Jørgen Formo Kihle

Smart thermal grid concepts for the Leangen area

Master's thesis in Energy and Environmental Engineering

Supervisor: Armin Hafner

June 2019

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NTNU
Norwegian University of Science and Technology
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Kunnskap for en bedre verden

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Sammendrag

Leangen består for øyeblikket av en veddeløpsbane. Koteng Eiendom AS planlegger å rive banen, for å deretter bygge et bærekraftig boligområde med lavt energiforbruk og klimavennlig fotavtrykk. Byggeplanene inkluderer boliger og næringsvirksomhet, i tillegg til helse- og velferdstjenester som barnehage og vernede boliger.

Veddeløpsbanen ligger ved siden av et idrettsanlegg, som blant annet består av flere isbaner. For å holde isbanene kalde er det installert en ammoniakkvarmepumpe. Informasjon fra kontaktpersoner ved dette anlegget viser at anlegget produserte 4 152 116 kWh med spillvarme i 2018. Dette betyr at det ligger en uutnyttet varmekilde med stort potensial i nærheten av Leangen. En mulighet for å dekke Leangens fremtidige varmebehov er derfor å bruke denne overskuddsvarmen som varmekilde, sammen med et lokalt lavtemperaturenerginett. Intensjonen med oppgaven er å utforske nettopp denne muligheten

Ettersom bygningsplanene enda ikke er fastsatt, er det valgt å designe en fiktiv bygning som skal representere den gjennomsnittlige boligblokken på Leangen. Energibehovet for denne bygningen er en kombinasjon av varme- og varmtvannsbehovet fra en passivhusbygning i Trondheim og simulerte kjølebehov fra SIMIEN.

For å best mulig dekke bygningens varmebehov ved bruk av skøytebanenes spillvarme, er det utviklet er energidistribusjonssystem. Systemet fokuserer på å bruke varmtvannet fra lavtemperaturnettet, både i et vannbårent gulvvarmesystem og som varmekilde for en varmtvannsvarmepumpe. Gulvvarmen skal dekke alt av romoppvarming og en CO_2 varmepumpe skal dekke varmtvannsbehovet. Den representative bygningen har i tillegg et betydelig kjølebehov. Det er derfor installert et vannbårent taksystem som absorberer den overflødige varmen i bygget. Dette er koblet til varmepumpen som ekstra varmekilde for varmtvannsoppvarming.

Resultatene viser at spillvarmen fra skøyteanlegget ikke vil være nok til å dekke Leangens varmebehov. Bygningens energidistribusjonssystem ble derfor justert til å kunne inkludere energigjenvinning fra gråvann. Innledningsvis ble det sett på mulighetene for å samle gråvannet fra den individuelle bygningen, og deretter koble gråvannstanken til varmepumpen som en ekstern kilde for varmtvannsoppvarming. Dette førte til en betydelig reduksjon i behovet for spillvarme fra skøyteanlegget. Reduksjonen var imidlertid ikke nok til å kunne dekke det totale varmebehovet. På bakgrunn av dette ble det sett på mulighetene for å samle den totale gråvannsmengden fra boligområdet. Ved å samle dette i et sentralt basseng/tank og så koble det til lavtemperaturnettet via en varmepumpe, ble det totale behovet for spillvarme nok til å dekke det årlige vamebehovet. Selv om denne løsningen dekket det årlige behovet totalt, var varmebehovet i januar fortsatt større enn den tilgjengelige spillvarmen. En mulig løsning er å koble lavtemperaturnettet til fjernvarmenettet. Dette medfører at fjernvarme kan importeres når spillvarmen ikke strekker til.

På tross av at varmebehovet i januar er for stort til å kunne dekkes av den tilgjengelige spillvarmen, vil det resten av året være et stort overskudd av spillvarme, inkludert desember. Dette muliggjør en eventuell lagring av overskuddvarme i et termisk lager for senere bruk. Dersom overskuddsenergien i desember kan lagres for senere bruk, er det mulig å bruke desembers overskudd til å dekke mangelen på spillvarme i januar.

Abstract

At the present, Leangen is occupied by a race course. Koteng Eiendom AS are going to demolish it and build a sustainable community with a low environmental footprint in its stead. The construction plans for this area include housing and businesses, in addition to public services such as a kindergarten and a health and welfare center.

Next to the racecourse is a sports facility containing several ice rinks. To keep the rinks cold, the facility has installed a large ammonia heat pump. Information provided by a contact person at the facility shows that the heat pump system produced a total of 4 152 116 kWh of waste heat in 2018. The excess heat was released into the ambient air through dry coolers on the roof. This means that there is a local source of untapped heat nearby ready to be exploited. A possible way of covering Leangens future residential heating demand can therefore be to use the skating rinks as a heat source in combination with a low temperature thermal grid. The objective of this assignment is to explore this very option.

Given that the construction plans are not yet definitive, it was decided to design a fictive building representing the average residential building at Leangen. The buildings demands are a combination of the hot water- and space heating demands from another passive house building in Trondheim. The cooling demands are a result of simulations in SIMIEN.

To cover the demands by way of the Sports facility's waste heat, an energy distribution system for the representative building was designed. The system focuses on applying hot water from the low temperature thermal grid as the source for both a waterborne floor heating system and as the heat source for hot water heating. The floor heating system is set to fully cover the space heating demands, while a CO2 heat pump covers the hot water demands. The representative building also has a considerable need for cooling. The distribution system covers the cooling demand by way of a waterborne ceiling system. This is connected to the heat pump as an extra heat source for hot water heating.

The results showed that it is not possible to cover Leangens heating demands by using the waste heat from the skating rinks alone. It was therefore decided to adjust the energy distribution system, making it able to include heat recovery from greywater as well. Initially, greywater from each building was accumulated in greywater tanks connected to the hot water heat pump as an additional heat source. This reduced each buildings energy consumption from the low temperature thermal grid. Even so, this reduction was not enough for the sports facility's waste heat to suffice. Subsequently, the accumulation of the entire residential area's greywater production was explored. It was discovered that by accumulating this in a centralized pool and connecting it to the LTTG with a heat pump, the greywater potential increased tremendously. When the greywater energy was directly imported into the LTTG the total annual available waste heat was enough to cover the annual waste heat consumption. However, the waste heat consumption in January still exceeded the available waste heat. A proposed solution was to connect the LTTG to district heating. The district heating could thus be an emergency heat source for when the system is at a lack of waste heat.

Even though the consumption in January exceeded the available waste heat, there was a considerable excess of waste heat the rest of the year. This includes December. If the

excess energy in December can be stored for later use, this can cover the expected deficit in January.

Preface

This master's thesis is written at Norwegian University of Science and Technology (NTNU) during the spring of 2019. The thesis grants 30 credits and concludes my 2-year master's programme.

The supervisors were Armin Hafner, professor at the Department of Energy and Process Engineering, and Hanne Kauko, researcher at SINTEF Energy Research. Thank you for your guidance and your enthusiastic brainstorming regarding the different possibilities for the energy systems developed during this project. Without you, this would not have been possible.

