space heating.

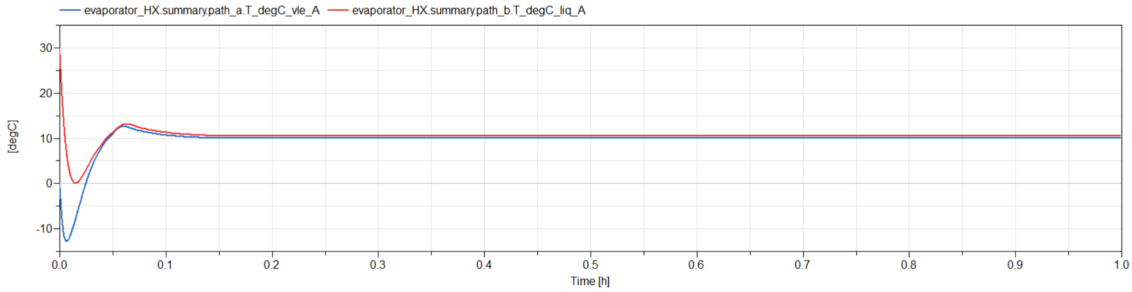

Figure 80: CO_2 heat pump with internal heat exchanger and superheat - Evaporator

In figure 80 the red line represents the water at the outlet of the evaporator with a temperature of 10,5°C. The blue line represents the inlet temperature of CO_2 which is 10,0°C while the outlet temperature will be 16,9°C. The pressure on the low and high side is equal to 45,0 bar and 100 bar and the graph of the pressures is shown in figure 81.

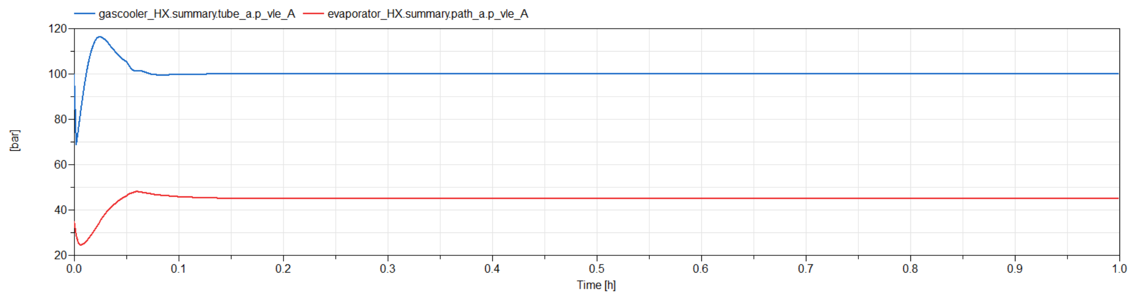

Figure 81: CO_2 heat pump with internal heat exchanger and superheat - pressures

The heat pump cycle made in DaVE is shown in figure 83. When using equation 2 from chapter 2.2.1 the enthalpy values of this cycle gives a COP of 4,506. The compressor have a work of 52,69 kJ/kg. The PI-regulator has the same values as for the model without internal heat exchanger. The signal of the PI-regulator is shown in figure 82.

$$COP_{heating} = \frac{490,62 - 253,18}{490,62 - 437,93} = 4,506$$

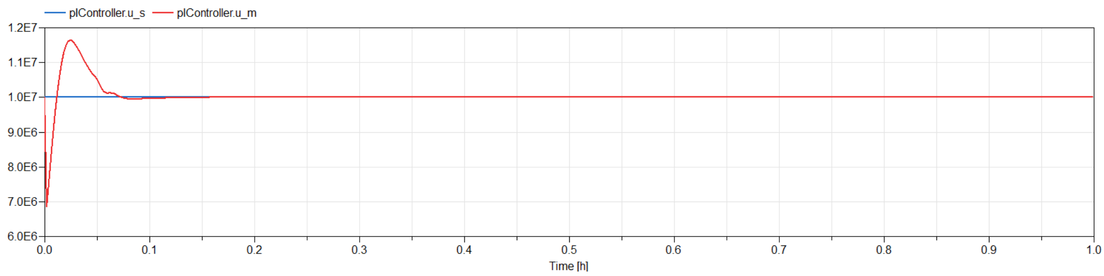

Figure 82: CO_2 heat pump with internal heat exchanger and superheat - PI-regulator

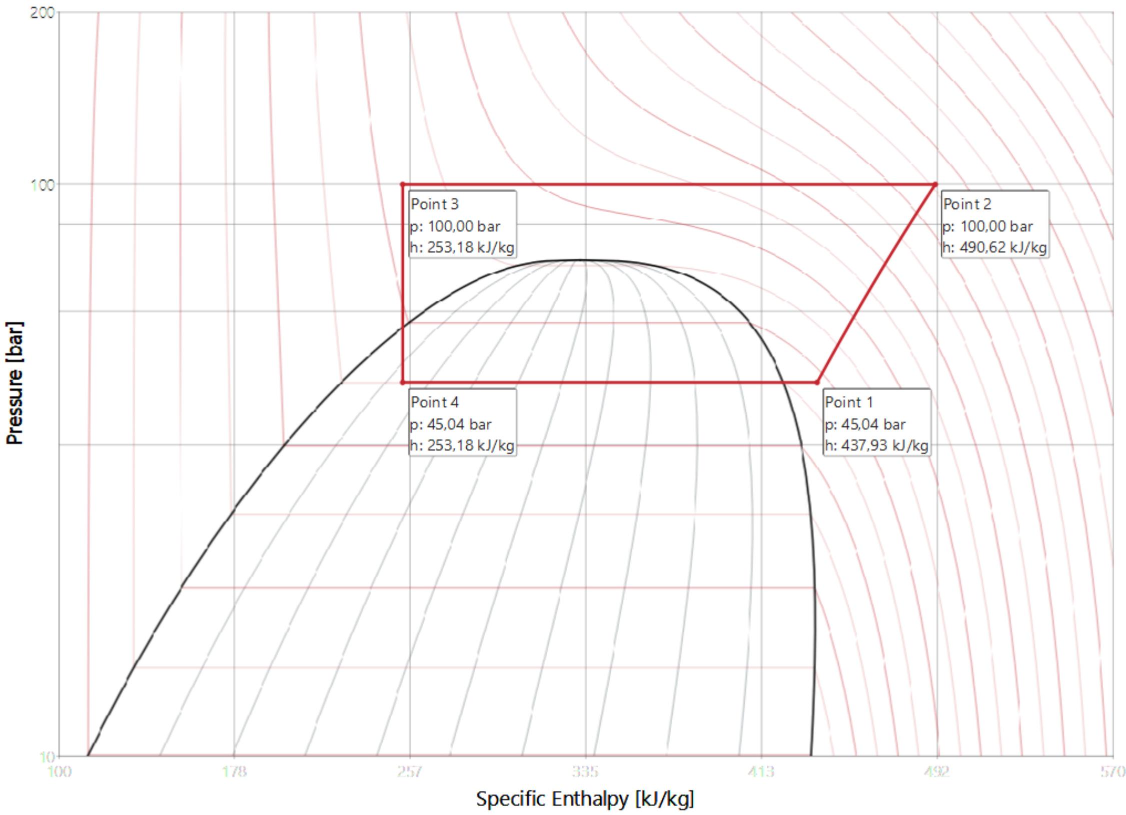

Figure 83: CO_2 heat pump with internal heat exchanger and superheat - heat pump cycles

The PI-regulator regulates the behavior of the compressor. As shown in figure 84 the compressor will stabilize itself at a stable level of 29,27Hz.

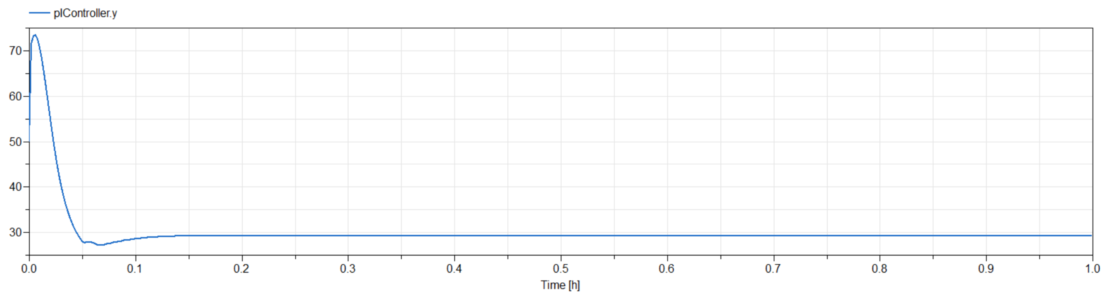

Figure 84: CO_2 heat pump with internal heat exchanger and superheat - y-signal

This model have a lower temperature of the CO_2 coming out of the gas cooler and a higher temperature coming into the compressor. This model have a higher temperature going into the return line of the low-temperature district heating system.

Heat pump in heating mode with IHX and user profile

The next step is to consider this heat pump when the user profile is attached to the model. The data attached to the model is shown in figure 85 as the inlet mass flow. We want to simulate one day to see how the demand changes. According to the given data, the demand is shown in figure 86. The highest demand appears at the 4th of January, and this day begins after 72 hours into the year. We simulate the mass flow of the demand from 72 hours to 96 hours.

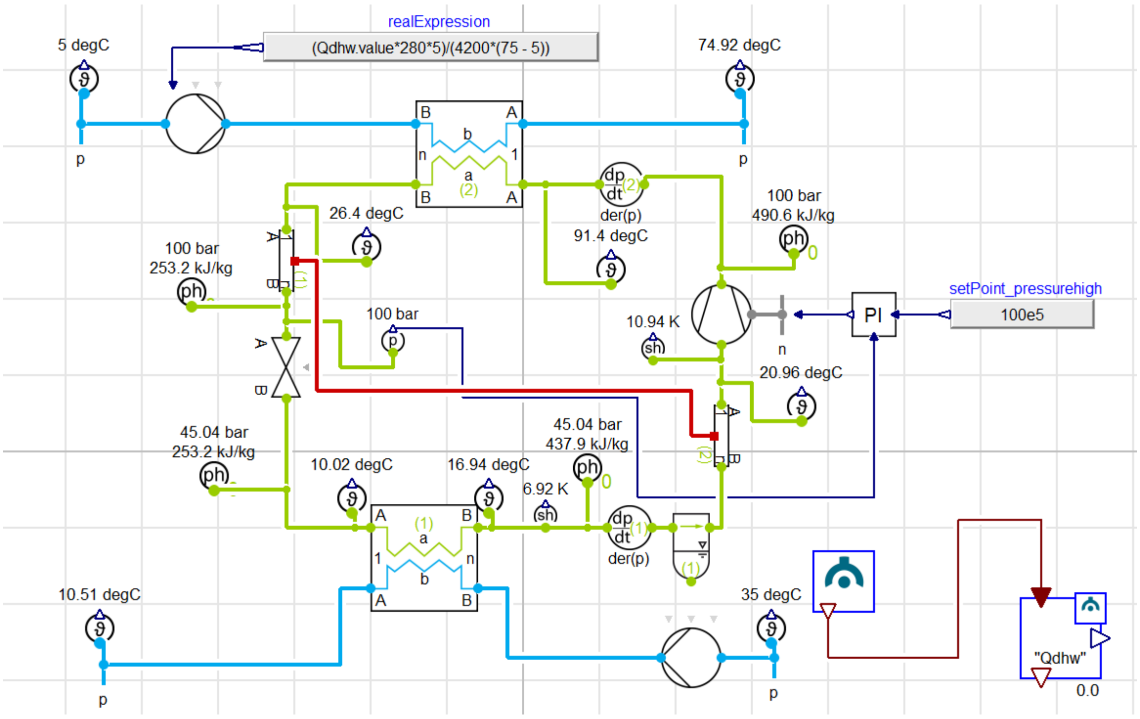

Figure 85: CO_2 heat pump with user profile

As figure 85 indicates, the simple pump will vary due to the water demand coming from the passive house building. This shows the mass flow which is needed to cover the demand during 24 hours at the day with the highest demand.

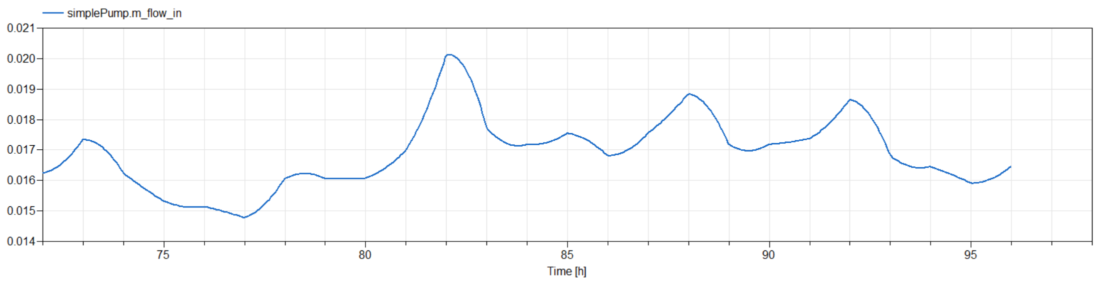

Figure 86: CO_2 heat pump with mass flow demand from user profile

By evaluating the mass flow generated as shown in figure 86 it is possible to calculate the volume of the thermal storage tanks which is needed to cover the total energy demand of domestic hot water for one building. It is noticeable that the biggest use of domestic hot water according to the data given is at the given time of 7:00, 9:00, 11:00, 19:00 and 22:00, in which 7:00, 19:00 and 22.00 have the highest peaks. The information about specific values of domestic hot water demand given in W/m^2 will be used to calculate how big the water tanks should be. This will be done and discussed in chapter 4.7.7.

4.5.4 Heat pump connected to cooling circuit

The temperature of water which goes into the heat exchanger for cooling should be 10°C in order to gain a temperature level of 15°C at the water side going into the building for cooling purposes. We will have an inlet temperature of water into the evaporator of 15°C which comes from the return line of the space cooling circuit. It is desirable to investigate the COP value of both the heating of DHW as well as the COP value of producing cold water for space cooling.

Simple heat pump
The heat pump with a very small amount of superheat is shown in figure 87. This flow is set to 0.23 kg/s to reach the desired temperature levels in the system. The mass flow of cold city water is set to 0,024kg/s, and the temperature of the evaporator have an outlet temperature of 10,0°C which goes into the heat exchanger for space cooling.

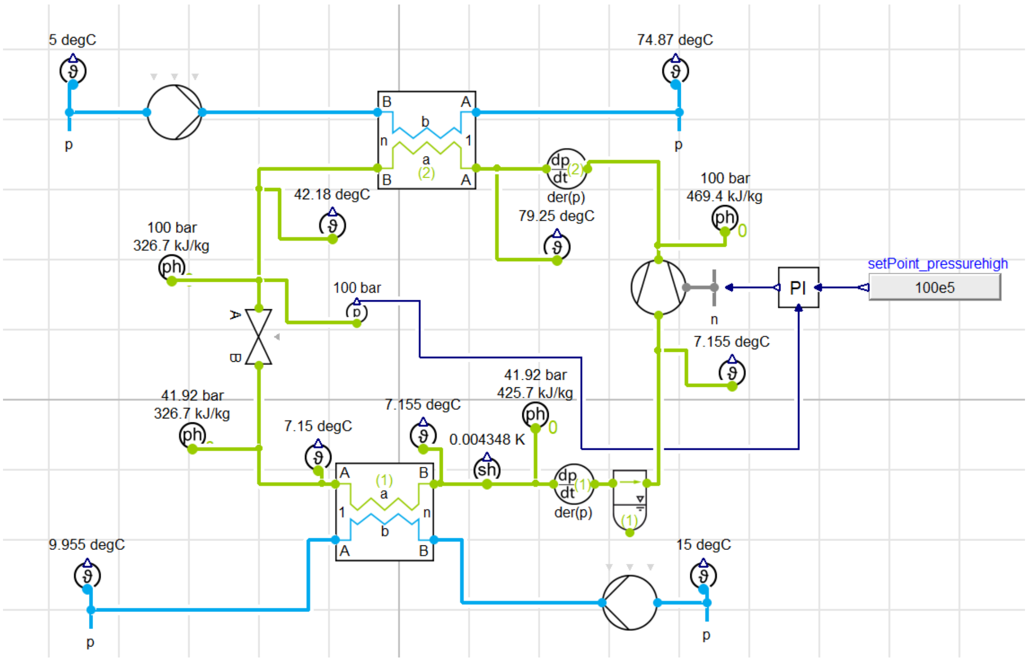

Figure 87: Simple CO_2 heat pump

Figure 88 shows the temperatures within the gas cooler in the heat pump. The size of the valve is set to 0.6e-6 m^2, the volume of the liquid separator is still 6e-3 m^3.

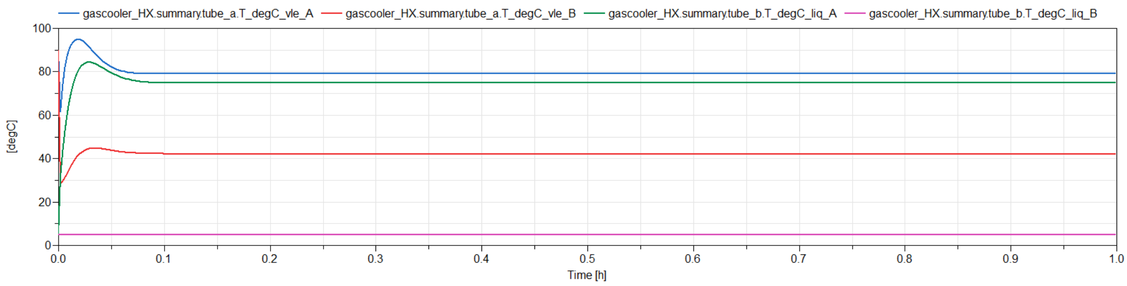

Figure 88: Simple CO_2 heat pump - Gas cooler

In figure 88 the pink line represents the cold city water with a constant temperature of 5,0°C, while the green line represents the heated city water of 74,9°C. The red line represents the cooled CO_2 with a temperature of 42,2°C, while the blue line represents the hot CO_2 with a temperature of 79,3°C. Figure 89 shows the temperatures within the evaporator.