And to my friend, Thomas. The fact that you took the time to read through this entire thesis and proof read, is something I will never forget. You truly are one of a kind.

Jørgen Formo Kihle
Trondheim
June 11th, 2019

Table of contents

Table of figures

List of tables

1. Introduction

1.1 Background and objective

With the recent years focus and emphasis on the climate, pressure on reducing the overall energy demand has increased substantially. It is estimated that buildings today are responsible for about 40 % of the global energy consumption (Rage, Chigot , Anker Hviid, & Köhler, 2017). When aiming for a minimization of the overall energy consumption, applying measures to reduce the energy requirement for buildings will therefore go a long way. Consequently, when planning a new residential area such as Leangen, it is important to ensure that it is energy efficient as well as a worthwhile contribution to a sustainable future.

1.2 Structure and goals

The primary goal of this project is to evaluate different solutions and systems regarding the use of waste heat to cover the heating demand of the future residential area at Leangen. The models include the use of a local low temperature thermal grid (LTTG) for the distribution of the selected sources. The LTTG will supply waste heat to an energy distribution system at each building, which is the main focus of this paper. Waste heat can be applied and exploited in several ways. The systems presented, explore the positive and negative sides to utilizing waste heat in certain ways, as well as the potential of the respective waste heat sources. The evaluation of each system is mainly based on its need for waste heat compared to the available waste heat from the selected source. To best answer the task at hand, intermediate objectives have been set, as listed below.

1. Review of relevant literature.
2. Develop models representing the energy system of the Leangen area.
3. Perform dynamic simulations.
4. Analyze the results in terms of system performance and energy consumption.
5. Proposals to further work.

The assignment consists of 6 chapters. The first chapter is an introduction and is meant to give the reader insight into the background that the assignment is built upon. Chapter 2 is a literature review. This lays the foundation for the models that have been designed in this project. Chapter 3 is the methodology which a gives a deeper explanation of the models' basis and the tools used to design them. The results are displayed and discussed in chapter 4 while the conclusion is presented chapter 5. Chapter 6 is reserved for suggestions to further work.

1.3 Limitations and parametres

Leangen is a rather large area, and the plan is to build between 1660 and 1770 apartments in buildings varying from 1-8 stories high. To narrow down the circumference of the project, several parameters have been set. Instead of looking at the entire area, the emphasis has been on an average building of five stories and a total of four apartments per floor. As the possibilities are many, it was decided to look at two main waste heat sources on which to focus the work and analysis. Each system is assigned two modes. One mode using the LTTG as heat source for domestic hot water production, and one mode that uses space cooling as heat source. The two models are referenced as *Normal mode* and *Cooling mode*.

2. Literature study

2.1 4^{th} Generation District Heating (4GDH)

The distribution of energy is an essential part of exploiting an energy source. Being able to transport the energy in an energy efficient and exploitable way is fundamental for securing and maintaining a sustainable system. District heating is such a distribution system.

District heating is often divided into four generations. The first generation used steam as its heat carrier. Consisting of steam pipes in concrete ducts, steam traps and compensators, the first-generation district heating was the primary technology until the 1930s. The high temperatures required for steam lead to significant heat loss and have sometimes been the cause of extensive and deadly explosions. Also, the condensation pipes have been prone to corrosion leading to an even lower efficiency. The second generation was developed during the 1930s. The heat carrier changed to pressurized hot water with supply temperatures often exceeding 100 °C. It typically consisted of water pipes in concrete ducts, tube-shell heat exchangers and large valves. The third generation, often called "Scandinavian district heating", still used pressurized hot water, however the supply temperature was now mostly below 100°C. With an emphasis on prefabricating the components, the district heating infrastructure was easier to build, and the lower temperatures lead to a decrease in heat loss. In accordance with the obvious trend, the 4^{th} generation district heating (4GDH) has an even lower supply temperature, and is mostly comprised of prefabricated and easy-to-install components. The philosophy behind 4GDH is its important role in the future of sustainable systems as it opens for new ways of exploiting renewable energy sources. (Henrik Lund, 2014)

One of the most prominent abilities of 4GDH is the ability of supplying low temperature district heating for space and DHW heating. With a sufficiently low temperature of 30-40°C, the district heating can supply a wall- or floor heating system, reducing the overall energy demand of the building with a substantial amount. Another possibility is to use the district heating for DHW. By applying substations without DHW-storage at the end user, as well as using pipes with small enough dimensions, the how water volume will be low enough for the problem with legionella to be minimized. Thus, a temperature of 45-55°C of DHW supply could suffice. If the supply temperature is lower, around 30°C, the 4GDH can be used to preheat the DHW. The preheated water can then be lifted to acceptable DHW temperatures by a local heat pump device. (Henrik Lund, 2014)

Another benefit of the 4GDH is that the low supply temperature leads to lower heat loss. A distribution network supplying 50°C and a return temperature of 20°C, will cut the average temperature in half, if compared to the 3^{rd} generation. In combination with lower pipe dimensions and a lower peak flow, heat loss can be reduced by a factor 4, in comparison to the 3^{rd} generation district heating. (Henrik Lund, 2014)

Since the requirements for temperature levels are substantially reduced in the 4GDH compared to the earlier generations, the potential of waste heat recovery from industry and commercial buildings is greatly increased. (Henrik Lund, 2014)

2.2 Waste heat recovery

The focus on waste heat recovery has seen an increase the last few decades. Waste heat recovery has a positive effect on the environment as it decreases the need for power and will ideally influence worldwide power production. In addition to its environmental effects, waste heat recovery also allows for economical rewards. A reduced need for power means less expenses in electricity purchases.

2.2.1 Westhill recreation centre

A common source for waste heat is skate rinks. Skate rinks, both indoor and outdoor, inhabits large refrigeration systems that keep the ice surfaces cold. These refrigeration systems work by way of exporting excess heat allowing for the ice to stay at the desired temperature. In most cases the excess heat from the skating rinks are dumped to the ambient, wasting the heat.

In 2012 Westhill recreation centre was commissioned in Canada. The complex contains an NHL size indoor skating rink, an outdoor skating rink and a skating trail joining the two together. In addition to the skating rinks and trail, it holds a 20-lane bowling alley, restaurant/lounge, party rooms and a large office space with several sport-related tenants. The recreation centre is set to use the produced waste heat from the skate rinks to cover the heating demand for the rest of the compound.

To keep the skating rinks cold, the recreation centre inhabits a large ammonia refrigeration system. During the winter, the waste heat from refrigeration is rejected through a recovery condenser, heating glycol to about $28°C$. The heated glycol is directly fed to a floor heating system providing heat to about $1765m^2$ of public space. Because of the thermal properties of concrete, the $28°C$ glycol produces a floor-surface-heat of 22-$24°C$, providing excellent levels of comfort throughout the compound.