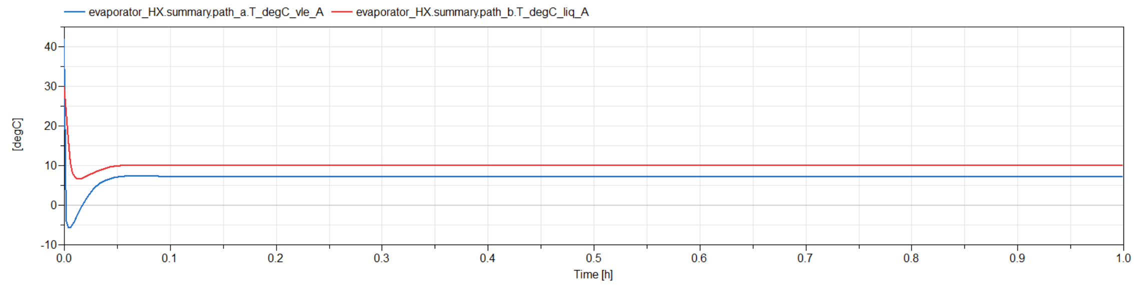

Figure 89: Simple CO_2 heat pump - Evaporator

To cool the building, we need 10,0°C into the heat exchanger for space cooling. Therefore, the evaporator is designed in such a way that this is possible. In figure 89 the red line represents the water at the outlet of the evaporator with a temperature of 10,1°C. The blue line represents the inlet temperature of CO_2 which is 7,2°C.

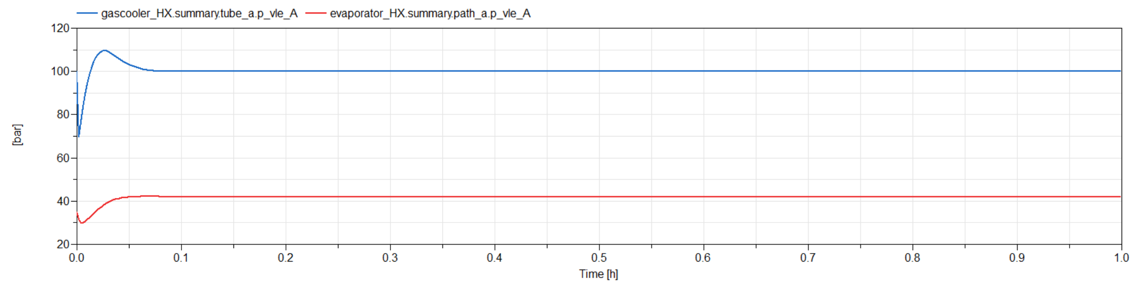

Figure 90: Simple CO_2 heat pump - pressures

Figure 90 shows the pressure of CO_2 at the low and high side with values of 41,9 bar and 100,0 bar. The behavior of the PI-regulator is shown in figure 91.

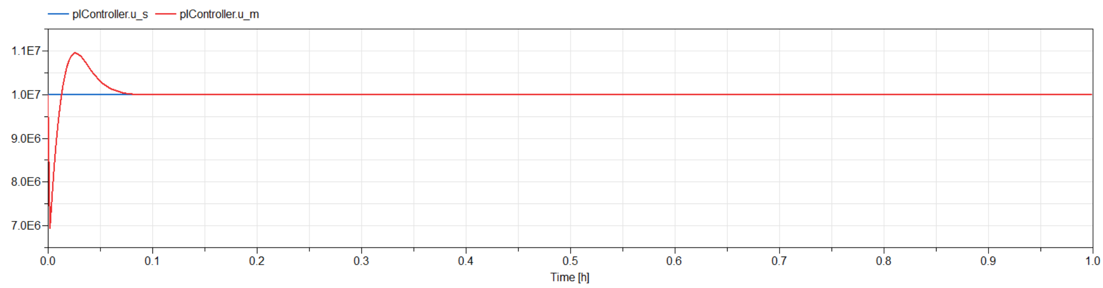

Figure 91: Simple CO_2 heat pump - PI-regulator

Figure 92 shows the compressor behavior when implementing the PI-regulator. It stabilizes around 44,41 Hz.

The heat pump cycle made in DaVE us shown in figure 93. When using equation 2 from chapter 2.2.1 the enthalpy values of this cycle gives a COP of 2,262 when the heat pump is in cooling mode The work of the compressor is 43,77 kJ/kg. The heat pump in cooling mode will have a COP of 2,262.

$$COP_{heating} = \frac{469,45 - 326,69}{469,45 - 425,68} = 3,262$$

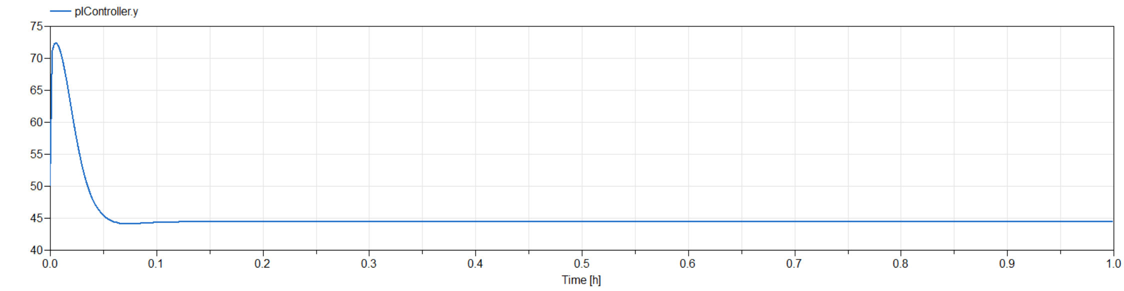

Figure 92: Simple CO_2 heat pump - y-signal of compressor

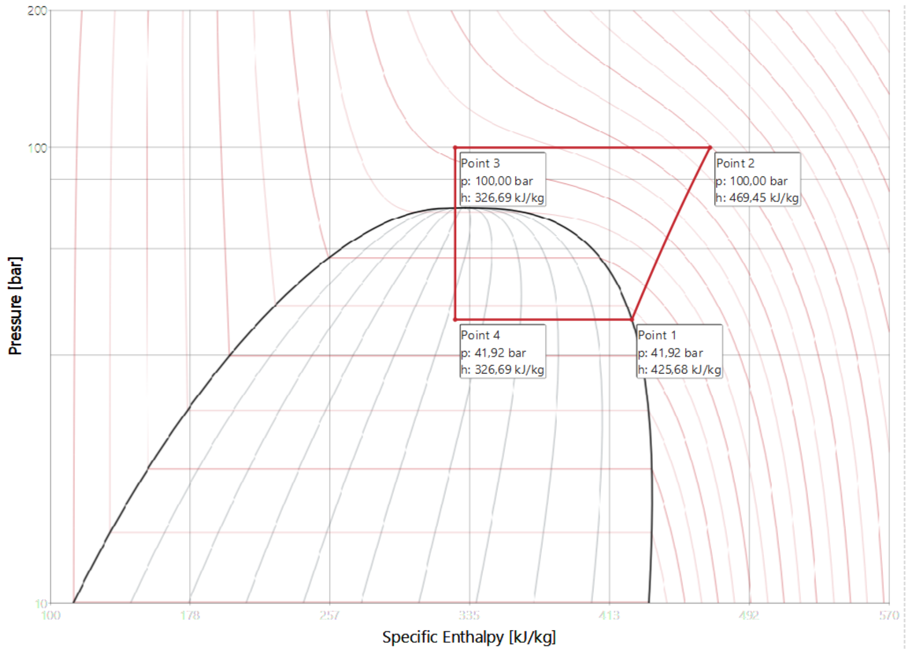

Figure 93: Simple CO_2 heat pump - heat pump cycle

Simple heat pump with superheat

We want to increase the COP value further. Therefore we allow superheating of the evaporator of about 5K. The model is shown in figure 95. Hot water is flowing from the ice skating rink to the heat exchanger for space heating is set to 0.26 kg/s. The water outlet temperature of the evaporator have a temperature of about 10,1°C and goes into heat exchanger for space cooling. The mass flow of cold city water is still the same and set to 0.024kg/s. The size of the valve is 0.365e-6m^2 and the volume of the liquid separator is the same and set to 1.35e-3m^3.

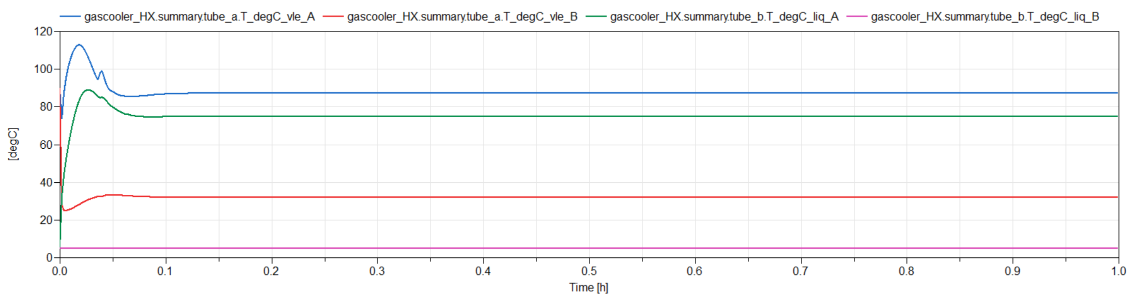

Figure 94: Simple CO_2 heat pump with superheat - Gas cooler

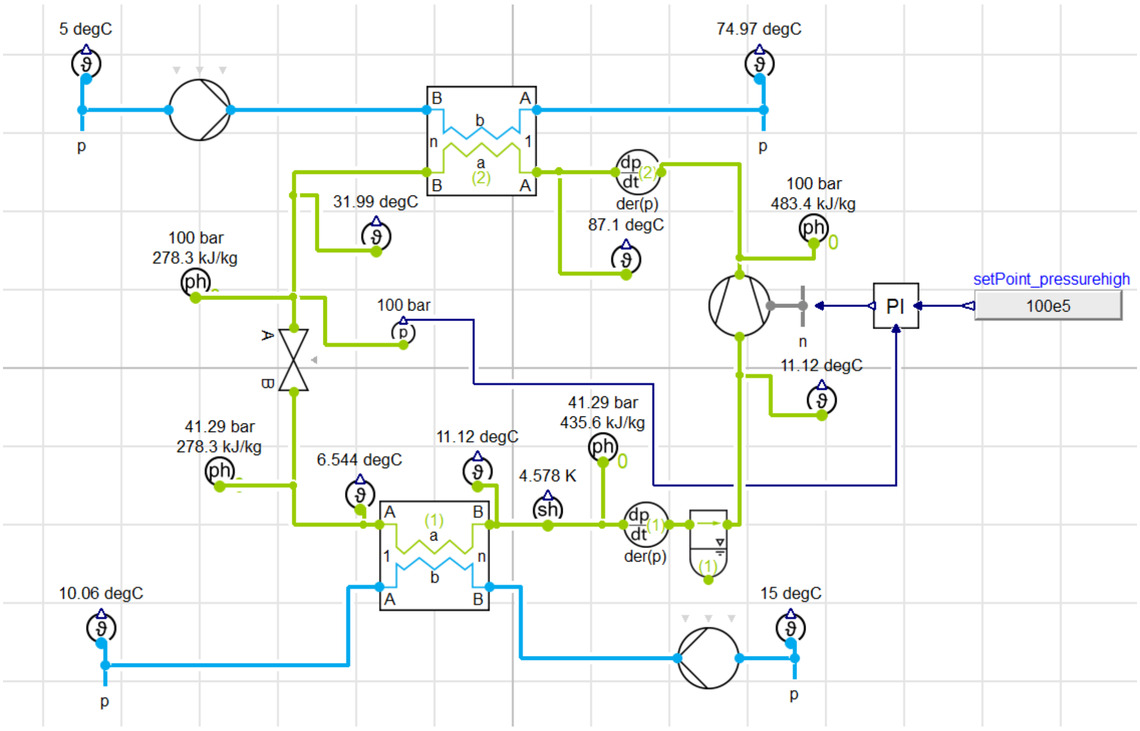

Figure 95: Simple CO_2 heat pump with superheat

In figure 94 the pink line represents the cold city water with a constant temperature of 5,0°C, while the green line represents the heated city water of 74,9°C. The red line represents the cooled CO_2 with a temperature of 32,0°C, while the blue line represents the hot CO_2 with a temperature of 87,1°C. Figure 96 shows the temperatures within the evaporator. We assume an inlet temperature of water of 15°C coming from the outlet of heat exchanger for space heating.

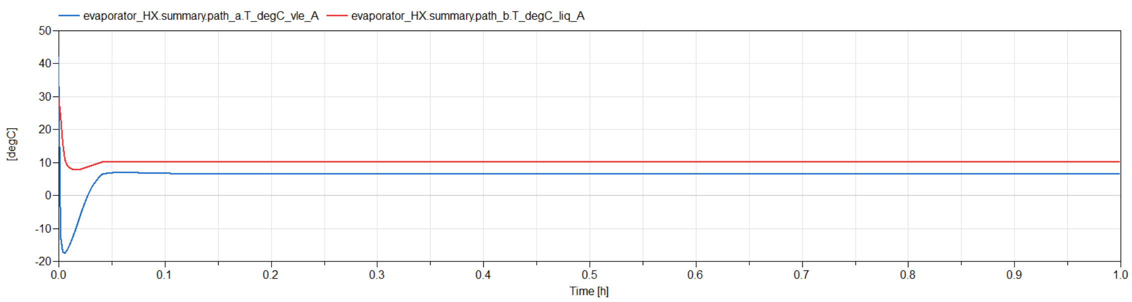

Figure 96: Simple CO_2 heat pump with superheat - Evaporator

In figure 96 the red line represents the water at the outlet of the evaporator with a temperature of 10,1°C. The blue line represents the inlet temperature of CO_2 which is 6,5°C and the outlet temperature is 11,0°C. The pressure on the low and high side is equal to 41,3 bar and 100 bar and the graph of the pressures is shown in figure 97.

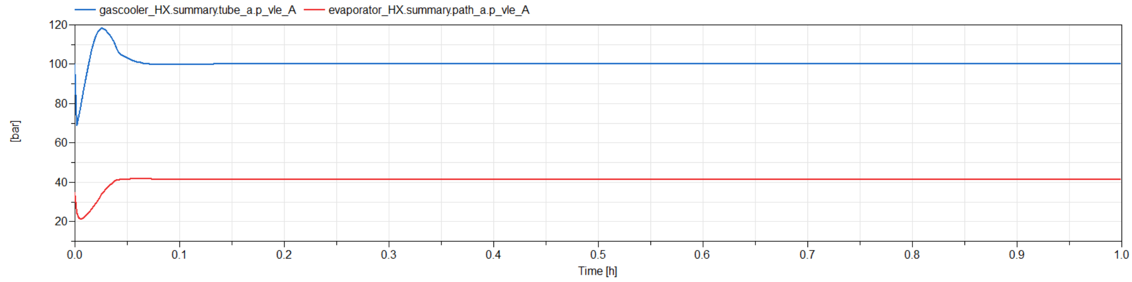

Figure 97: Simple CO_2 heat pump with superheat - pressures

The heat pump cycle made in DaVE is shown in figure 98. When using equation 2 from chapter 2.2.1 the enthalpy values of this cycle gives a COP of 3,287. The compressor have a work of 47,84 kJ/kg. The heat pump in cooling mode will have a COP of 3,287.

$$COP_{heating} = \frac{483,44 - 278,33}{483,44 - 435,60} = 4,287$$

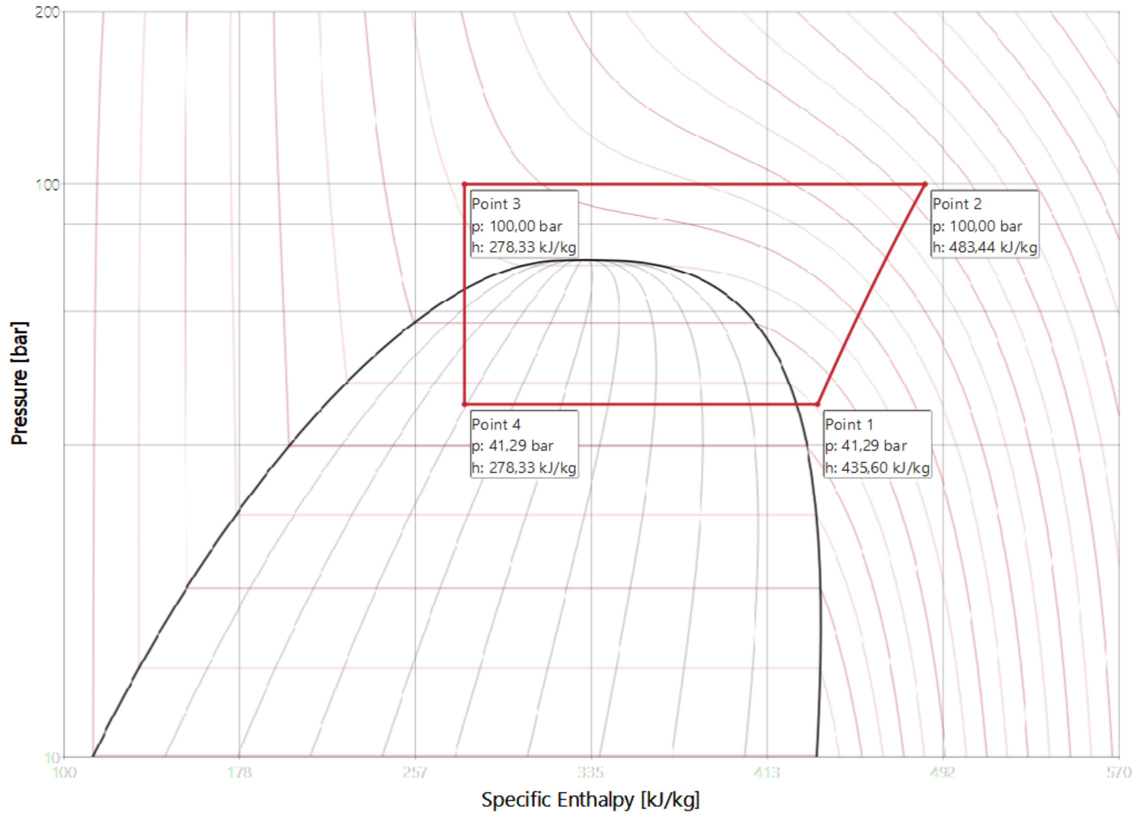

Figure 98: Simple CO_2 heat pump with superheat - heat pump cycle

The PI-regulator has the same values as for the model without internal heat exchanger. The signal of the PI-regulator is shown i figure 99.

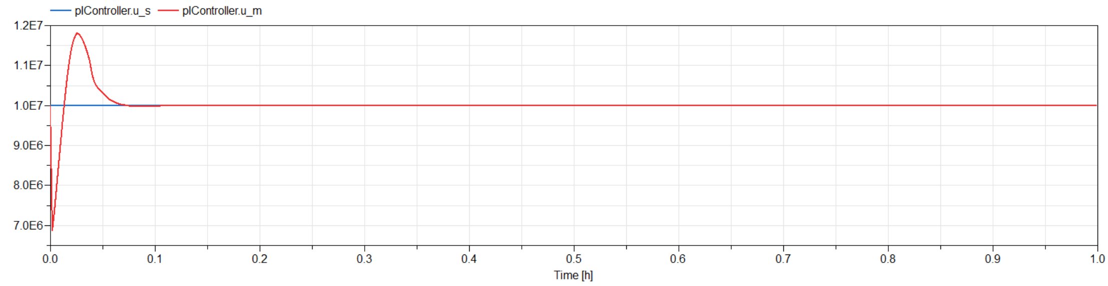

Figure 99: Simple CO_2 heat pump with superheat - PI-regulator

The PI-regulator regulates the behavior of the compressor. As shown in figure 100 the compressor will stabilize itself at a stable level of 33,75 Hz.

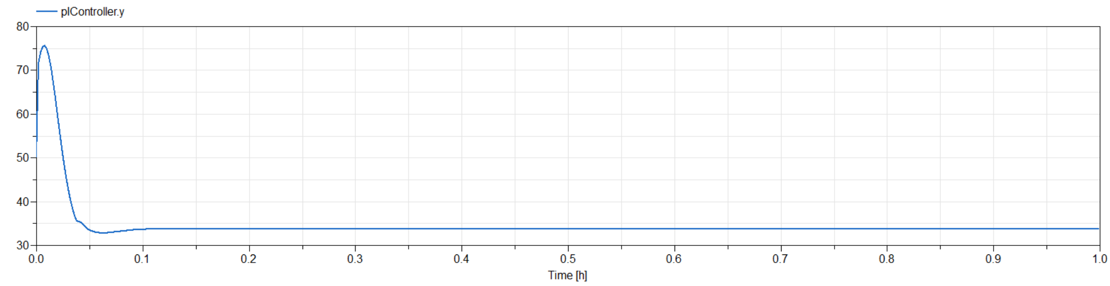

Figure 100: Simple CO_2 heat pump with superheat - y-signal of compressor

Heat pump with internal heat exchanger and superheat
The implementation of an internal heat exchanger is shown in figure 101. This implementation will contribute to a lower temperature level of the CO_2 at the outlet of the gas cooler and a higher temperature into the compressor.

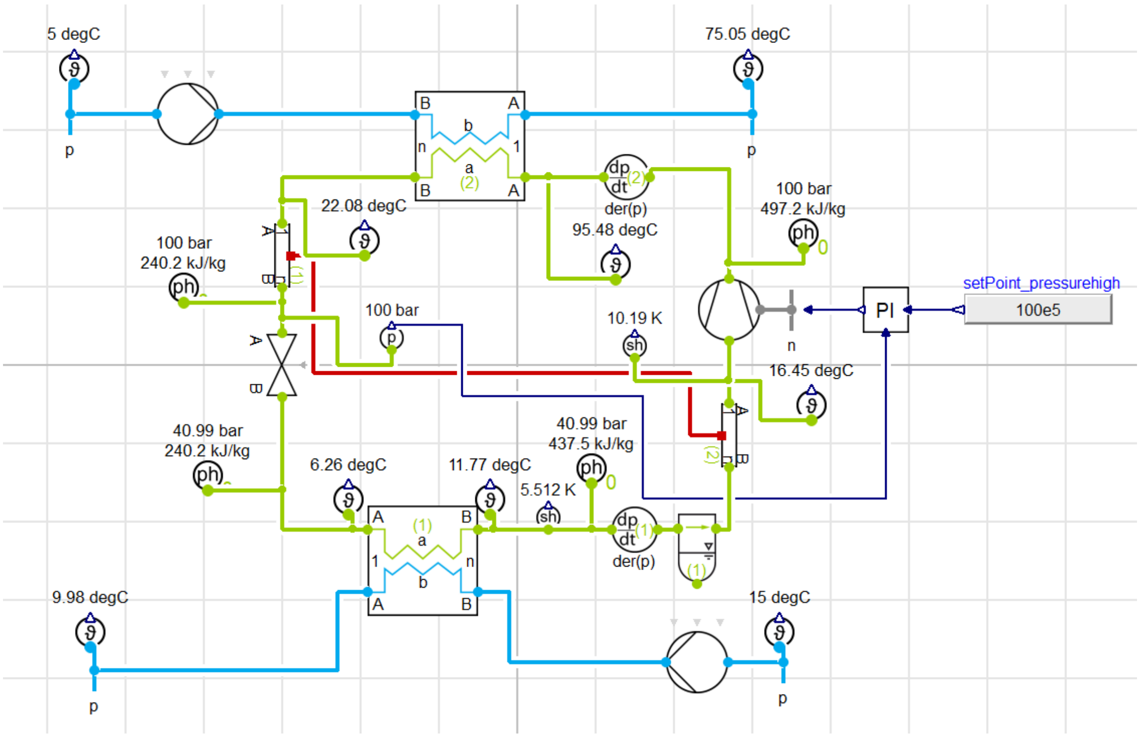

Figure 101: Model of heat pump with internal heat exchanger and superheat

Hot water is flowing from the ice skating rink to the heat exchanger for space heating is set to 0.265 kg/s. The water outlet temperature of the evaporator have a temperature of about 10,0°C and goes into the return line of the low-temperature district heating system. The mass flow of cold city water is still the same and set to 0.024kg/s.

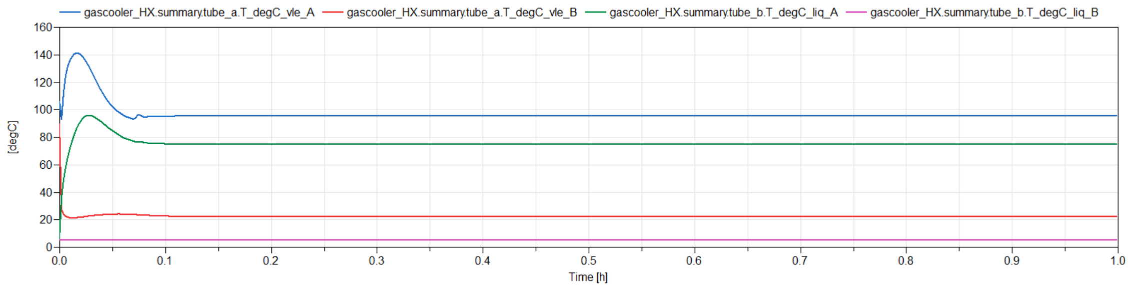

Figure 102: CO_2 heat pump with internal heat exchanger and superheat - Gas cooler

The size of the valve is still 0.28e-6m^2 and the volume of the liquid separator is the same and set to 1.85e-3m^3. In figure 102 the pink line represents the cold city water with a temperature of 5°C, while the green line represents the heated city water of 75,1°C. The red line represents the cooled CO_2 with a temperature of 22,1°C, and the blue line represents the hot CO_2 with a temperature of 95,5°C. Figure 103 shows the temperatures within the evaporator. We assume an inlet temperature of water of 35°C coming from the outlet of heat exchanger for space heating.

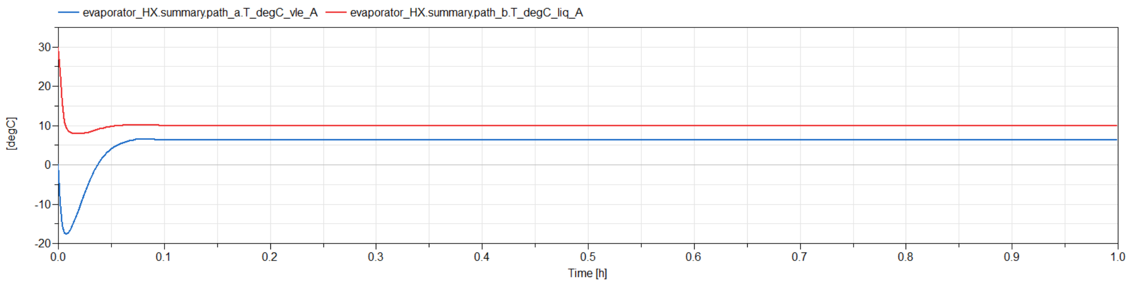

Figure 103: CO_2 heat pump with internal heat exchanger and superheat - Evaporator

In figure 103 the red line represents the water at the outlet of the evaporator with a temperature of 10,0°C. The blue line represents the inlet temperature of CO_2 which is 6,3°C and the outlet temperature is 11,8°C. The pressure on the low and high side is equal to 41,0 bar and 100 bar and the graph of the pressures is shown in figure 104.

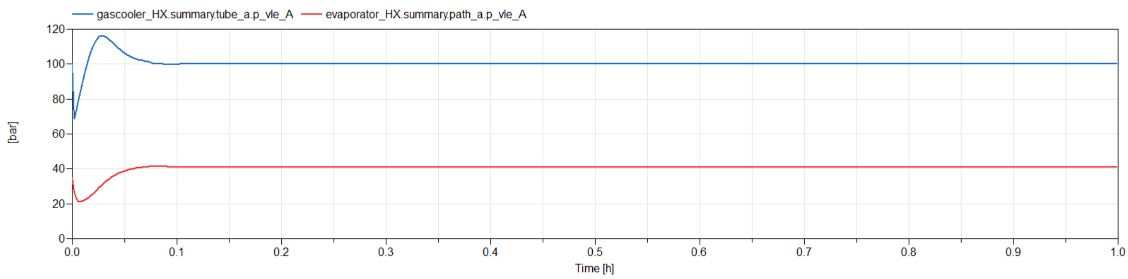

Figure 104: CO_2 heat pump with internal heat exchanger and superheat - pressures

The heat pump cycle made in DaVE is shown in figure 106. The compressor have a work of 59,65 kJ/kg. The COP of the cooling mode will have a value of 3,308 . The PI-regulator has the same values as for the model without internal heat exchanger. The signal of the

PI-regulator is shown i figure 105.

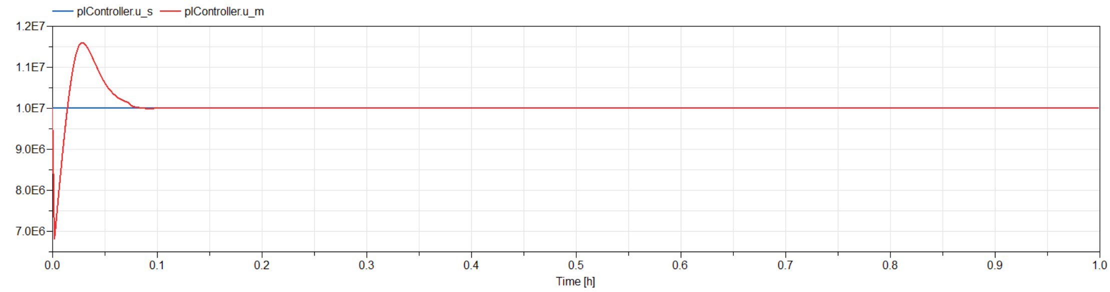

Figure 105: CO_2 heat pump with internal heat exchanger and superheat - PI-regulator

$$COP_{heating} = \frac{497,15 - 240,16}{497,15 - 437,50} = 4,308$$

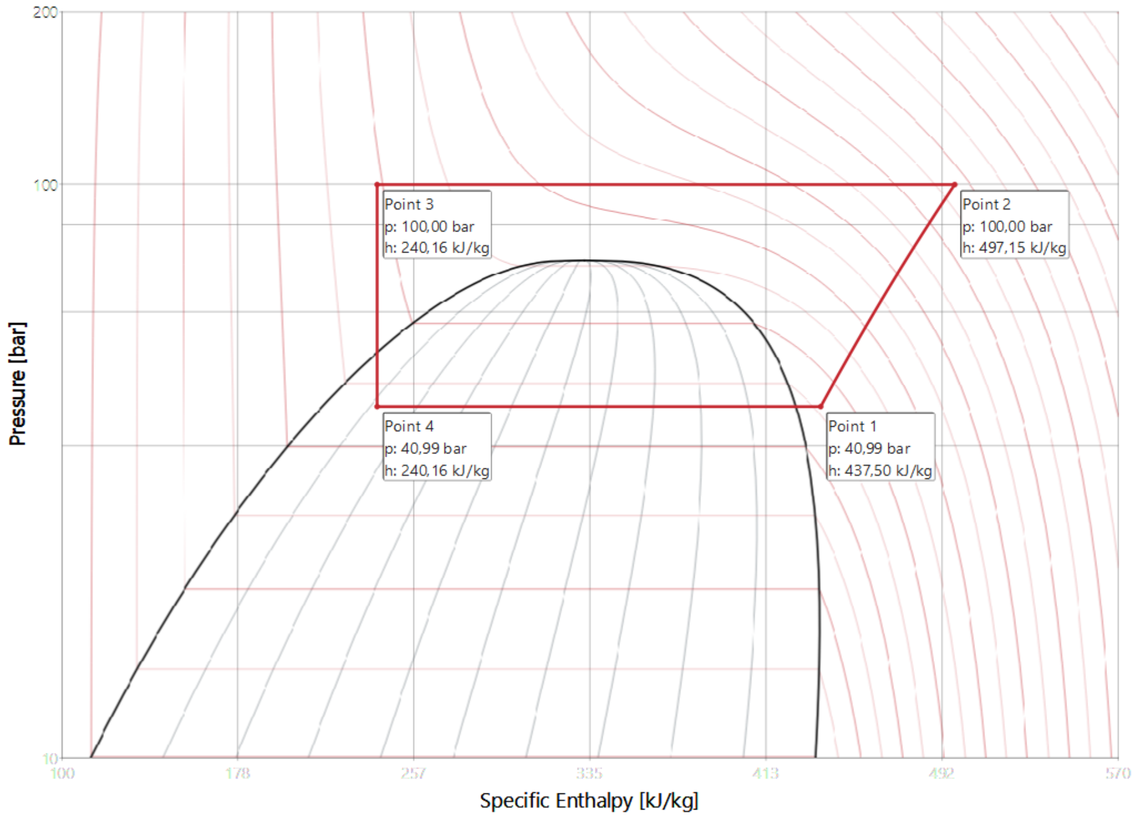

Figure 106: CO_2 heat pump with internal heat exchanger and superheat - heat pump cycle

The PI-regulator regulates the behavior of the compressor. As shown in figure 107 the compressor will stabilize itself at a stable level of 29,98 Hz. This model have a lower temperature of the CO_2 coming out of the gas cooler and a higher temperature coming into the compressor. This model have a higher temperature going into the return line of the low-temperature district heating system.

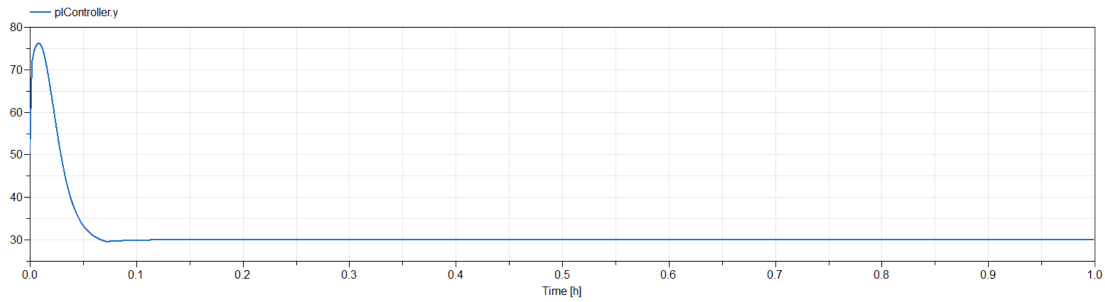

Figure 107: CO_2 heat pump with internal heat exchanger and superheat - y-signal of compressor

4.5.5 Space heating and cooling

By using table 15 it is possible to calculate the overall heat coefficient and the mass flows of water. The calculations is presented in table 39. The capacities is based on the highest specific capacity values of space heating from the user profile which is 15,332 W/m^2. This results in a capacity of 21,464 kW for space heating. The capacity for cooling is however difficult to determine. It exists little information about specific cooling demand for passive house buildings for inhabitants. For the evaluation of cooling demand data from Simien have been used to make a relationship between the heating and cooling demand simulated. We know that the specific value for space heating is 28,1 kWh/m^2 and for cooling the value is 10,6kWh/m^2 We use this relationship to determine the specific effect given in W/m^2 for our reference building. The result will be a specific capacity of 5,784 W/m^2 of the heat exchanger for space cooling which for our reference building is 8,097 kW. Now it is possible to look at the models in Dymola, and plate heat exchangers is being used.

$$Q_{cooling} = \frac{10,6kWh/m^2}{28,1kWh/m^2} * 15,332W/m^2 = 5,784W/m^2$$

Table 39: Calculated values for heat exchangers for heating and space cooling

Parameter	space heating	space cooling
Q [W]	21464	8097
Cp_{water} [J/kgK]	4200	4200
Δ T_1 [K]	5,00	10,00
Δ T_2 [K]	10,00	5,00
Δ T_{lm} [K]	19,427	2,787
UA [W/K]	1104,857	2905,770
Mass flow $water_1$ [kg/s]	1,022	0,386
Mass flow $water_2$ [kg/s]	0,511	0,193

The temperature of the water which is being heated or cooled inside the building is set to 25°C. This is however not a correct value for the whole year. The ambient temperature outside the building will influence this temperature and decide the energy input required. The geometry outside and inside of the heat exchanger for heating and cooling is presented in table 40. This geometry will be changed to see how the geometry influences the heating and cooling process.

Table 40: Geometry for heat exchangers for heating and space cooling

Parameter	space heating	space cooling
Number of plates	50	50
Plate length [m]	0,3	0,3
Plate width [m]	0,1	0,1
Pattern angle in deg	35	35
Wall thickness [m]	0.75e-3	0.75e-3
Pattern amplitude [m]	2e-3	2e-3
Pattern wave length [m]	12.6e-3	12.6e-3
nCells	20	20

Space heating
The heat exchanger for heating is shown in figure 108. The temperature sensors indicates the initial temperature values for the sensors, and is not the actual temperature in the system. In figure 109 the blue line constitutes the hot water from the waste heat source, while the red line constitutes the cold water in the building being heated. The graph indicates that the mass flows based on theory can not achieve the desired temperature for space heating. Due to this issue, we need to change the values of UA or the mass flows of water.