The refrigeration system has a night set back control. This means that at certain parts of the day, the compressors are shut off, no longer able to provide the necessary heat for space heating. To account for this, the skating rinks are comprised of a sub-floor thermal storage. Traditionally, the sub-floor heating systems are kept at $4.5°C$ to preventing frost heaves in the foundation caused by long term refrigeration. In this case, the sub-floor system has extra insulation between the ice pad and the heating floor. This allows the sub-floor system to hold temperatures up to $24°C$ without causing the ice pad to melt. With the combination of long refrigeration run times and the sub-floor thermal storage, the system has uninterrupted supply of energy throughout the day. As well as providing heat for space heating, the ammonia refrigeration system supplies heat for the facilities domestic hot water demand. The water is preheated to $49°C$ by the main refrigeration system while an additional heat pump, using the energy recovery system as a source, provides the resulting energy, lifting the water up $60°C$. Thus, the waste heat from the skating rinks cover both the space heating- and domestic hot water demands of the entire complex.

After covering the complex's heating demands, only 40% of the waste heat is recovered. Instead of dumping the remaining 60%, it is sent through 4GDH system to a nearby housing development providing a source for the household heat pumps. Thus, every bit of the waste heat produced by the skating rinks are consumed instead of dumping it to the outdoor air. (Sutherland, 2015)

2.2.2 Greywater heat recovery

The improvement on building standards have been substantial during the last years. In fact, the current regulations for buildings in Norway states that any new building must have a heating demand equal to that of a passive house. However, even though the heating demand in buildings have decreased, the hot water demand has seen little improvement. According to *A key review of non-industrial greywater heat harnessing* (Mazhar, Liu, & Shukla, 2018), the hot water demand at a passive house energy level, represents almost 50% of the total mean energy demand in residential buildings, as displayed in Figure 1.

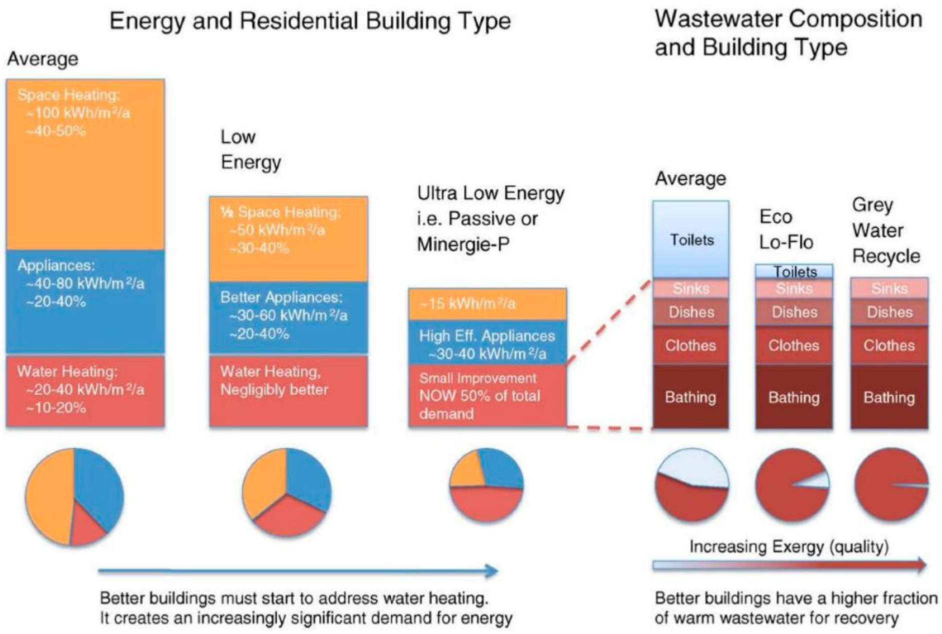

FIGURE 1: DISTRIBUTION OF ENERGY USE IN DIFFERENT BUILDING TYPES

The same figure also shows that the majority of hot water consumed is flushed down the drain as greywater. Thus, large amounts of heat are poured down the drain every day in both domestic households and commercial buildings. Representing most of the energy budget, hot water demand offers the largest reduction potential in buildings, granted that they are close to a passive house standard. Because of this, there has been an increasing interest in harnessing the waste heat, which would not only reduce the energy demand, but also make the buildings more efficient and environmentally friendly.

Drain water can be split into three main groups: light load greywater, heavy-/dark load greywater and blackwater. Light load represents the drainage from showers, basins and bathtubs. This is the cleanest of the three, containing mainly soaps and organic particles from people. The heavy/dark load is typically the drainage from dishwashers, kitchen sinks and washing machines, and typically contains leftover pieces of food, oil and grease, heavy metals, detergents and bacteria. The last one, blackwater, is mostly toilet flushes and contains urine, feces, and toilet paper. Of the three types the light load greywater is the most promising, since it is of a generally high temperature but still relatively clean. (Mazhar, Liu, & Shukla, 2018)

In the domestic household, water is used for several different purposes by both the household's occupants and its many devices. In order to map the potential for the available greywater it is important to know the total use of water as well as what it is used for. In *A key review of the domestic water use in Britain,* the domestic water use of the average household in Britain is mapped (Mazhar, Liu, & Shukla, 2018). The results are listed in Table 1 along with the estimated average temperatures of each appliance (Energy saving trust, 2013). It is important to note that the temperatures listed in Table 1 are the input temperatures of the appliances, and that the output temperature is usually 5-10°C lower than the consumed temperature.

TABLE 1: THE DISTRIBUTION OF WATER USAGE ON DIFFERENT APPLIANCES. (MAZHAR, LIU, & SHUKLA, 2018) (ENERGY SAVING TRUST, 2013)

Water appliance	Share of water usage [%]	Average temperature [°C]
Toilet	22	8
Shower	25	37-45
Bath	8	37-45
Dishwasher	1	60-85
Hand wash dishes	4	50
Washing machine	9	30-60
Bathroom (hot tap)	7	-
Other(cold taps)	22	<20
Garden	1	8
Car	1	8
Total	100	-

Of the many appliances of water usage, the most promising sources of hot greywater is showers/baths, washing machine and the dishwasher. The shower/bath delivers a high volume with relatively high temperature, while being characterized as light load greywater. The wastewater from the dishwasher and washing machine is characterized as dark load greywater, and does not deliver high volumes, but it has a very high average temperature. The hand washing of dishes might seem better than the dishwasher, but the hand washing of dishes takes place in the kitchen sink and is more unpredictable as people vary a lot on the use of hot water in the kitchen sink. Lastly is the cold taps.

The average water consumption habits from the relevant appliances, as well as their respective wastewater production, are listed in Table 2-Table 5. Table 6 displays a combination of the greywater production as well as the average input and output temperatures.