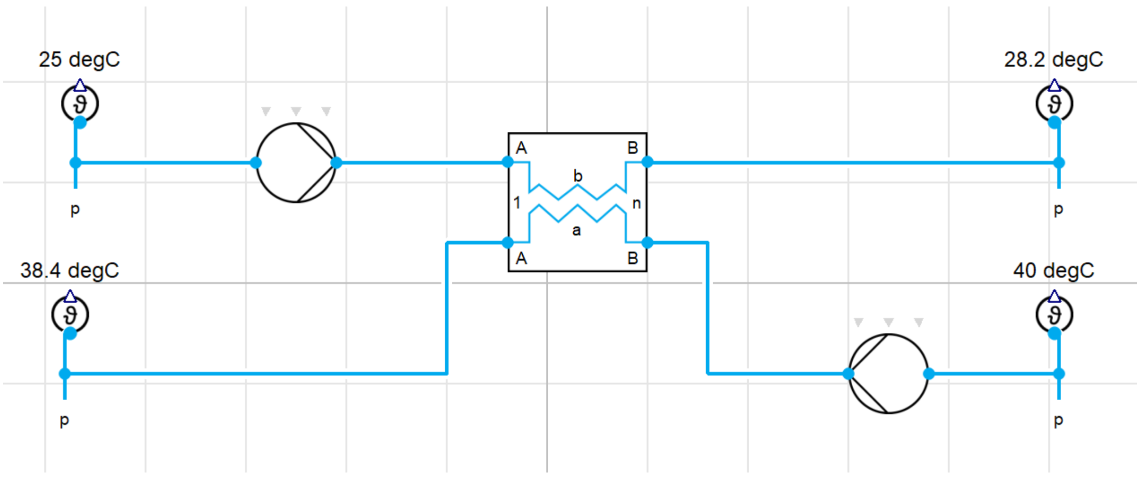

Figure 108: Heat exchanger for space heating made in Dymola

We only cool the waste heat source from 40°C to about 38,4°C, while the cold water in the building is heated from 25°C to about 28,2°C as shown in figure 109. Water flow 1 (lower flow) and 2 (upper flow) was set to the theoretical values of 1,022 kg/s and 0,511 kg/s. The UA value was set to 1104 W/k.

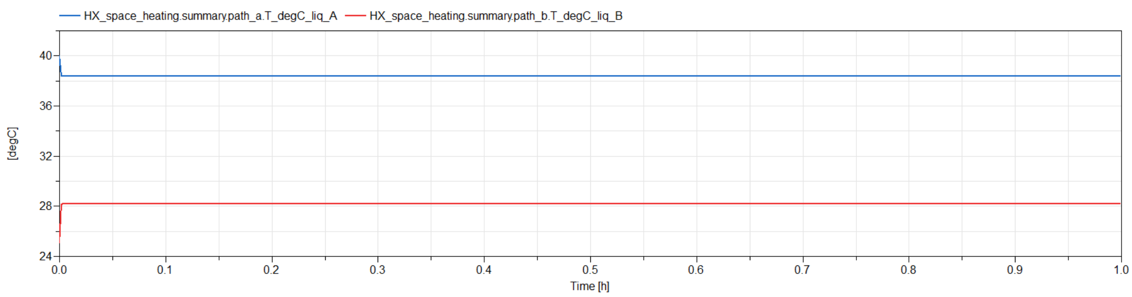

Figure 109: Temperature changes inside heat exchanger for heating

To achieve the desired temperature for space heating we want to increase the UA value and/or lower the mass flow of water to absorb more heat. Data from one year of a user profile for energy demand of both space heating and production of domestic hot water was given from Sintef Energi. The user profile was based on a passive building given as specific energy demand which varies from the values of 0 to about 15 W/m^2 which is 5 W/m^2 more than for the theoretic values. The file implementation to the Dymola model is shown in figure 110. The temperature of the building will not be 25°C all the time, and will depend on the ambient temperature outside, and at the day of the year with the highest demand, we have no information about the water temperature of the building.

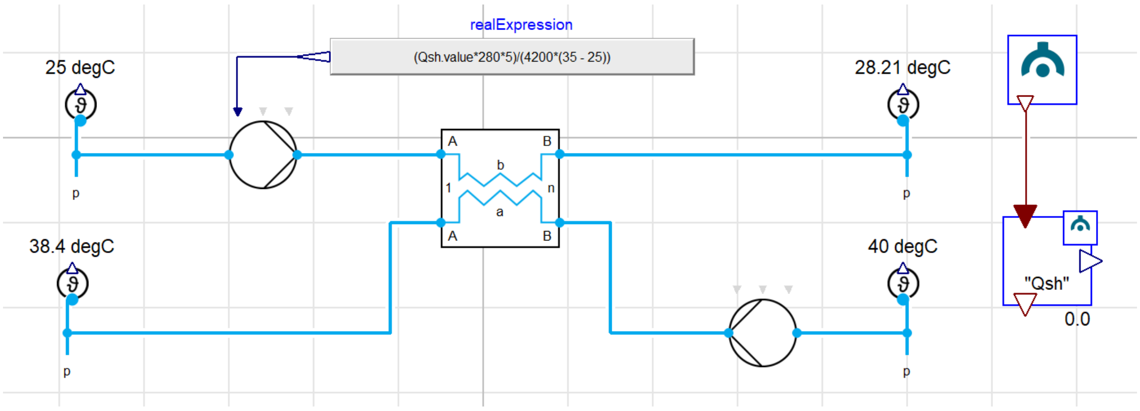

Figure 110: User profile for space heating

It is not of interest to investigate the model further due to lack of information. However, it is of interest to look at how the heating demand changes during one year since the changes is quite big when comparing the different months. A simulation of the demand for one year is shown in figure 111

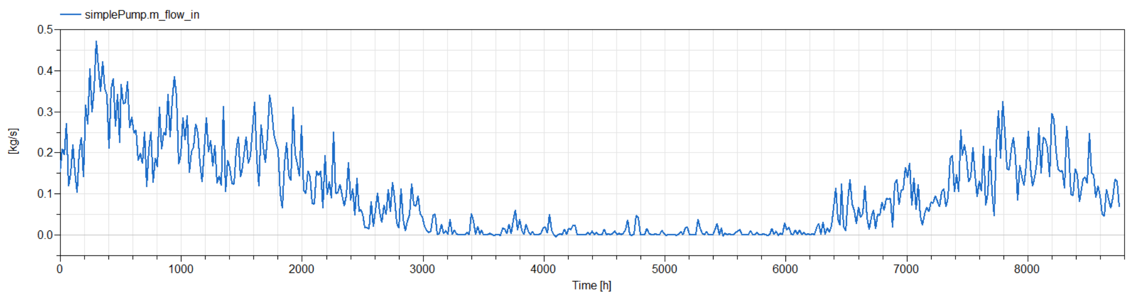

Figure 111: User profile for space heating given in mass flow

The reference building simulated in Simien also visualize that passive house buildings do not need space heating during several months in a year. As figure 111 indicates, the real demand of this user profile is very low from the end of April to the beginning of September. This is matching with the heating demand for our Simien reference building.

Space cooling

The heat exchanger for space cooling is shown in figure 112. The graph of the temperatures in this heat exchanger is shown in figure 113. The red line constitutes the cold water from the evaporator outlet, while the blue line constitutes the hot water in the building being cooled.

It is assumed a cooling demand of 5,784 W/m^2 which is equal to 8,097 kW for our Simien reference building. We need a mass flow of 0,386 kg/s on the ice skating rink water side, and a mass flow of 0,193 kg/s on the building side to achieve the required demand. The UA value is set to 2905 W/K.

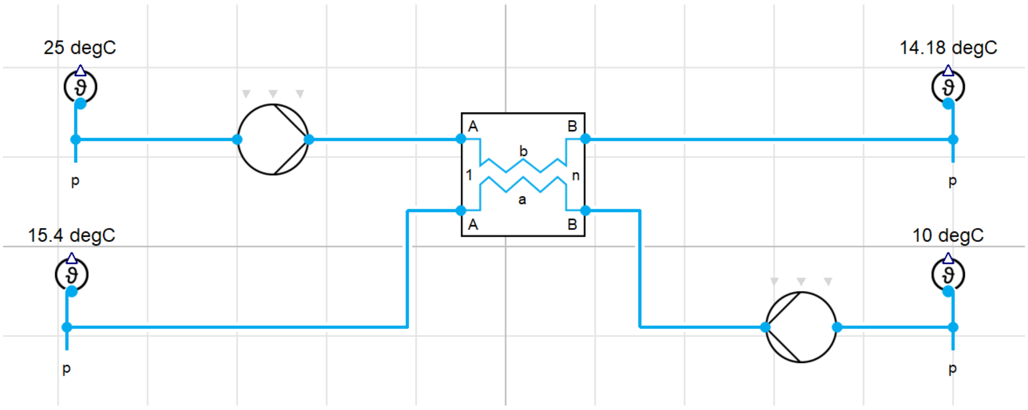

Figure 112: Heat exchanger for space cooling made in Dymola

Figure 113 indicates that the mass flows based on theory can not achieve the desired temperature for space cooling. The cold water from the outlet of the evaporator have a temperature of 10°C and heats up to about 15,4°C, while the hot building water only cools down from 25°C to about 14,2°C which is too cold. Due to the data lack of temperature of building water it is not interesting to investigate this model further.

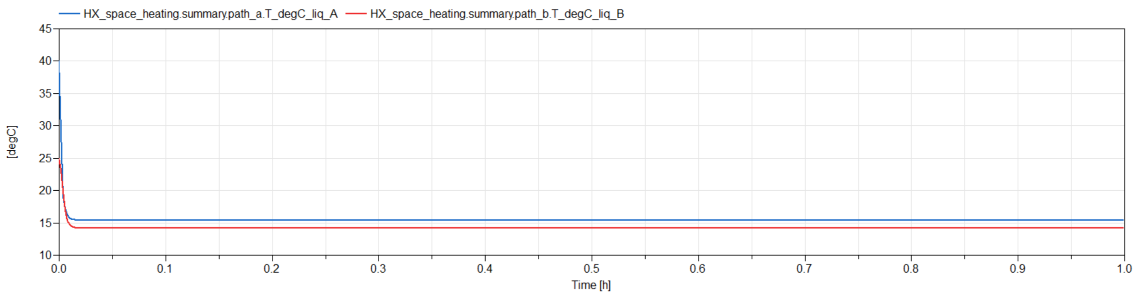

Figure 113: Temperature changes inside heat exchanger for space cooling

4.5.6 Use of high temperatures in storage tanks

We use values for the Simien building to show why high temperatures for the domestic hot water tanks is a good idea. We do know that the demand for one building for DHW is 41711 kWh which is equal to 4,762 kW. We assume constant production 24 hours which is equal to 86400 seconds, and the density of water is 1000 kg/m^3. The different temperatures investigated is shown in table 41.

Table 41: Storage tanks and different temperatures

Inlet temperature [°C]	Water temperature [°C]	Mass flow [kg/s]	Volume [m^3]
5	55	0,023	1,959
5	60	0,021	1,781
5	65	0,019	1,633
5	70	0,017	1,507
5	75	0,016	1,399
5	80	0,015	1,306
5	85	0,014	1,225

Figure 114 shows how the volume changes when the temperature increases. It is noticeable that if the water temperature in the tank is being increased, the size of the storage tank will be reduced. A water tank with 75°C will have 14,3% reduction in volume compared to a tank with a temperature of 65°C which is 10°C difference. A difference of 20°C from 60°C to 80°C in the storage tank will be a reduction of 26,7% of the volume with lower temperature.

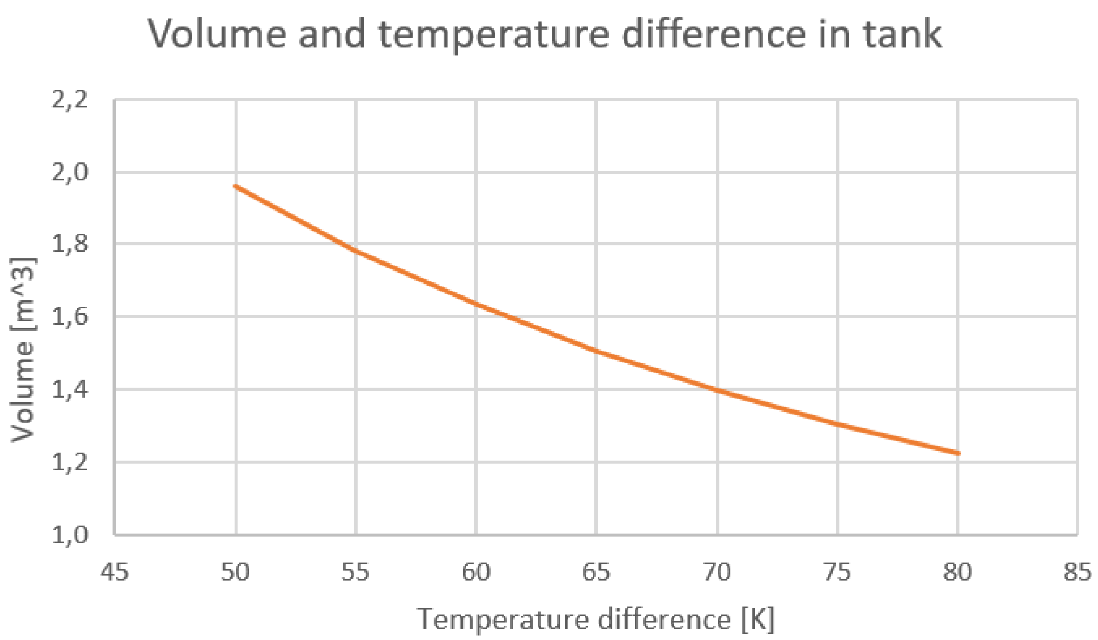

Figure 114: Different temperatures in storage tanks

4.5.7 Design of DHW storage tanks

We want to investigate how the DHW storage tanks can be sized to cover the energy demand for hot tap water. We do have a user profile for a passive house building available which is the most realistic case to look into. However, we also want to investigate our demand of the Simien file with theoretic values as well as looking at values from reality and experiences. In this way we gain results of measured, theoretic and real values.

User profile demand
We do have a overview of how the demand changes during a day. The demand for domestic hot water varies a lot with several peak hours, but has a quite regular pattern from day to day. It is desirable with a production of domestic hot water for 24 hours a day, which indicates that the production must be constant. The day with the highest demand must be considered when designing the accumulation tank. When finding the highest demand the MAX function in excel is being used. In this way it is possible to see that the highest value appears after 72 hours. We investigate the hours of 72-96 of the year in Dymola. The highest demand will be 4,222W/m^2, which for the reference building will be 5,91kW. The red line in figure 115 shows the energy demand of the 4th of January in which the highest demand appears. The demand is given in m^3 per hour as shown in figure 115. The black line represents a constant production of DHW and gives us the equation equal to the area. The average mass flow will be 0,0168 kg/s.

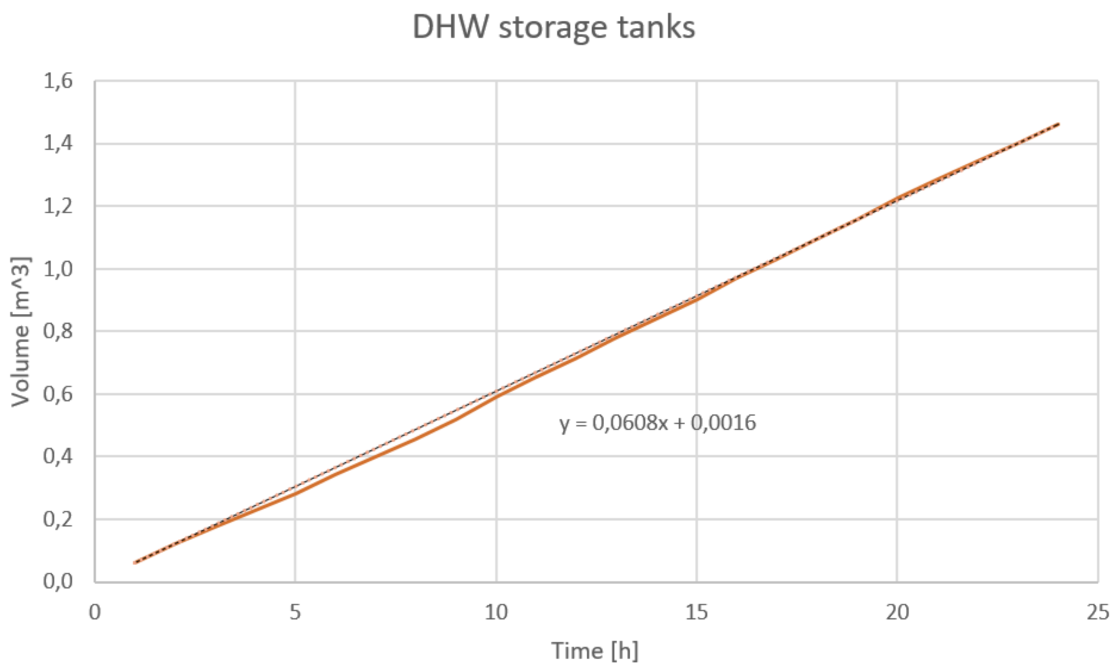

Figure 115: Design of storage tank - user profile

After 24 hours, there will be used 1,460m^3 which is equal to 1460L. The storage tanks should be able to cover this demand.

Theoretic calculations
We also want to investigate the Simien file. We do not have any information on how the demand changes during the day, however we do have the yearly amount of demand which is 41711 kWh for one building. For passive house buildings the DHW demand is 5,1W/m^2 which in our case is 7140W. When heating up water from 5 to 75 degrees this will result in a mass flow of 0,024 kg/s. The production curve is presented in figure 116. We need a tank of 2098L or 2,10m^3 to cover this demand.

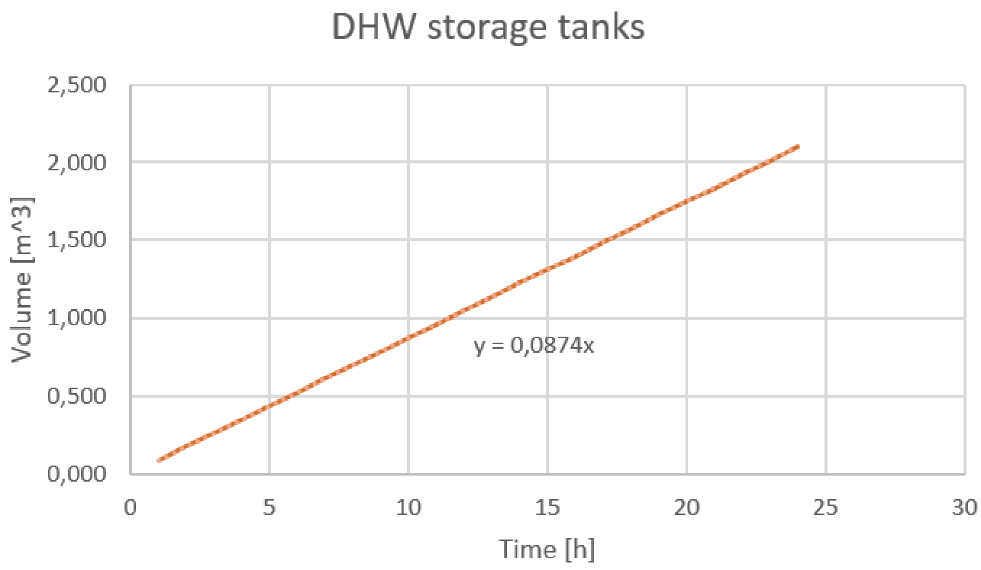

Figure 116: Design of storage tank - theoretic values

Calculations based on experience
According to information given by Sintef an apartment of 70m^2 will have a demand of 100L per day, but it varies from 80-120L. To maintain safety the tanks should be designed according to a demand of 150L.

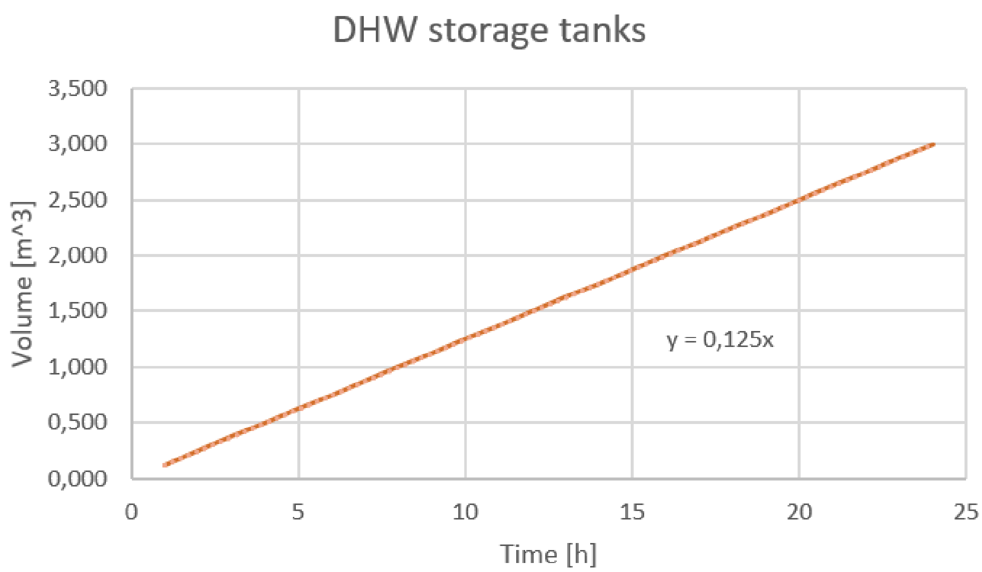

Figure 117: Design of storage tank - values from reality

The information given was based on an inlet temperature of 10°C and 60°C out. This makes a temperature difference of 50°C. We do operate with 20 apartments of 70m^2 which will give a demand of 3000L or 3m^3. A constant production during the day will be 0,125m^3. A constant production of domestic hot water will in this case be as shown in figure 117. We collect the three different approaches in one graph shown in figure 118 in which the theoretic values of what is needed is way bigger than simulated and measured values. As we can see the theoretic values is way bigger than for the simulated and measured ones.

The measures ones from the user profile is however the most reliable source since we know when the day with the highest demand is and can use the values from this day to design a storage tank for the highest demand. The theoretic values have a temperature difference of 50K in the water while the user profile and Simien graph have a temperature difference of 70K. As mentioned in chapter 4.7.6 a tank with a temperature difference of 50K will have a bigger tank volume compared to a temperature difference of 70K.

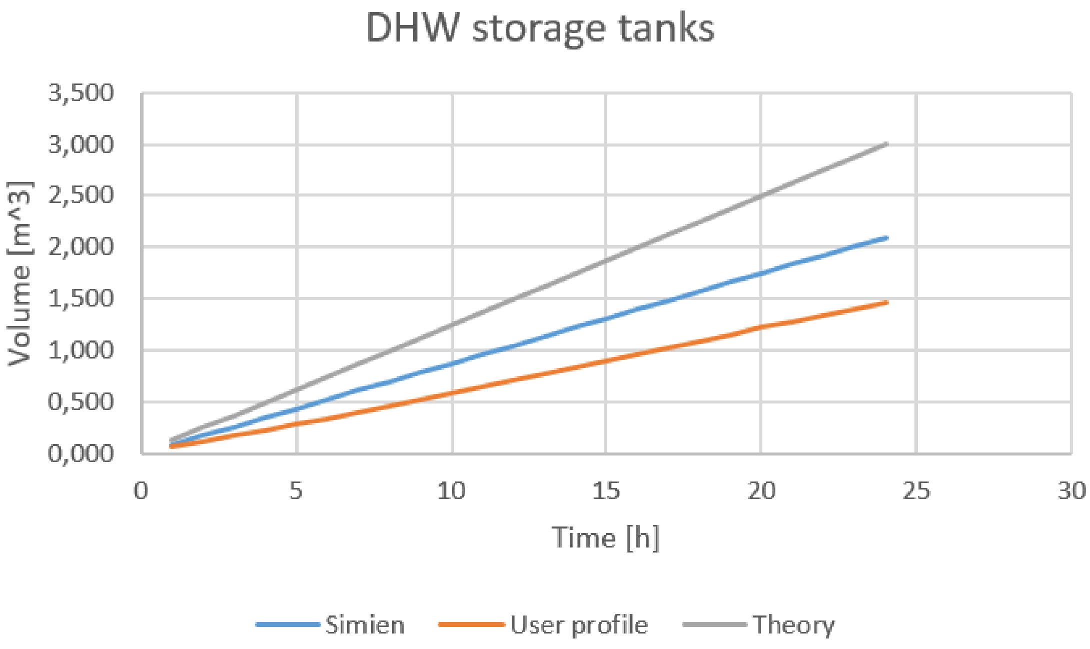

Figure 118: Design of storage tank - user profile, theoretic values and values based on reality

4.6 Energy budget for one building

This chapter will include information about the system solution for a smart, thermal grid at Leangen. It is of interest to analyze a system which utilizes the available external waste heat from the ice skating rink and the internal waste heat of greywater inside the building. We want to make an illustration of how different energy sources covers the energy demand of one building at Leangen.

The system solution utilizes waste heat sources for space heating and further on the waste heat coming from the outlet of the heat exchanger for space heating will be reused as a heat source for a CO_2 heat pump producing DHW as explained in Model 2. In case of such a solution, there must be applied some extra heat coming from the main district heating distribution system as a safety. Some of the months have too much waste heat from the ice skating rink while others have too little. Table 42 presents the energy budget for each month with an energy balance. Positive numbers represents the amount of energy which is needed, while negative numbers represents the case in which there exists too much waste heat. We assume we have a heat pump with a COP of 4.

In the column "Buy DH [kWh]" the months which have negative numbers do have too much waste heat. Figure 119 presents the energy budget of a conventional and new system. A conventional system needs a yearly amount of 81024 kWh per building coming from a high-temperature district heating system.

Table 42: Energy budget for Simien building

Month	Total waste heat [kWh]	HP heat demand [kWh]	Buy EL [kWh]	Buy DH [kWh]
January	7591	2551	850	2069
February	7450	2438	813	597
March	6670	2551	850	815
April	4634	2551	850	225
May	4159	2776	925	-575
June	3901	2663	888	-1237
July	2273	2663	888	390
August	5403	2776	925	-2627
September	5285	2551	850	-1326
October	5084	2551	850	1275
November	11516	2551	850	-3206
December	11014	2663	888	-785

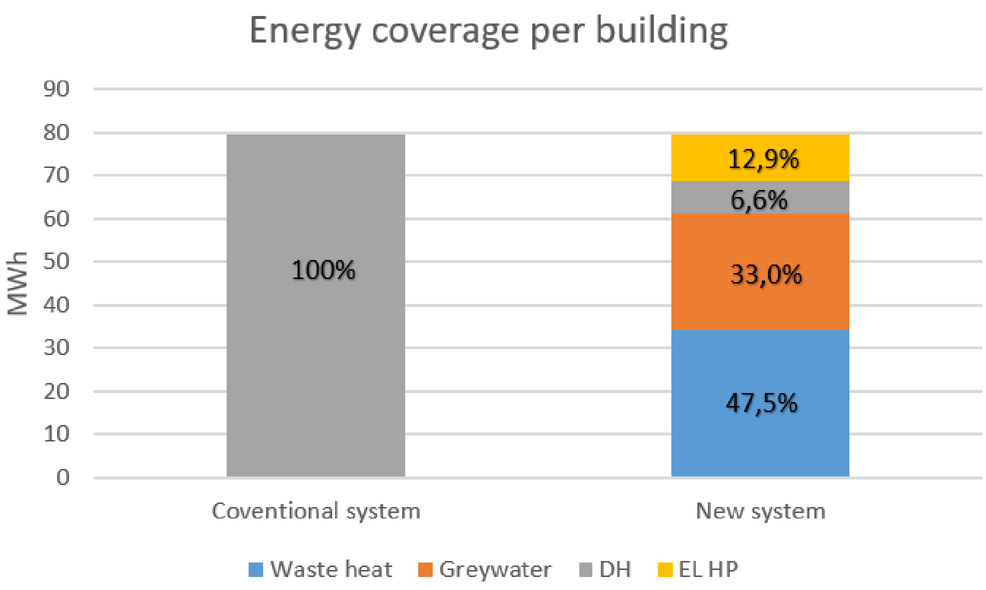

Figure 119: Conventional system vs new system

The total amount of waste heat from the ice skating rink and greywater which is available for use during a year for one building is 65225 kWh. The ice skating rink and the greywater separate will constitute 38530 kWh and 26695 kWh. The new low-temperature distribution system will require energy from the high temperature district heating system of an amount of 5372 kWh for one building. This information indicates that by implementing the new system solution, the waste heat from the ice skating rink which is available for usage during each month will cover 47,5% of the total energy demand. By utilizing greywater for harnessing this will represent a percentage of 33,0% of the total energy demand, and energy which have to be bought for the compressor of the heat pump will constitute 12,9%. The remaining amount of energy which should be coming from the Statkraft Varme and the main district heating system is 6,6%.

4.7 Cost analysis of system

We use the Nordic spot price in table **??** and analyses the use of low-temperature district heating and cooling instead of electricity, as well as using a heat pump for heating of DHW instead of electricity.

4.7.1 Electricity versus district heating

According to the the key account manager for Marked in Statkraft, Terje Berg, the prices of district heating is nearly the same as for the electricity. StatKraft do have the same tariffs for district heating as for the power grid rents in the same area. However, for district heating it does not exist a regular amount neither an ENOVA fee for private households. Due to this fact, it is hard to acknowledge if there will be a cost reduction potential for customers by using low-temperature district heating for space heating instead of electricity. Table **??** refers to the Nordic spot prices of electricity found in an annual report of 2017 by Nord Pool. At the date of 21.04.19 the value of the euro was set to 9,57 NOK. There will be a energy saving potential in general by using waste heat which would have disappeared into the ambient. Table 43 shows the cost saving potential of utilizing waste heat when the market prices is at its lowest and highest according to the energy demand of the heat pump.

Table 43: Different prices of electricity and cost saving potential

Year	Waste heat [kWh]	Electricity price [NOK/kWh]	Cost saving potential [NOK]
2015	4152492	0,211	876176
2010	4152492	0,507	2105313

The energy saving potential is big, and the condensers of the ammonia refrigeration system generates in total throughout the year an amount of 4152492 kWh. This is energy which today is being fed into the ambient air. The prices of electricity varies a lot. If we consider the lowest annual price and the highest annual price of electricity of the period of 2007-2017 we get lower and higher numbers. We consider the scenario with the lowest and highest prices which is the year of 22 EUR/MWh in 2015 and 53 EUR/MWh in 2010. According to Nord Pools data of annual electricity prices [97], the waste heat coming from the ice skating rink have a possible value of between 876176-2105313 NOK.

4.7.2 Implementation of greywater

As table X shows, the total yearly amount of greywater will be 2295773 kWh. We use the low and high spot prices to look at the amount of money which can be spared. We use the values for low and high prices shown in table 43 and make table 44. We get a cost saving potential of between 484 408 - 1 163 957 NOK per year.

Table 44: Cost saving potential by implementing greywater solution

Electricity price [NOK/kWh]	Available greywater [kWh]	Cost saving potential [NOK]
0,211	2295773	484408
0,507	2295773	1163957

4.7.3 Production of domestic hot water

By implementing a heat pump for production of tap water, it is possible to reduce the operational costs. This is shown in 45 and 46. In this table the prices for different COP of the heat pump is listed. The calculations is based on a yearly energy demand for domestic hot water which is 41711kWh for the reference building, and the calculations is based on the yearly prices of electricity for the cheapest and most expensive price of electricity given in the period of 2007-2017. The reduction of price is the yearly cost reduction. The cost saving potential when considering the cheapest average annual price is shown in table 45. One euro is set to 9,57 NOK at the date of 21.04.19.

Table 45: Cost saving potential with heat pump when price is low

COP	EL demand [kWh]	EL for HP [kWh]	Energy difference	Price reduction [NOK]
1	41711	41711	0	0
2	41711	20856	20856	4391
3	41711	13904	27807	5855
4	41711	10428	31283	6586
5	41711	8342	33369	7025
6	41711	6952	34759	7318
7	41711	5959	35752	7527
8	41711	5214	36497	7684

The relationship between increasing COP value is shown in figure 120 when considering the cheapest price of 22 EUR/MWh in 2015. It is noticeable that by gaining a COP of the heat pump of about 2-3 will have the biggest increase of cost savings, and that from a COP of 4 and further the increase is not as abrupt.

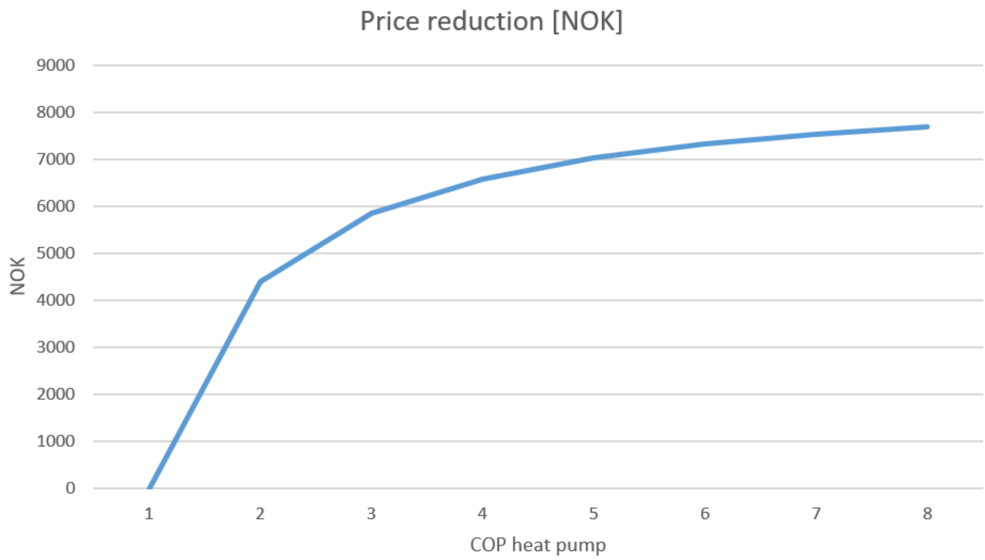

Figure 120: Price reduction with increasing COP when price is low

Table 46 shows the annual cost saving potential by implementing a heat pump when the prices of the market is high.

Table 46: Cost saving potential with heat pump when price is high

COP	EL demand [kWh]	EL for HP [kWh]	Energy difference	Price reduction [NOK]
1	41711	41711	0	0
2	41711	20856	20856	10578
3	41711	13904	27807	14104
4	41711	10428	31283	15867
5	41711	8342	33369	16925
6	41711	6952	34759	17630
7	41711	5959	35752	18134
8	41711	5214	36497	18512

The relationship between increasing COP value is shown in figure 121 when considering the highest price of 53 EUR/MWh in 2010. It is noticeable that by gaining a COP of the heat pump of about 2-3 will have the biggest increase of cost savings, and that from a COP of 4-5 and further the increase is not as abrupt.

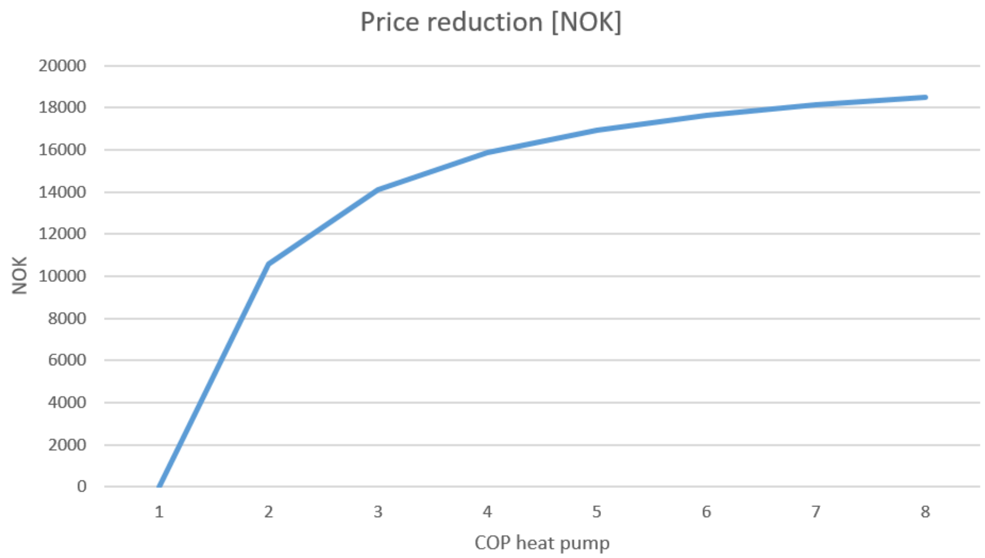

Figure 121: Price reduction with increasing COP when price is high

When the building is being heated, our model of the CO_2 heat pump will have a COP of about 3,7 for the heat pump without superheat which results in a yearly saving potential of between 6408-15438 NOK for one heat pump unit and 551119-1327697 NOK for 86 buildings. When we allow superheat to happen we have a COP of 5,1 which results in a yearly saving potential of between 7060-17008 NOK for one heat pump unit and 610000-1469544 for 86 buildings.

5 Discussion

This chapter contains several subsections with discussions regarding different topics being evaluated throughout this master thesis. The chapter considers low-temperature thermal grids, the energy demand of the reference building and waste heat available, the system solution and the cost saving potential of implementing a smart, low-temperature thermal grid at Leangen.