Table 2: Average daily shower wastewater production per person (Biermayer, 2006)

Parameter	Value
Showers per day per person	0.70
Average shower duration [min]	8.20
Shower flow rate [kg/min]	9.46
Shower wastewater per person per day [kg]	**54.32**

Table 3: Average daily bath wastewater production per person (Energy saving trust, 2013)

Parameters	Value
Baths per day per person	0.186
Average water use per bath [kg]	80
Bath wastewater per person per day [kg]	**14.86**

Table 4: Average daily washing machine wastewater production per person (Energy saving trust, 2013) (Mazhar, Liu, & Shukla, 2018)

Parameters	Value
Washing machine use per day per person	0.67
Washing machien wastewater per use [kg]	50
Wahing machine wastewater per day per person [kg]	**33.5**

Table 5: Average daily dishwasher wastewater production per person (Energy saving trust, 2013)

Parameters	Value
Dishwasher per day per person	0.23
Average water use per use [kg]	14
Dishwasher wastewater per day per person [kg]	**3.22**

TABLE 6: GREYWATER PRODUCTION AND TEMPERATURE OF THE RESPECTIVE WATER APPLIANCES. (ENERGY SAVING TRUST, 2013) (MAZHAR, LIU, & SHUKLA, 2018)

	Wastewater per day per person[kg]	Temperature input [°C]	Temperature output [°C]
Shower	54.32	41	33.5
Bath	14.857	41	33.5
Washing machine	13	45	37.5
Dishwasher	3.22	73	65.5
Total	85.40	42.82	35.32

The potential energy that can be harvested from greywater can be calculated by equation (1) and (2). In equation (1), Q is the energy potential in [kJ], m is the water mass in [kg], C_p is the specific heat capacity in $[\frac{kJ}{kg \cdot K}]$ and ΔT is the change in temperature. In equation (2), ṁ is [kg/s], which changes Q from [kJ] to [kW].

$$\boldsymbol{Q = m \cdot C_p \cdot \Delta T} \tag{1}$$

$$\boldsymbol{Q = \dot{m} \cdot C_p \cdot \Delta T} \tag{2}$$

2.3 Hydronic heating and cooling systems

2.3.1 Heat transfer

To better understand the hydronic systems, it is important to grasp the principles under which they function.

Physical heat transfer happens either through radiation or conduction. Radiation is the transfer of energy through electromagnetic waves. Conduction is the transfer of energy on an atomic level through the movement of molecules, atoms and electrons. There is also a third mode called convection. Convection is the flow of a substance which has a dominant influence on the local conduction. Heat transfer through conduction, $\dot{Q}$, is calculated by equation (3) where A is the cross-section area normal to the x-direction of the heat flux, k is the materials conductivity and dT is the change in temperature. (Stavset, 2016)

$$\boldsymbol{\dot{Q} = -kA\frac{dT}{dx}} \tag{3}$$

Generally, convection is of far less significance than conduction or radiation, yet in systems such as floor- or wall heating, it is of great importance. For instance, when the floor is heated, air will be heated as well. When the air close to the floor is heated its density is reduced, causing it to float upwards past the surrounding, heavier, colder air.

This phenomenon is called natural convection and can be expressed by equation (4), where U is the heat transfer coefficient, A is the surface area, T_s is the surface area temperature while $T_\infty$is the temperature far away from the surface. (Stavset, 2016)

$$\dot{Q} = UA(T_s - T_\infty) \tag{4}$$

Radiation is emitted by all objects with a temperature higher than absolute zero. The emission of thermal radiation plays an important part of all heating systems, and can be calculated by equation (5), where A is the surface area, ε is the emissivity, σ is the Stefan-Boltzmann constant, $T_s$is the surface are and T_∞ is the surrounding temperature. (Stavset, 2016)

$$\dot{Q} = \varepsilon\sigma A(T_s^2 - T_\infty^4) \tag{5}$$

2.3.2 Heating systems

Hydronic heating systems are divided into two categories, the first one being systems that make use of the building surfaces such as walls, floors and ceilings. The second is systems that use more compact spot heaters such as radiators.

Because it is well suited for 4GDH, and has a great temperature gradient, floor heating has seen an upswing in popularity, especially in new buildings. It has a typical supply temperature of 35-40°C and a standard temperature drop of 5K through the pipes. The floor surface temperature is typically 23-28°C and vary based on the spacing of the water pipes, water flow rate, thermal resistance of the floor, and the spacing of the water pipes. According to *Heat and cool distribution systems within buildings,* a scientific report on hydronic cooling systems, the heating output can be up to $80 \frac{W}{m^2}$ (Stavset, 2016). The floor heating's vertical temperature gradient is displayed in Figure 2, along with the ideal temperature gradient. (Stavset, 2016) The figure shows that floor heating has a close to ideal vertical gradient. This is largely because of natural convection. By supplying heat from the floor, natural convection will automatically distribute the heat across the room.

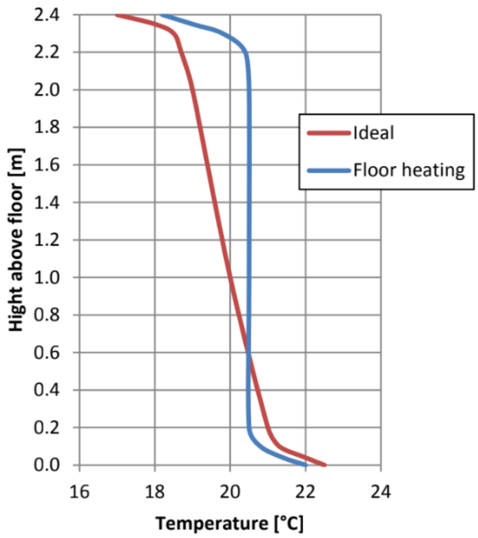

FIGURE 2: FLOOR HEATING TEMPERATURE GRADIENT COMPARED TO THE IDEAL TEMPERATURE GRADIENT (STAVSET, 2016)

Concrete is commonly used in buildings, and play a big role in the use of floor heating systems. Since concrete has a large thermal mass, a floor heating system in a concrete floor will ha a slow response time. In buildings where the heating demand is in constant change, this can become a challenge. However, if variations in demand can be predicted, the thermal mass can be used to reduce the demand during peak loads. (Stavset, 2016)

Since the heat output vary according to the air temperature, the system is partly self-controlling. Equation (9) and (10) show that as the floor heats the air, the temperature difference between the air and the source will be reduced which in turn reduces the heat exchange. Even though the system is partly self-controlling, there are ways of manually controlling the heat output. One way is increasing or decreasing the mass flow of water through the system. This can be done with valves, either in the entire system, or in certain areas. Thus, it is possible to reduce the heat output selectively in different rooms or parts of the system. Another possibility is changing the supply water temperature. As opposed to reducing the mass flow, this will affect the whole system. The best option is therefore to simultaneously use both. If the temperature is changed based on the weather and outdoor temperature, while the change in mass flow is used to control the heat in each room, the system can be optimized according to the heating demand.

Floor is not the only option. There are also systems based on heating/cooling in walls and ceiling. Floor heating is, however, the option with the best potential and the best temperature gradient fit, making it the preferred option.

2.3.3 Cooling systems

Hydronic cooling systems work in very much the same way as hydronic heating systems. It consists of the same components and work according to the same principles, but instead of supplying, it removes heat. Chapter *2.7.1 Heating systems* explain that floor systems are best suited for space heating because of natural convection. For the same reason, hydronic ceiling systems are best for cooling. Natural convection causes the excess heat to rise to the ceiling where the heat is absorbed into the cooling system and removed. Because of the risk of condensation, the mean water temperature of the cooling system should not be designed to be lower than $14 - 16°C$.