5.1 Low-temperature thermal grids

As residential buildings get more energy efficient, the demand of space heating and cooling decreases. A low-consumption or passive house building only needs water temperatures of about 35°C for space heating and 14-15°C for space cooling. Due to the lowered energy demand of heat, hot temperatures is not needed anymore to maintain comfortable temperatures for space heating in buildings. Therefore, a local low-temperature thermal grid of 40°C is able to cover the space heating demand of new buildings. A water temperature of 15°C coming from the outlet of an evaporator of a heat pump can cover the space cooling demand. Lower temperatures in the thermal grid will reduce the heat losses in the network and waste heat sources with low temperatures in our society can be utilized. The difficulties that arise when implementing a low-temperature thermal grid is the thermal treatment of DHW. It is desirable with temperatures over 65°C to prevent the legionella bacteria.

5.2 Energy demand and waste heat utilization

Space heating demand and DHW demand of the reference building simulated in Simien is 28,1kWh/m^2 and 29,8kWh/m^2 compared to the user profile with the values of 26,2kWh/m^2 and 30,6kWh/m^2. These specific values confirms that the energy demand of the reference building is nearly equal to the passive house user profile. The cooling demand for the reference building was 10,6kWh/m^2 while it is unknown for the user profile. The reference building is categorized as a low-energy consumption building with a heating demand of under 30kWh/m^2. The reference building utilizes waste heat sources such as external waste heat from the ice skating rink and internal waste heat in the form of greywater. The waste heat from greywater can be stored in storage tanks in under each building or be sent to centralized storages at the area. The waste heat from the ice skating rink together with waste heat in the form of greywater can cover 47,5% and 33,0% of the total energy demand for space heating and production of DHW for one building. Together this will constitute a percentage of 80,5%. Electricity for the heat pump is needed and constitutes 12,9%, while 6,6% of the energy demand should come from another energy source. Additional energy input to the low-temperature thermal grid as the area is constructed today could be the high-temperature district heating network.

5.3 System solution

In model 2 the low-temperature thermal grid will first heat up the space. It does not make any sense to focus only on efficient space heating when DHW production accounts for about 30% of the overall energy use in a building. Therefore the waste heat is further on being reused after space heating as a heat source of a CO_2 heat pump producing DHW. The issue with the legionella bacteria disappears when introducing the heat pump as the water temperature rises from 5-75°C. The heat pump also contributes to space cooling, which can be an important offer for the inhabitants in dense building areas. A cooling solution can prevent people from opening windows and thereby avoid noises outside. The uncertainty in the model appears when investigating the heat exchangers for heating and cooling. It was hard to investigate the heat exchangers further in Dymola without having a user profile which indicates the cooling demand and temperatures of the water on the building side. This temperature was set to 25°C but will vary during the year and is dependant on the ambient temperature outside.

It was desirable to create a simple, cheap system and therefore diurnal thermal storage was chosen above seasonal thermal storage. We only need to implement a heat pump and water tank in a technical room below the building. Underfloor heating of concrete is also a form of thermal energy storage as well and we achieve great thermal comfort by spreading the heat uniformly on the ground instead of only centering it at one place as conventional radiators. A higher temperature in the hot water storage tanks will result in a decrease of the size of the tank. This can be a good thing considering use of space in the technical room of the building. The storage tanks should be sized according to the day of the highest demand in the year. The user profile of the passive house building in Trondheim at the day of the highest demand suggests a volume of the storage tank of 1460L, while the theoretic method suggests that we should have a volume of 2098L. The theoretic case is based on theory about the demand for DHW being 5,10W/m^2. The case based on experience of tanks in general suggests a volume of 3000L. The volume of the passive house user profile is very small compared to the volume based on experience. However, the case based on experience is based on a lower temperature difference in which the inlet and outlet temperature of the water will be 10°C and 60°C and constitutes a temperature difference of 50K instead of 70K. Therefore the tank size based on experience must be bigger than the one for the user profile. We do however not have information about the day with the highest demand considering the theoretic case, and this case is based on an average demand. The storage tank for our solution must be over 1460L and below 3000L.

CO_2 was chosen as the working fluid for the heat pump producing DHW and the medium have a good temperature fit with water when producing DHW. CO_2 is an environmentally friendly working fluid, as well as non-toxic and non-flammable. Ammonia is toxic, and propane and other hydrocarbons is flammable. The only issue which occurs with CO_2 as a working fluid is that in very big concentrations people can choke. When using components which withstand high pressures and thereby adapts better for this working fluid, the efficiency of CO_2 heat pumps is very high and therefore economical competitive. CO_2 heat pumps for production of DHW is available at several companies such as Winns and thereby available for the market. Calculations of the heat pump cycles for ammonia, propane and CO_2 was done in Coolpack. For a heat pump producing DHW with a temperature temperature of 75°C. CO_2 obtained the highest COP value followed by ammonia and propane. The COP of the working fluids will increase with increasing value of evaporation temperature. In coolpack, we get a lot more information when investigating the transcritical cycle of CO_2 compared to the one-stage cycle of ammonia and propane. For CO_2 it was possible to investigate the outlet temperature of CO_2 coming from the gas cooler as well as different pressures. The highest value of COP for the CO_2 heat pump is generated when having high evaporating temperature, low pressure and a low outlet temperature out of the gas cooler.

Dymola and DaVE was used when investigating the temperature flows in our system on the skate rink side, hot tap water side and the CO_2 heat pump cycle. The heat pump was investigated in conjunction with a heating and cooling circuit. We use a plate heat exchanger for the evaporator due to plate heat exchangers can have an approach of 1°C and works well with low pressures. The gas cooler is a tube in tube heat exchanger due to these kind of heat exchangers is working nicely with high pressures and temperatures, as well as low mass flows. The PI-regulator is regulating the high-pressure side and stabilizes the pressure at a level of 100 bar. We want to have a pressure beneath 130 bar to protect the components over time. A pressure of 100 bar will give us the desirable high temperature of 75°C of DHW. This is a pressure of 5 bar over the optimal pressure mentioned in chapter 2.6.4. In our models superheat and internal heat exchanger was applied to our system to increase the COP value and decrease the outlet temperature of the gas cooler. By applying evaporator superheat of 6.6K or internal heat exchanger for rising the system superheat to 10K this let to an increase of the $COP_{heating}$ value from 3,3 to 4,5 when heating the building. This is a rise of COP value of 36,4%. When cooling the building the $COP_{heating}$ for DHW will rise from the value of 3,26 to 4,29 and the value of $COP_{cooling}$ will rise from 2,26 to 3,29. This is a rise of 31,6% and 45,6%.

5.4 Cost saving potential

The waste heat utilized from the ice skating rink will contribute to an energy saving potential for energy companies and decrease of the electricity demand. The operational cost savings by using a heat pump instead of a boiler and greywater implementation will benefit the inhabitants with lower energy bills. A report made by Nord Pool was investigated when considering the cost saving potential of utilizing the waste heat source. The report considered a ten year long period from 2007-2017 with information about the average annual Nordic/Baltic spot prices per year. We chose to look at the lowest and highest spot prices and investigated the range of money which can be spared by using a low-temperature thermal grid. The waste heat from the skate rink is equal to an annual saving potential of 876176-2105313 NOK. It must be applied a business plan for how the waste heat from the ice skating rink will be implemented in a low-temperature district heating system. When producing DHW it is possible to gain a $COP_{heating}$ of above 4 when heating and cooling the building. When considering the annual operational costs of production of domestic hot water the yearly amount of money saved for the area when using a heat pump with a COP of 4 is between 566428-1364577 NOK for the building area. By utilizing greywater the cost saving potential is between 484408-1163957 NOK per year for the building area. The total operational costs can thereby be reduced by implementing both CO_2 heat pumps with COP of 4 and greywater tanks by an amount of 1050836-2528534 NOK.

6 Conclusion

The goal of this master thesis was to investigate the energy saving potential of utilizing a low-temperature thermal grid based on waste heat from an ice skating rink. In addition, greywater implementation was investigated as a supplementary waste heat source for the thermal grid to cover the energy demand of the building area at Leangen. The system solution of model 2 was investigated further with CO_2 as working fluid in the heat pump producing DHW.

Considering the reference building simulated in Simien the waste heat from the ice skating rink is able to cover 47,5% of the total energy demand of one building after building stage three. Greywater will cover 33,0%, while electricity and other sources will cover 12,9% and 6,6% of the total energy demand for the building. The reference building have a space heating and cooling demand of 28,1kWh/m^2 and 10,6kWh/m^2 while DHW production constitutes an energy demand of 29,8kWh/m^2. The reference building simulated is thereby categorized as a low-energy consumption building. As the production of DHW constitutes about 30% of the total energy demand of one building, this issue have gotten a lot of attention in this master thesis. The new system solution allows us to utilize waste heat for space heating in buildings as well as reuse the heat further for production of DHW. The concept should enable the use of available energy sources with low temperatures for low-energy consumption and passive house buildings.

A heat pump with CO_2 as working fluid is environmentally friendly, and can achieve a high COP when producing DHW due to the nice temperature fit between CO_2 and water. Model 2 with a CO_2 heat pump was therefore chosen to be investigated further. A CO_2 heat pump producing DHW was evaluated in Dymola in which the inlet temperature of cold city water was 5°C and the outlet temperature was 75°C. Evaporator superheat and an internal heat exchanger was introduced to the heat pump to achieve a higher COP. A heat pump with a system superheat of 10K can provide a $COP_{heating}$ of 4,5 connected to the heating circuit and a COP of 4,3 connected to the cooling circuit which reduces the need of energy input a lot. Model 2 is a system solution which includes hot water storage tanks. This diurnal thermal storage method enables the deliverance of hot tap water to inhabitants at all times and lowers the peak demand of the energy distribution system. The size of such a tank should be between 1460-2098L per reference building according to a user profile and theoretic values for low-energy consumption and passive house buildings. The hot water storage tanks contains water with a temperature difference of 70K. As the temperature difference in a hot water storage tank increases the size is decreasing. Therefore, it will be a good idea to consider to produce and store high temperatures in dense areas.

The reference building made in Simien is a building made on data from reports by Asplan Viak AS and Lund Hagem Arkitekter AS. The area consists of mainly residential buildings in different sizes. It was necessary to simplify and make assumptions of the building area to generate data of the energy consumption of the area. It is therefore important to be critical to the results presented. However, the results will give an indication of the energy saving potential by implementing a local low-temperature thermal grid at the building area of Leangen.

7 Further work

The space heating demand for the user profile was less than for the Simien file. The uptime of heating and cooling may be incorrectly according to real conditions. It will therefore be interesting to look at the user behavior of different kinds of people and thereby increase or decrease the energy consumption of the building area at Leangen. A floor of concrete will work as a diurnal thermal energy storage, and therefore the heating process may not be necessary all the time which also will influence the uptime of our system. This issue should also be investigated further.

The greywater system solution connected to the thermal grid should be investigated, as well as how the high-temperature district heating system and other solutions can be implemented in our low-temperature thermal grid. An evaluation of whether or not the greywater implementation should be placed in the technical room in storage tanks of one building or if the greywater should be sent to centralized storages in the area of Leangen. Different waste heat sources with different temperatures connected to one low-temperature grid could be illustrated and investigated further on in Dymola to gain knowledge of the flexibility of the system. Greywater implementation and filters should be evaluated and there should be made a cost analysis of the equipment needed to make such a system and thereby evaluate if it is cost efficient or not to implement such a solution.

The real real data for space cooling demand of underfloor cooling should be investigated further to make sure the capacity of the heat exchanger is sized correctly. The mass flow coming from either the heat exchanger for space heating or cooling should also be investigated to make sure that the optimal flow into or out of the evaporator on the skate rink side will fit nicely to the mass flow requirement of the heat exchanger. Therefore, it will be of interest to make a bigger model in Dymola including both heat exchangers and heat pumps together. To make an even more dynamic system, it would have been interesting to include the building interface system together with the low-temperature thermal grid.

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A Graphs of waste heat from ice skating rink