Table 7 displays the heating and cooling capacity for the different hydronic systems. The table shows the superiority of floor heating and ceiling cooling.

TABLE 7: HEATING AND COOLING CAPACITY OF DIFFERENT HYDRONIC SYSTEMS (STAVSET, 2016)

Distribution system	Supply water temperature	Heating/cooling capacity
Floor heating	$30 - 40°C$	$80 \frac{W}{m^2}$
Wall heating	$35°C$	$70 \frac{W}{m^2}$
Ceiling heating	$30°C$	$30 - 40 \frac{W}{m^2}$
Ceiling cooling	$14 - 16°C$	$80 \frac{W}{m^2}$

2.4 Heat pumps

A heat pump is a device used to transfer heat from a heat source to a heat sink. In its most basic form, the heat pump consists of four main components, a compressor, a condenser, a throttling device (usually an expansion valve) and an evaporator. These four components work together using a refrigerant that flows through the cycle to collect heat from the heat source and reject it at the heat sink.

The evaporator is a heat exchanger and can be viewed chronologically as the first component in the cycle. Simply put, the evaporator is the component through which the refrigerant picks up heat from the heat source. Before reaching the evaporator, the refrigerant first flows through a throttling device. The throttling device, which usually is an expansion valve, expands the refrigerant causing it to drop in both pressure and temperature level. As the refrigerant then flows through the evaporator, its low pressure reduces the boiling point so that the collected heat is enough to turn the refrigerant gaseous. Thus, the name evaporator.

After the evaporator comes the compressor. Increasing the pressure means increasing the condensation temperature. This in turn means that if the pressure is sufficiently raised, the refrigerant can give off heat at a higher temperature than the heat source. As well as increasing the pressure, the compressor acts as the heart of the operation, and keeps the refrigerant flowing.

After being compressed, the refrigerant moves on to the condenser to drop of heat. Like the evaporator, the condenser is a heat exchanger, but as opposed to picking up heat and evaporating, the refrigerant drops off heat and condenses.

When the refrigerant is condensed, it flows through the throttling device. The throttling device expands the refrigerant, causing it to drop in both pressure and temperature, readying it for heat absorption in the evaporator.

2.4.1 Pressure enthalpy

The pressure-enthalpy (p-h) diagram is a tool often used in the field of heat pumps as it displays the relationship of a refrigerants pressure and enthalpy. It also shows the temperature and entropy allowing for a good overview of the refrigerant state while performing calculations related to the performance. Figure 3 is the p-h diagram for ammonia, which is here used as an example. One of the most important parts of the p-h diagram is the saturation curve. The saturation line represents pressure-enthalpy points at which the refrigerant is either saturated gas or liquid. The critical point is the top point of the saturation curve, where the refrigerant is both vapor and liquid at the same time. The portion of the saturation curve to the right of the critical point represent the part of the diagram with saturated gas, while the left side is saturated liquid. The area under the saturation curve is the mixed state area, where the refrigerant is part liquid and part gas. The red lines represent constant temperature while the blue lines are constant entropy.

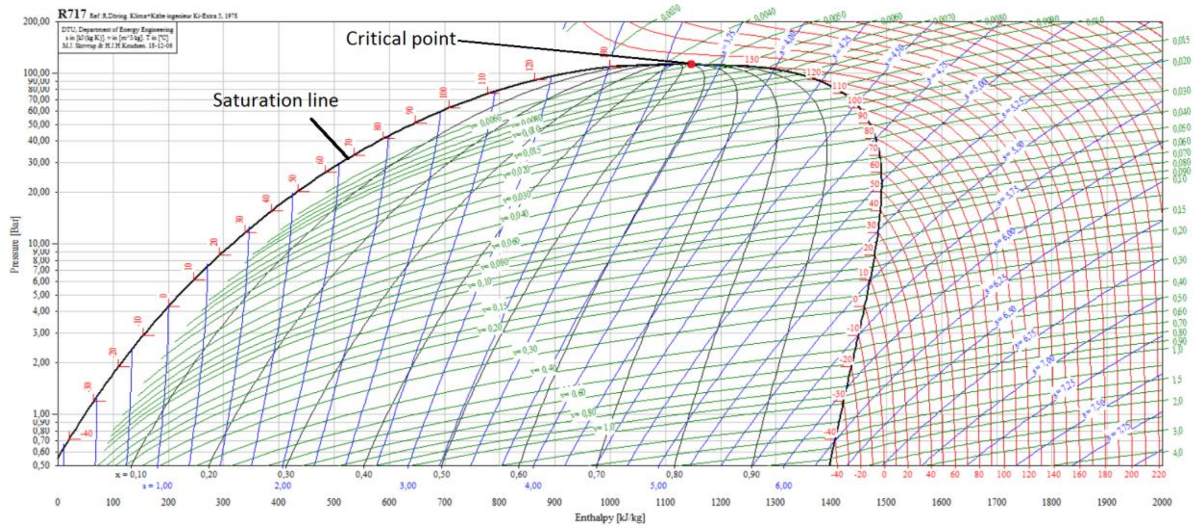

FIGURE 3: AMMONIA P-H DIAGRAM

Figure 4 shows how an ideal ammonia heat pump would be depicted in the p-h diagram. The numbers depicted on the figure describe the state of the refrigerant in between the components in the cycle. 1 is between the evaporator and the compressor, 2 is between the compressor and the condenser, 3 is between the condenser and the throttling device and 4 is between the throttling device and the evaporator.

- 1-2: Compressor input
- 2-3: Condenser output
- 3-4: Throttling
- 4-1: Evaporator input