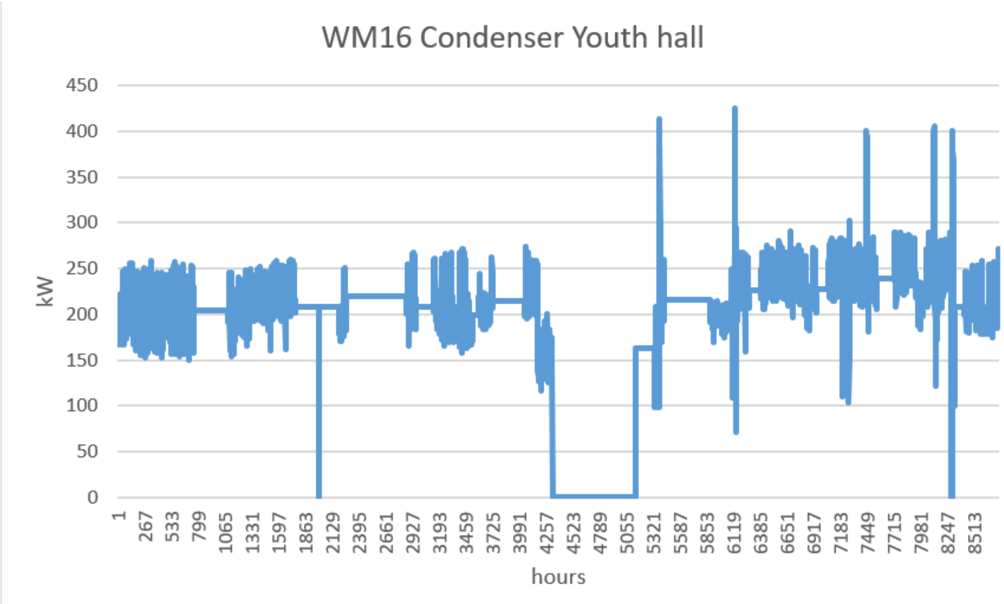

Figure 122: Waste heat from condenser of youth hall

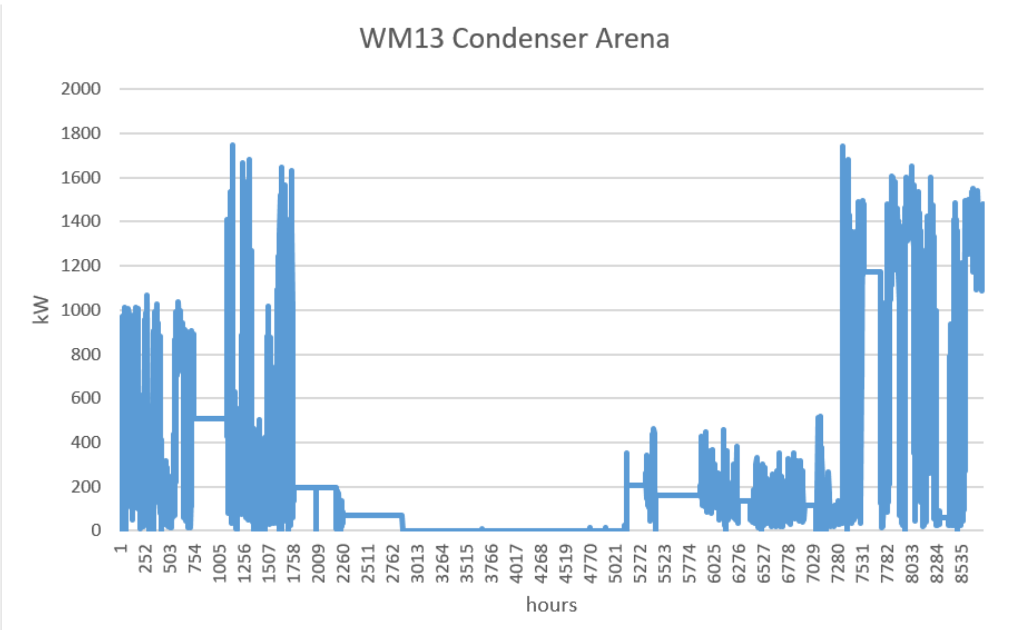

Figure 123: Waste heat from condenser of Arena

B Ammonia heat pump cycles

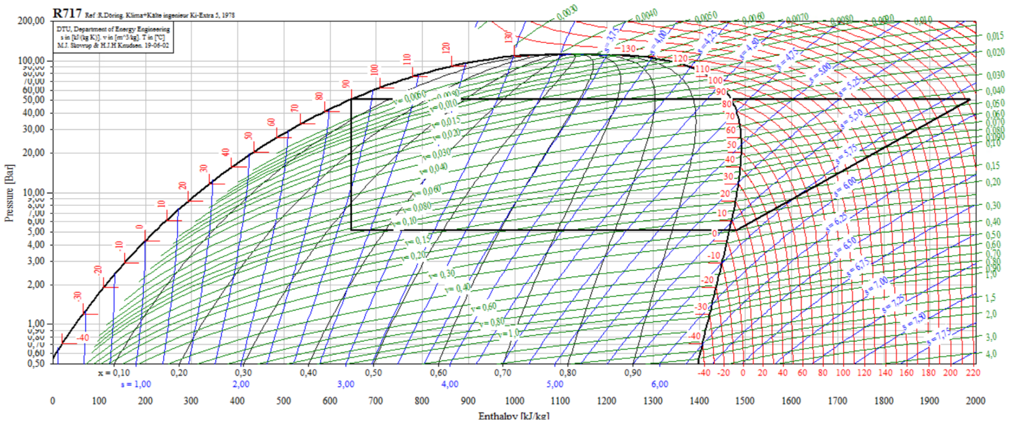

Figure 124: Ammonia cycle - attempt 1

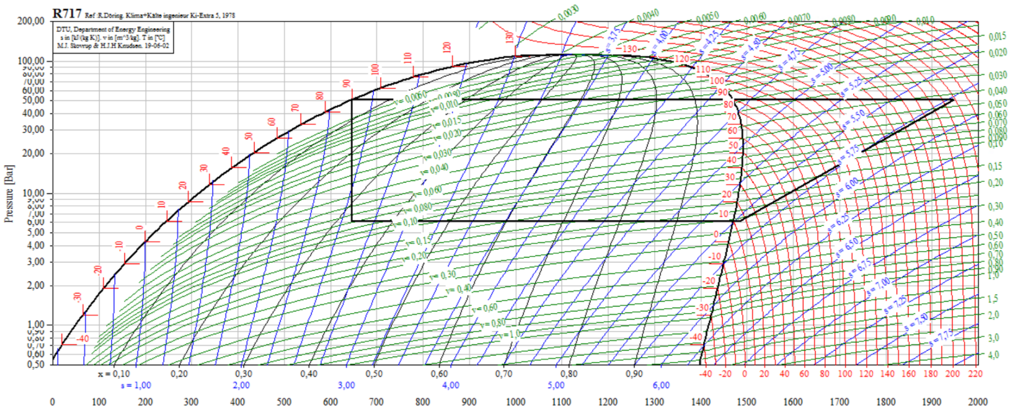

Figure 125: Ammonia cycle - attempt 2

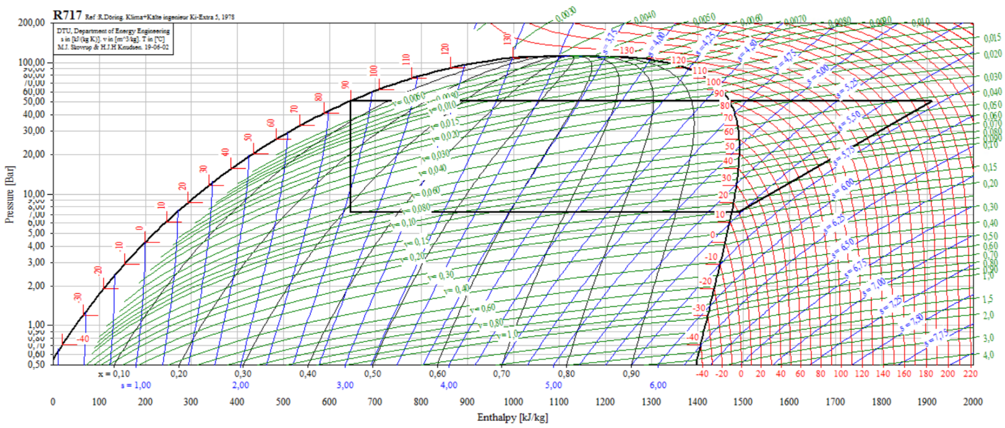

Figure 126: Ammonia cycle - attempt 3

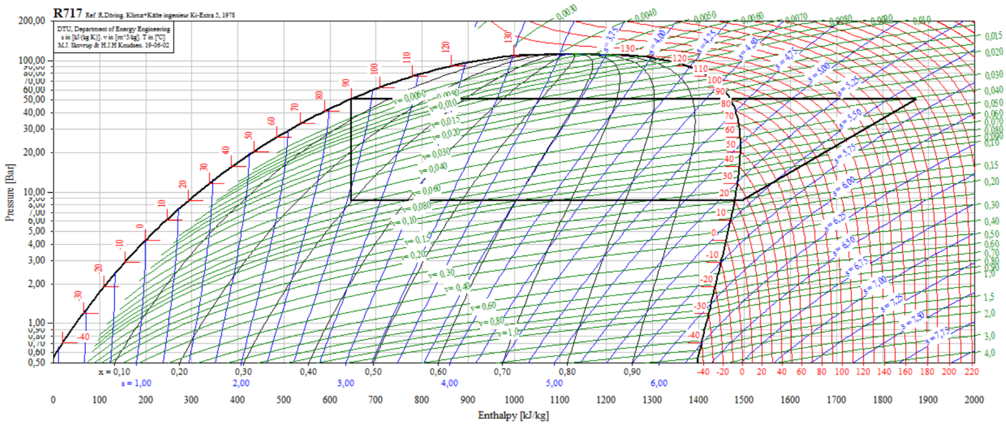

Figure 127: Ammonia cycle - attempt 4

C Propane heat pump cycles

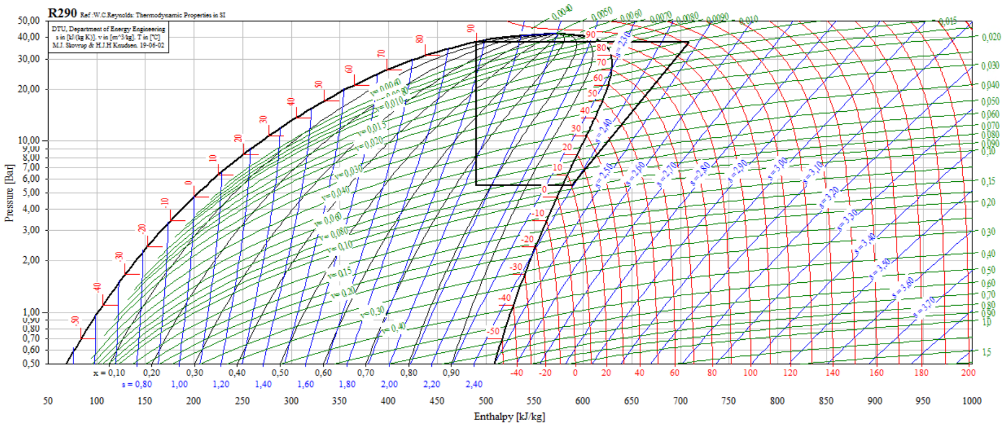

Figure 128: Propane cycle - attempt 1

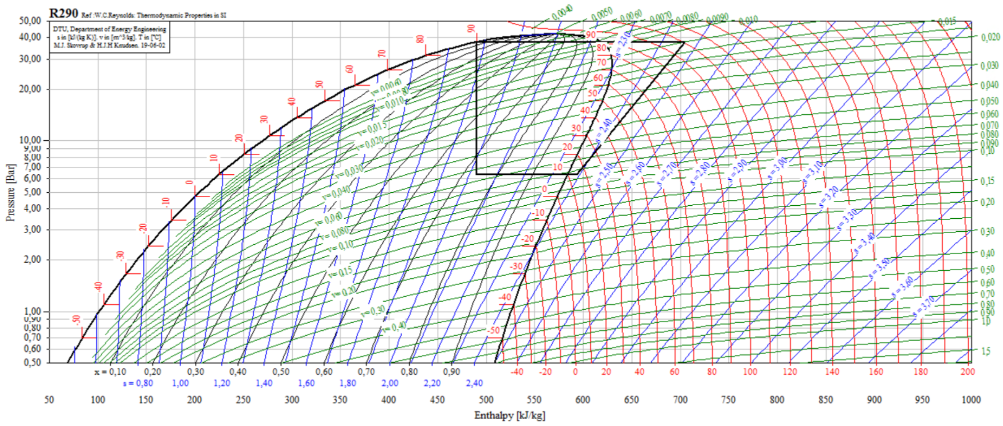

Figure 129: Propane cycle - attempt 2

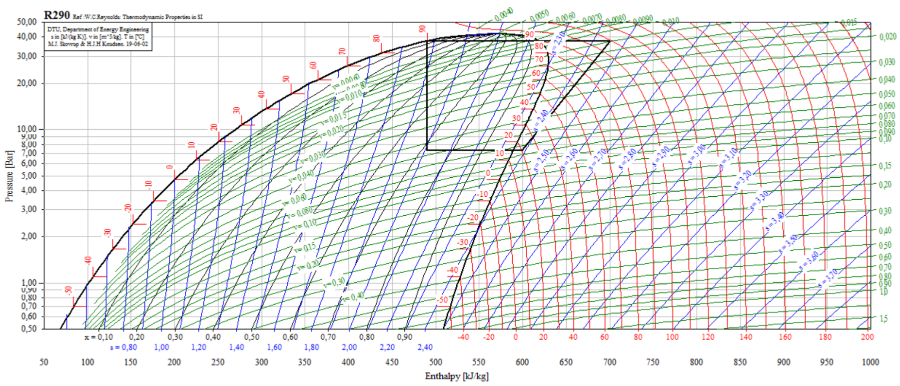

Figure 130: Propane cycle - attempt 3

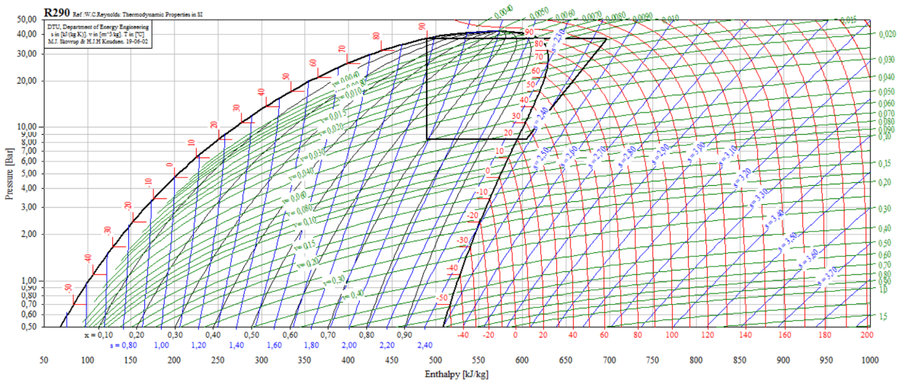

Figure 131: Propane cycle - attempt 4

D CO_2 heat pump cycles - different pressures

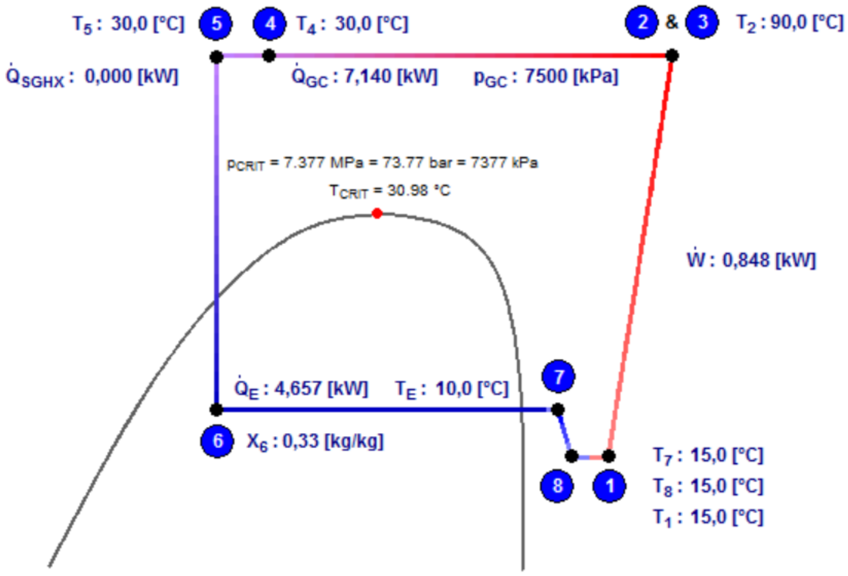

Figure 132: CO_2 heat pump - Pressure, attempt 1

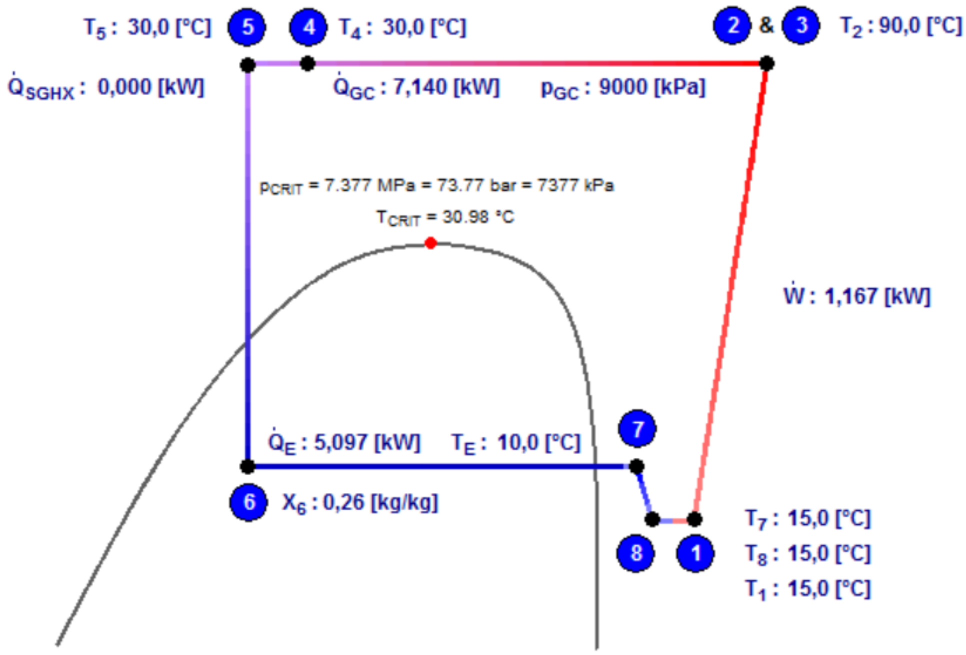

Figure 133: CO_2 heat pump - Pressure, attempt 2

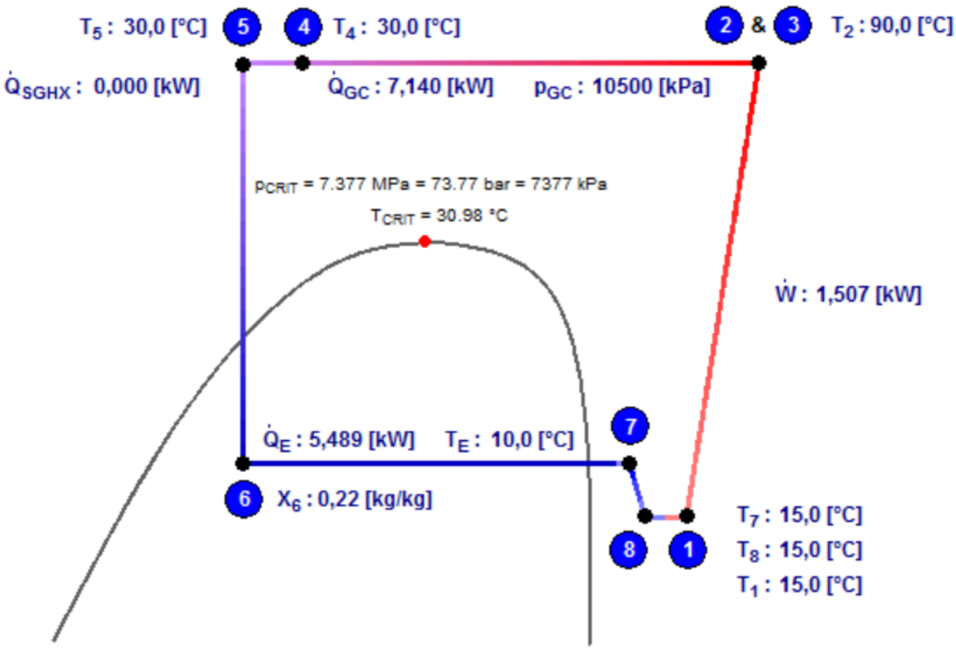

Figure 134: CO_2 heat pump - Pressure, attempt 3

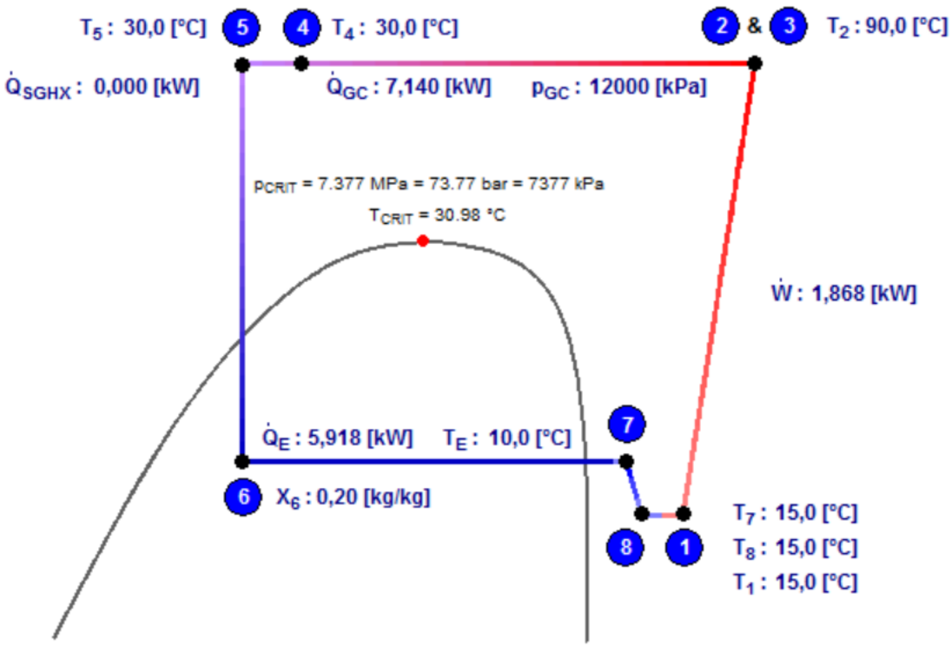

Figure 135: CO_2 heat pump - Pressure, attempt 4

E CO_2 heat pump cycles - different T_E

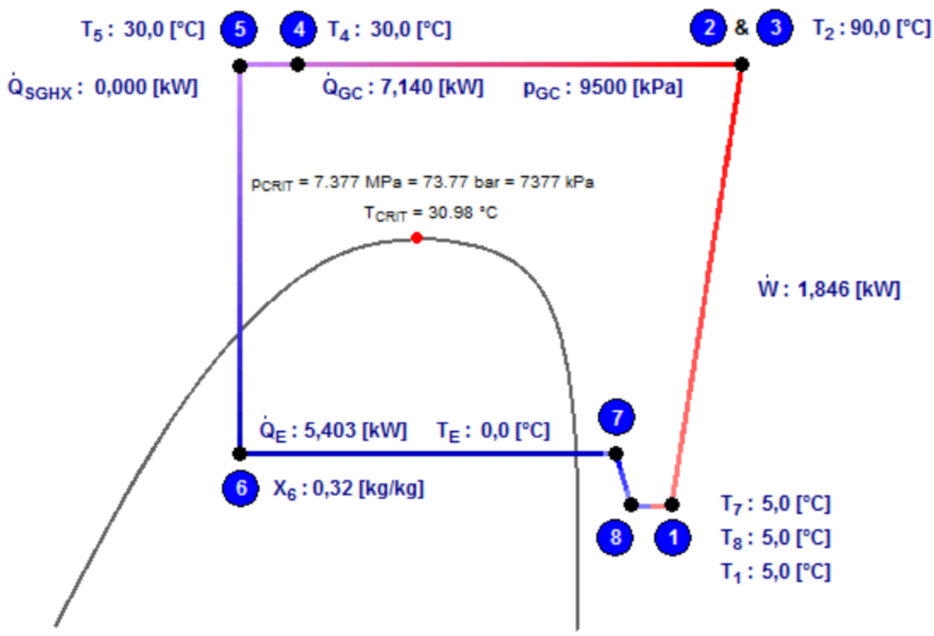

Figure 136: CO_2 heat pump - Evaporating temperature, attempt 1

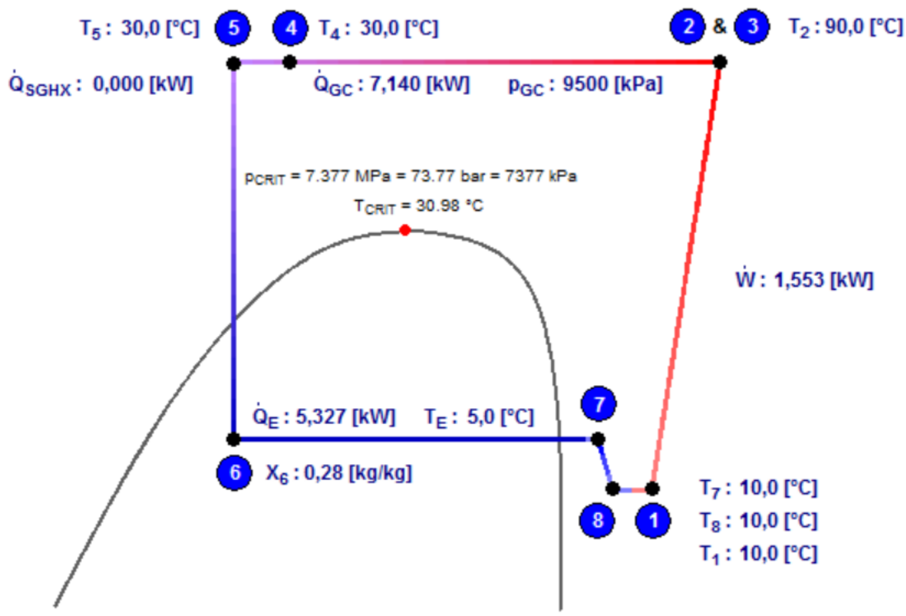

Figure 137: CO_2 heat pump - Evaporating temperature, attempt 2

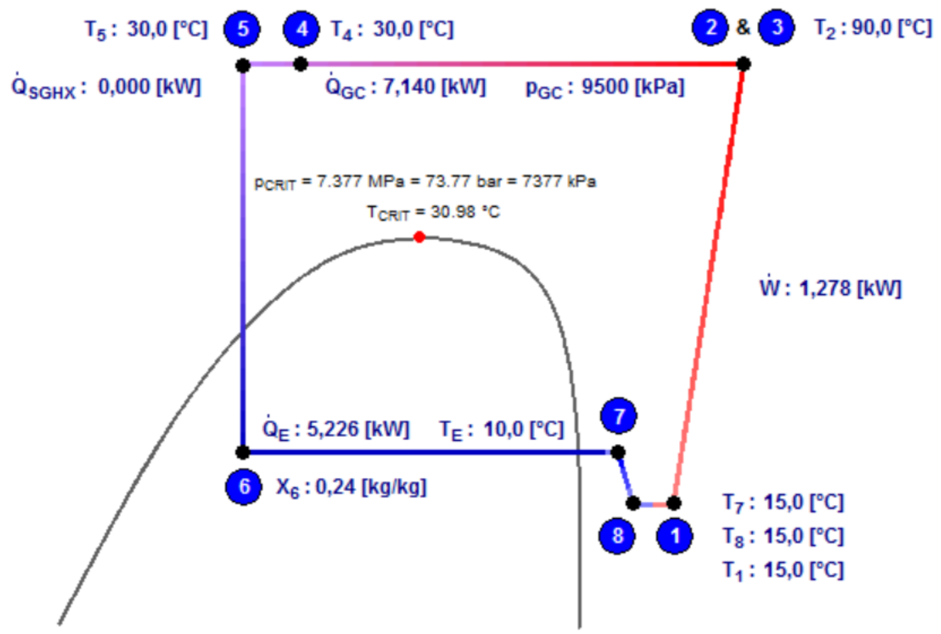

Figure 138: CO_2 heat pump - Evaporating temperature, attempt 3

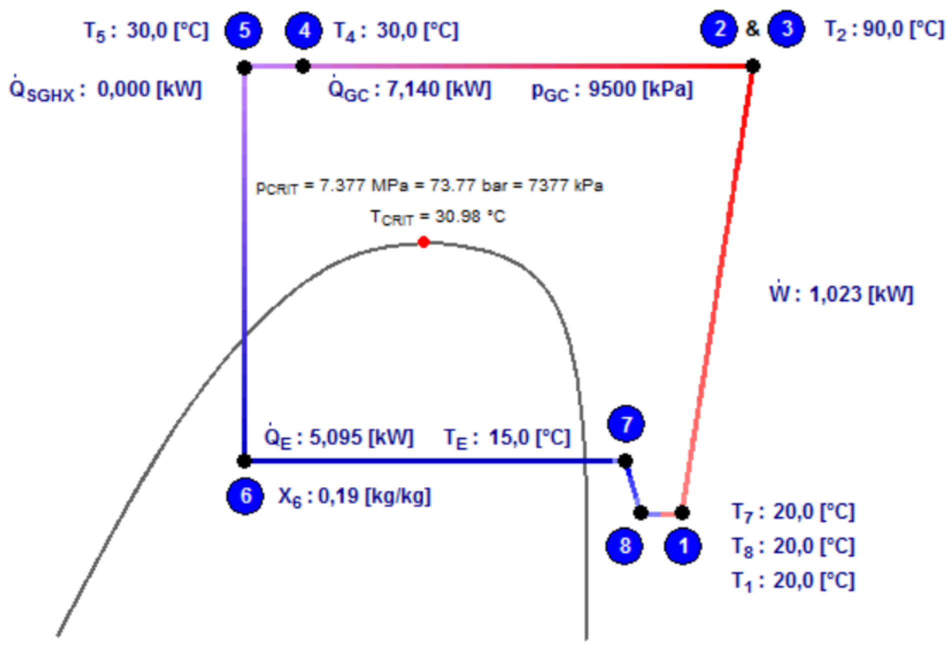

Figure 139: CO_2 heat pump - Evaporating temperature, attempt 4