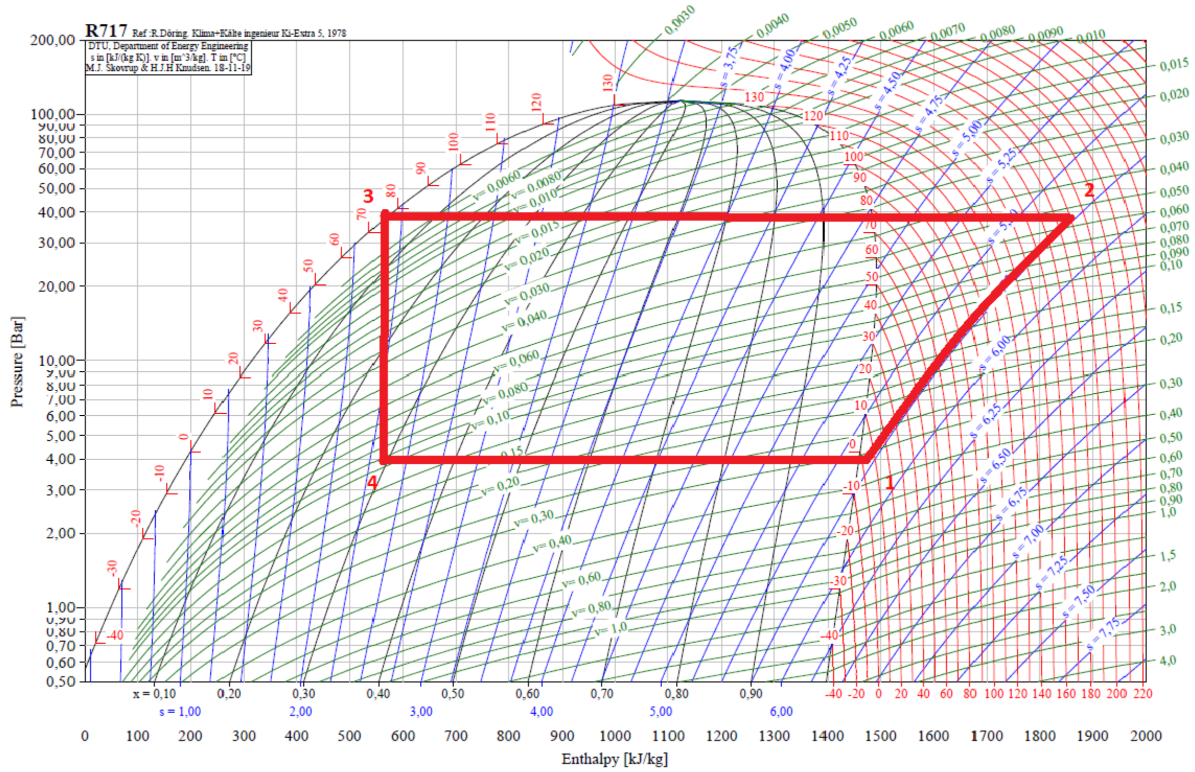

FIGURE 4: EXAMPLE OF AN IDEAL R717 HEAT PUMP CYCLE DEPICTED ON AN R717 P-H DIAGRAM.

The heat delivered from the heat pump equals the energy output from point 2 to 3. This is the heat output from the condenser, and is defined by equation (6). Where Q_c is the condenser heat output in [W], Q_e is the evaporator heat input, and W is the work done by the compressor on the system.

$$\boldsymbol{Q_c = Q_e + W \quad [W]} \tag{6}$$

By introducing the enthalpy and the refrigerant mass flow, equation (6) can be expressed as equation (7). Where h_2 and h_3 is the enthalpy in $\left[\frac{kJ}{kg}\right]$ at point 2 and 3. $\dot{m}[\frac{kg}{s}]$ is the refrigerant mass flow.

$$\boldsymbol{Q_c = \ (h_2 - h_3)\dot{m}\ [W]} \tag{7}$$

The heat input in the evaporator can likewise be defined as equation (8). Where h_1 and h_4 is the enthalpy in point 1 and 4.

$$\boldsymbol{Q_e = \ (h_1 - h_4)\dot{m}\ [W]} \tag{8}$$

In a theoretical, lossless cycle the compression is an isentropic process. In Figure 3 this is illustrated by drawing the compression line (1-2) parallel to the isentropic lines. The difference between theoretical and actual power consumption is defined by the compressor's isentropic efficiency η_{is} and the adiabatic efficiency η. Figure 5 shows a realistic depiction of the heat pump cycle.

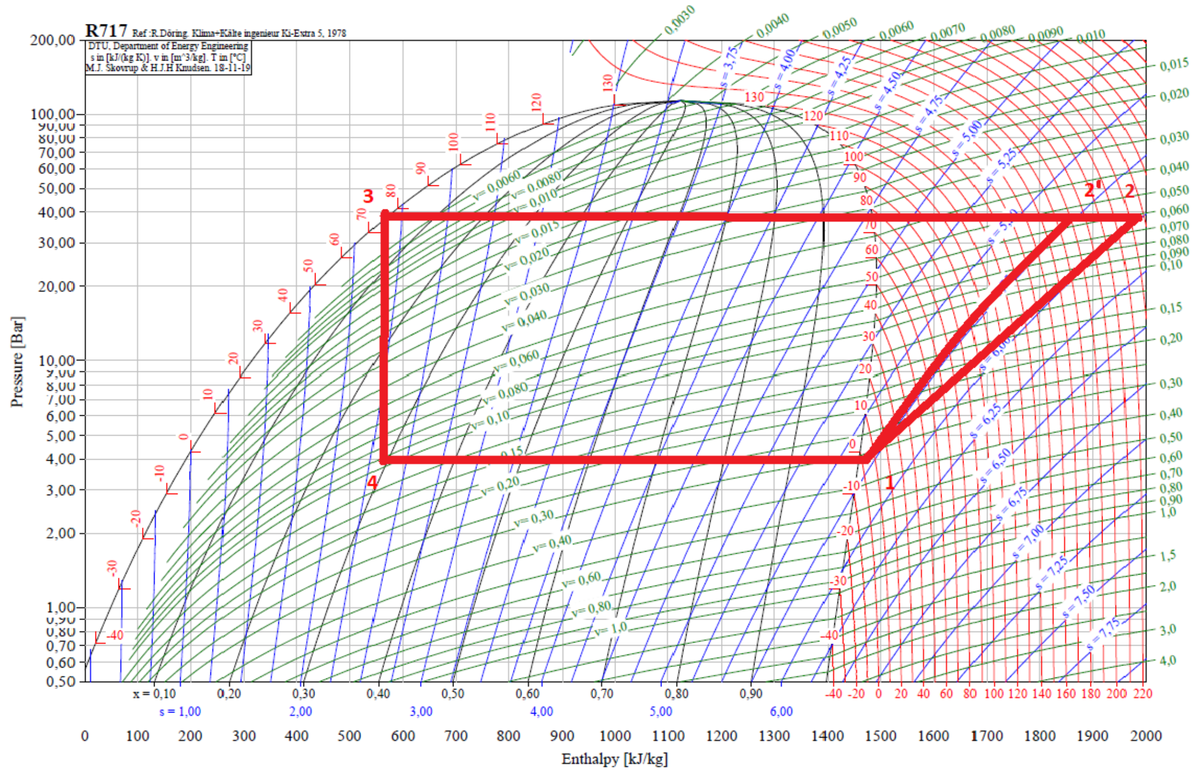

FIGURE 5: REALISTIC DEPICTION OF R717 HEAT PUMP CYCLE ON AN AMMONIA P-H DIAGRAM

The calculation of the assumed compressor work can thus be written:
Δh_{is} is the theoretical, isentropic, compressor input $h_{2'} - h_1$.

$$w = \frac{\Delta h_{is}}{\eta_{is}} \left[\frac{kJ}{kg}\right] \tag{9}$$

$$h_2 = w \cdot \eta \left[\frac{kJ}{kg}\right] \tag{10}$$

$$W = w \cdot \dot{m} \, [W] \tag{11}$$

2.4.2 Coefficient of performance (COP)

The Coefficient of performance of a heat pump is the ratio of provided heat versus the work required to produce it. In a heat pump, the only induced work is that of the compressor elevating the refrigerant to the high pressure and temperature state of heat rejection. The COP_h can therefore be defined as the heat output vs. the compressor work input, as in equation (12).

$$\boldsymbol{COP_h = \frac{Q_C}{W}} \tag{12}$$

COP is not only used for heating, but also for cooling systems. However, there is an important distinction in calculating the cooling COP_c and the heating COP_h. When used for heating, the heat pump collects heat from a heat source through the evaporator, and transports it to the designated area, where it rejects it through the condenser. When used for cooling it is the evaporator that collects heat from the designated area that needs cooling and transports the heat away to be dumped through the condenser. For a heating device with a COP of 5, the requirements to distribute 5 kW from the condenser would be 1kW input from the electric grid and 4 kW input from the evaporator. For a cooling device, on the other hand, a COP of 5 means that to 1 kW of electricity from the grid is needed for the evaporator to extract 5kW. It is therefore important to distinguish between the COP for cooling and heating. COP_c can thus be described as equation (13).

$$\boldsymbol{COP_c = \frac{Q_e}{W}} \qquad (13)$$

2.4.3 CO2 as a refrigerant

2.4.3.1 The transcritical process

For the conventional refrigerants used in standard heat pumps, the critical temperature is typically 80-130°C (Denmark, 2012). A high critical temperature means that it is possible to have a relatively high condensation temperature, which in turn means it is also possible for the condenser to deliver heat at high temperatures. Unlike the conventional refrigerants, CO2 has an especially low critical temperature of 31,1°C. In the supercritical state, gasses cannot condense but instead change in density as they reduce in temperature. Standard CO2 heat pumps using a condenser can therefore not operate with a condensation temperature exceeding 31,1°C. In fact, to have any substantial heat output, the condensation temperature must be even lower. It is reckoned that the maximum condensation temperature for CO2 is 27-28°C.

It is, however, possible to conduct the heat process even though the refrigerant is in the supercritical state. This is called a transcritical process and is illustrated in Figure 6. *a* illustrates a conventional heat pump, where the rejection of heat take effect below the critical point, while *b* shows a transcritical process. In a transcritical process the heat is rejected through the cooling of the refrigerant in gas cooler, rather than a condenser and the process can therefore operate above the critical point.

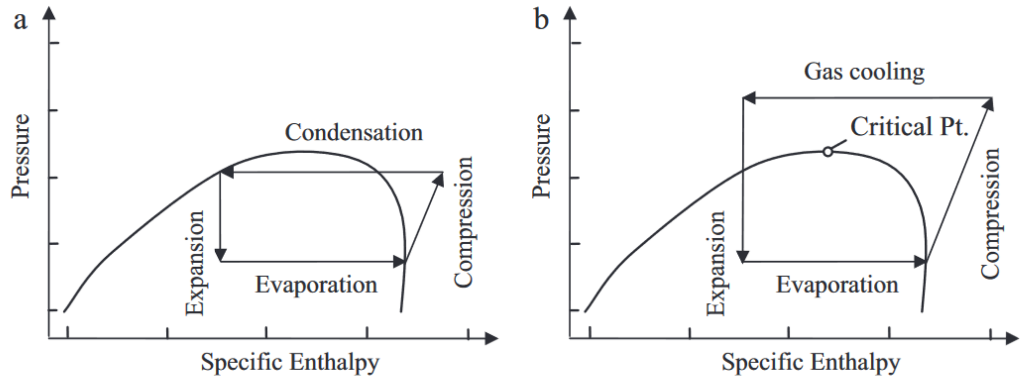

FIGURE 6: (A) EXAMPLE OF A SUBCRITICAL HEAT PUMP CYCLE ON A P-H DIAGRAM, (B) EXAMPLE OF TRANSCRITICAL HEAT PUMP CYCLE IN A P-H DIAGRAM.

The operation of a conventional process must happen at a subcritical level where heat is rejected through latent heat at a constant temperature. In a transcritical process however, the critical point is no longer a restriction, allowing a much higher operating temperature. A higher operating temperature means a larger potential difference between the gas cooler outlet and inlet (temperature glide), which again increases the potential heating performance of the heat pump.

2.4.3.2 Pressure-enthalpy process

Figure 8 shows the principle sketch of a transcritical CO2 heat pump process, while Figure 7 shows the coherent p-h diagram. The installation is comprised of a compressor, a gas cooler, an expansion valve, a suction gas heat exchanger (SGHE), a low pressure receiver (LPR) and an evaporator. The SGHE is an internal heat exchanger that transfers heat from the return gas to the output gas, from the gas cooler and evaporator respectively. The extra cooling, prior to the evaporator allow for more heat absorption, increasing the evaporator capacity. The extra heat absorption subsequent to the evaporator provides superheating and reduces the chance of moisture in the compressor. The extra super heat prior to the compressor reduces the necessary heat input form the compressor which increases COP. The LPR acts as a refrigerant storage, increasing or decreasing the amount of refrigerant in the system as the pressure in the gas cooler is regulated. (Haukås, 2016)

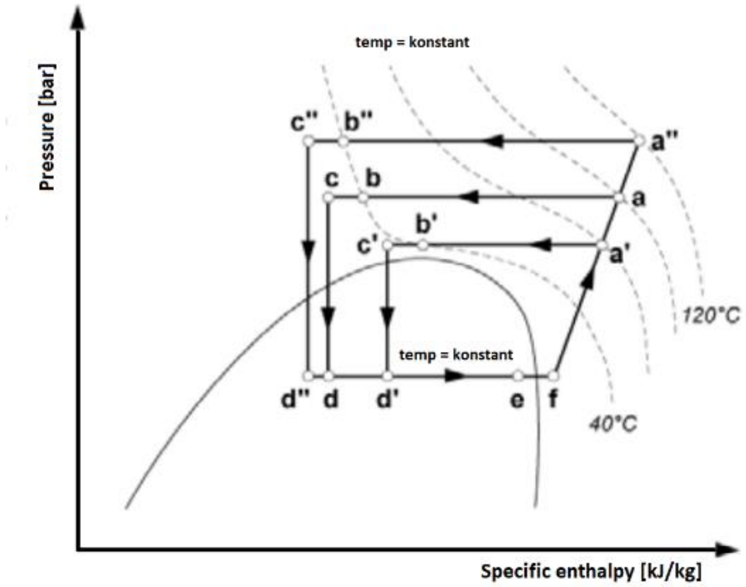

FIGURE 7: P-H DIAGRAM ILLUSTRATION A TRANSCRITICAL PROCESS AT DIFFERENT GAS COOLER PRESSURES. (HAUKÅS, 2016)