F CO_2 heat pump cycles - Different outlet temperatures

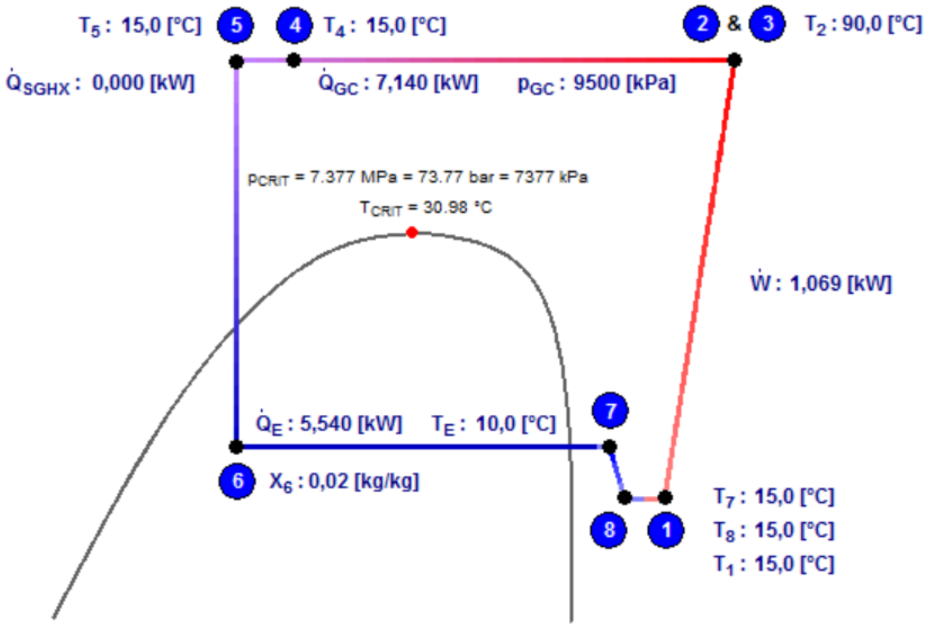

Figure 140: CO_2 heat pump - Outlet temperature, attempt 1

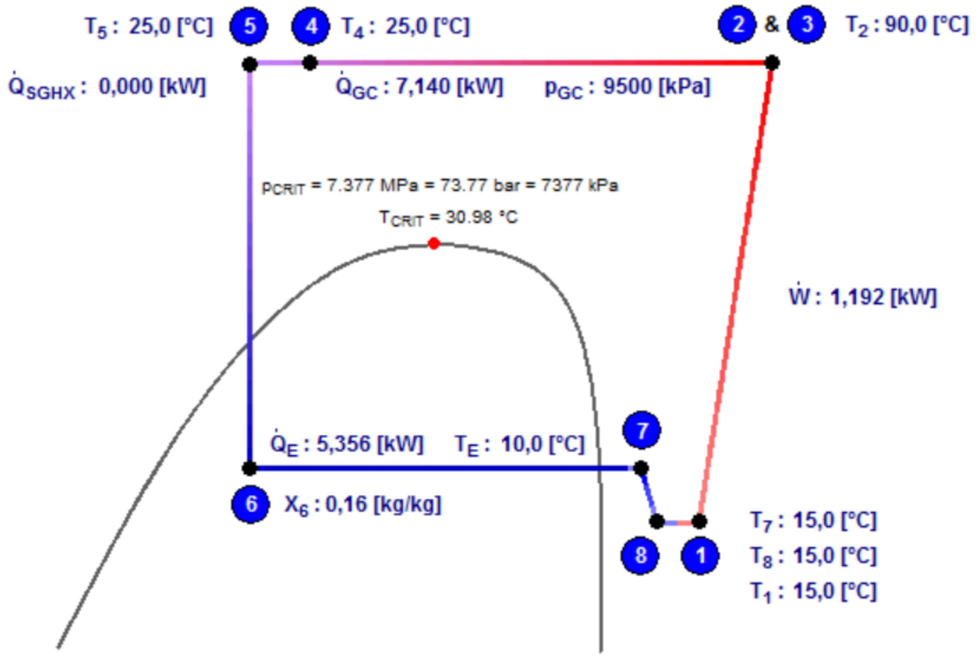

Figure 141: CO_2 heat pump - Outlet temperature, attempt 2

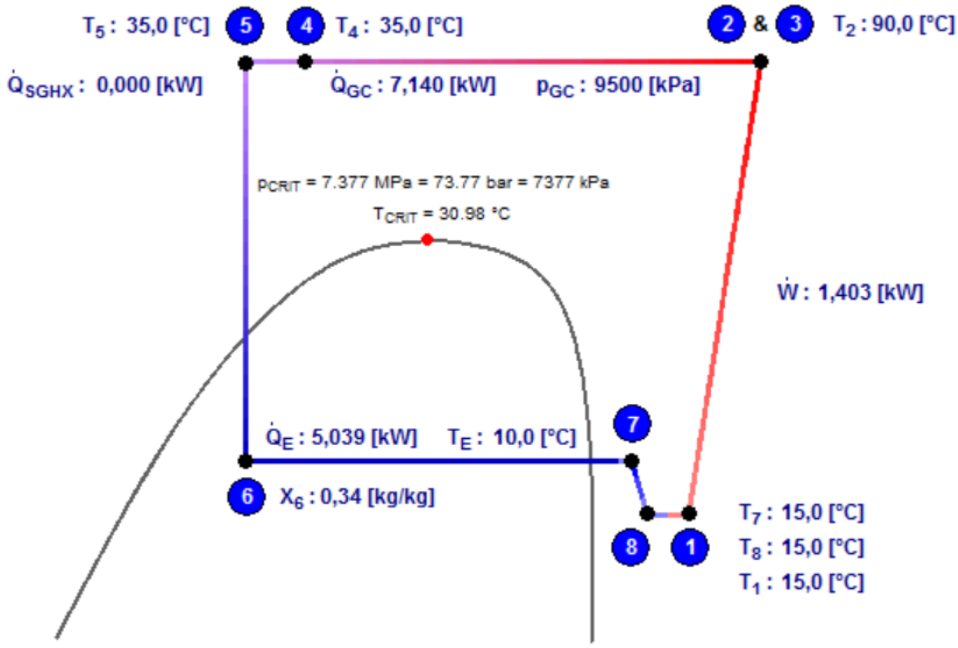

Figure 142: CO_2 heat pump - Outlet temperature, attempt 3

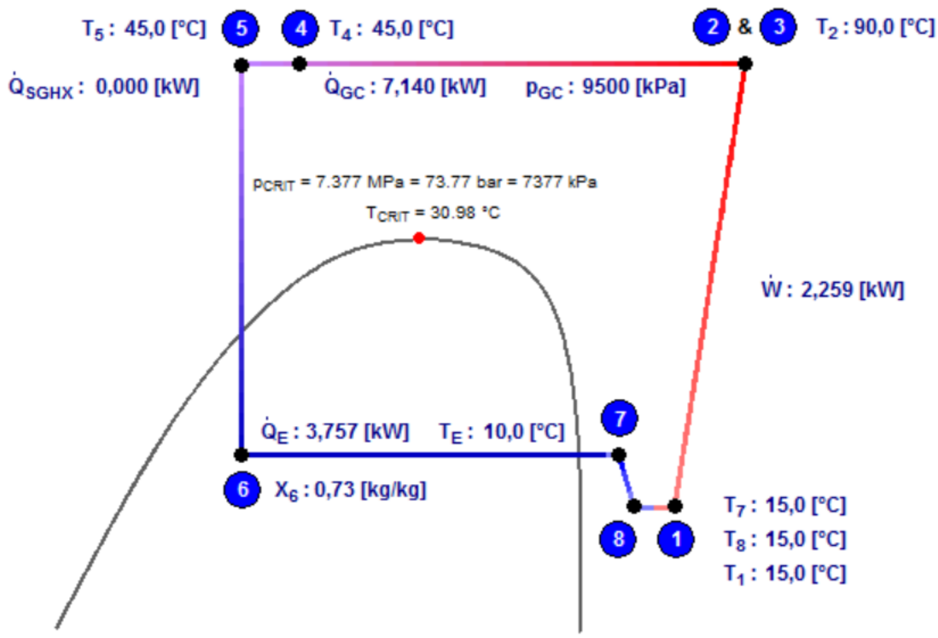

Figure 143: CO_2 heat pump - Outlet temperature, attempt 4

G System sketch of skate rink

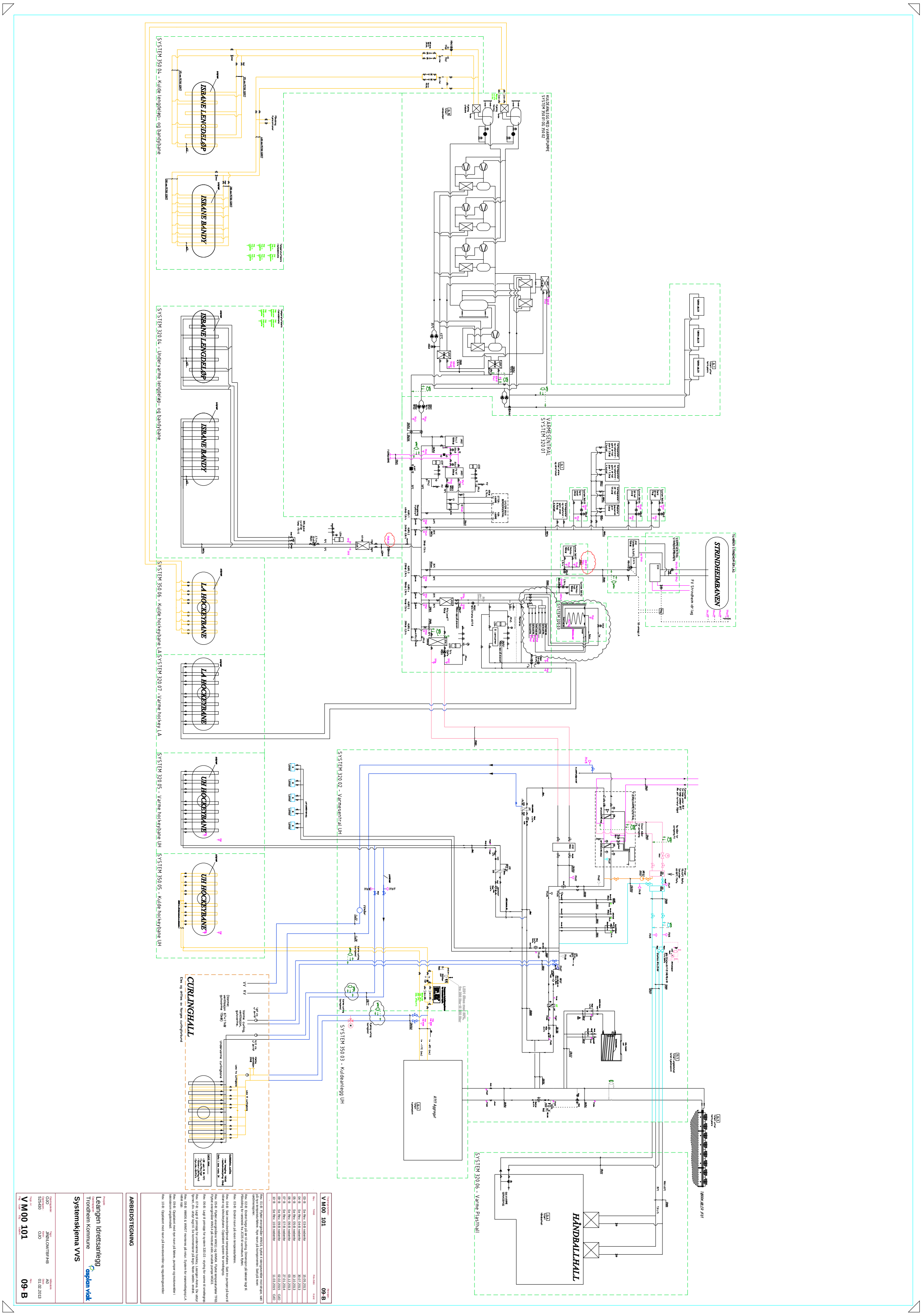

Figure 144: System sketch of skate rink

H Risk assessment form

NTNU	Risk matrix	prepared by	Number	Date
		HSE Section	HMSRV2604	8 March 2010
		approved by	Page	Replaces
HSE/KS		Rector	4 of 4	9 February 2010

Unit: *Department for Energy and Process Engineering* **Date:**

Line manager: *Therese Løvås*

Participants in the identification process (including their function): *Hovedveileder Armin Hafner*

Short description of the main activity/main process: *Master project Eirin Vannes Sundal, "Energy flow analysis of a smart thermal grid at Leangen".*

Is the project work purely theoretical? (YES/NO): Yes *Answer "YES" implies that supervisor is assured that no activities requiring risk assessment are involved in the work. If YES, briefly describe the activities below. The risk assessment form need not be filled out.*

Signatures: *Responsible supervisor:* *Student:* Eirin Vannes Sundal

ID nr.	Activity/process	Responsible person	Existing documentation	Existing safety measures	Laws, regulations etc.	Comment
1	Gather data from the municipalty of Trondheim considering energy meters at an ice skating rink at Leangen.	Eirin Vannes Sundal				

Figure 145: Risk assessment form