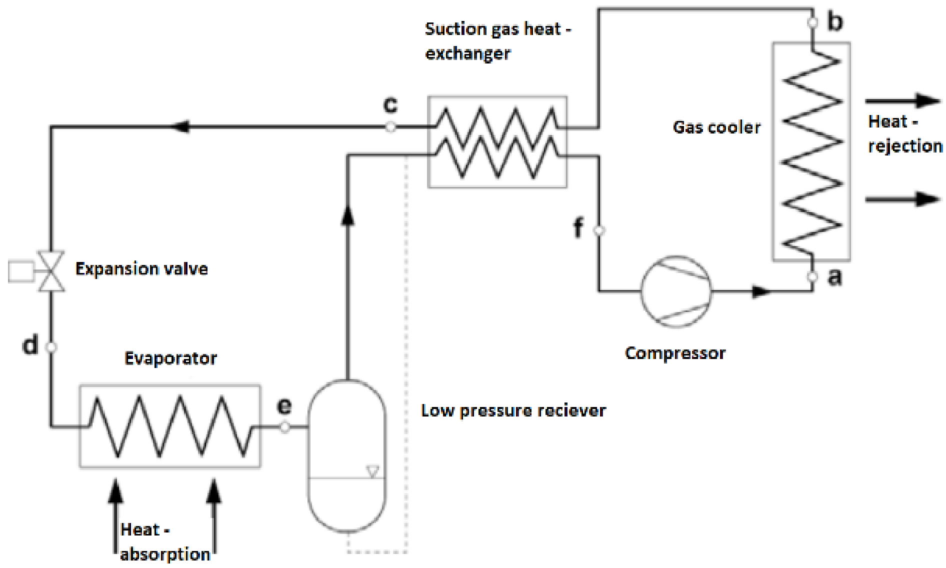

FIGURE 8: SIMPLE SKETCH OF A TRANSCRITICAL HEAT PUMP SYSTEM (HAUKÅS, 2016)

The p-h diagram displays three situations with different gas cooler pressures. The CO2 output temperature remains the same. The stippled lines depict constant temperatures and how they change according to pressure and enthalpy, 120°C to the right and 40°C to the left. The letters in the diagram are referring to the letters in the sketch, and describe the refrigerant state in-between the components.

- **a-b** - gas cooler
- **b-c** - high-pressure side of the SGHE
- **c-d** - expansion valve
- **d-e** - evaporator
- **e-f** - low-pressure side of SGHE
- **f-a** - compressor

It is prominent from the diagram, that the heat performance increases with the gas cooler pressure. The effect of the SGHE can be seen by the reduction of enthalpy in **c-b** and increase of enthalpy (superheat) in **e-f**. The p-h diagram also displays that a CO2 gas cooler outlet temperature close to the critical temperature greatly reduces the potential temperature glide, which reduces the COP.

The transcritical cycle operates with a greater pressure difference between the heat absorption and -rejection than the common subcritical cycle. However, despite the pressure difference being greater, the pressure ratio is lower. Some systems can operate with pressure ratios up to eight, the CO2 systems typically operates at around three or four. A lower pressure ratio means that the compressor can work with a greater efficiency. This is displayed in Figure 9. The figure shows a diagram comparing the compressors efficiency at different pressure ratios for R717 and R22 at different rpms. The effect is most prominent at the lines representing the compression at 725 rpm, where the efficiency is reduced with a higher pressure ratio. (Chapter 4, Compressors)

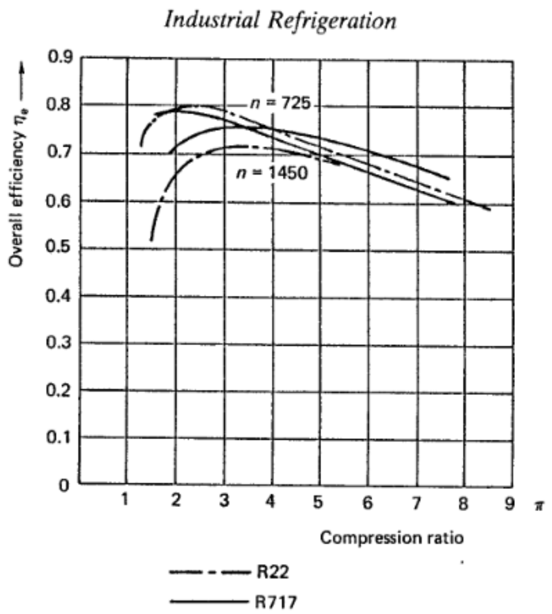

FIGURE 9: OVERALL COMPRESSOR EFFICIENCY AT DIFFERENT COMPRESSOR RATIOS (CHAPTER 4, COMPRESSORS)

Relying on sensible cooling rather than the rejection of latent heat from condensation, allows for a high gas cooler temperature, making transcritical CO2 suitable for systems requiring a high temperature heat supply. One such system is domestic hot water (DHW) heating. Being able to deliver hot water at 60-95℃, the CO2 heat pump eliminates the requirement for supplementary heating, which is often required when dealing with a conventional system.

2.4.3.3 Optimal gas cooler pressure

CO2 has a close to ideal temperature profile compared to water. However, it is important to dimension the gas cooler accordingly. Figure 10 and Figure 11 shows the temperature glide in a gas cooler at two different gas cooler pressures. The difference in CO2 outlet temperature and water inlet temperature is Δt_a. Δt_a is seen as a measurement for how well the installation is dimensioned for transcritical operation. A high Δt_a means a high temperature drop across the expansion valve, indicating large expansion losses which reduces COP. High Δt_a is often a symptom of low gas cooler pressure. By correctly dimensioning the gas cooler and adjusting the gas cooler pressure, Δt_a can get as low as 2-4K. The Δt_a displayed in Figure 11 shows that the figure represent an installation well suited and dimensioned for the trans critical process (Haukås, 2016). Table 8 lists the optimal gas cooler pressures for DHW heating by CO2 heat pump at different DHW temperatures and the respective COP. The information presented in table 1 is based on an evaporation temperature (Stene, Karbondioksid (R744) som arbeidsmedium i varmepumper TEP16, 2018).

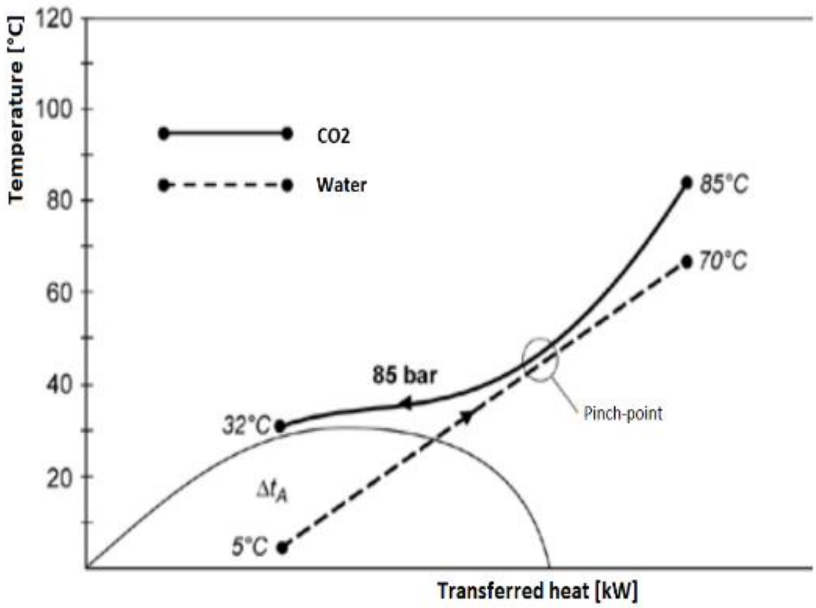

FIGURE 10: TEMPERATURE GLIDE IN THE GAS COOLER IN HOT WATER HEATER AT LOW GAS COOLER PRESSURE (HAUKÅS, 2016)

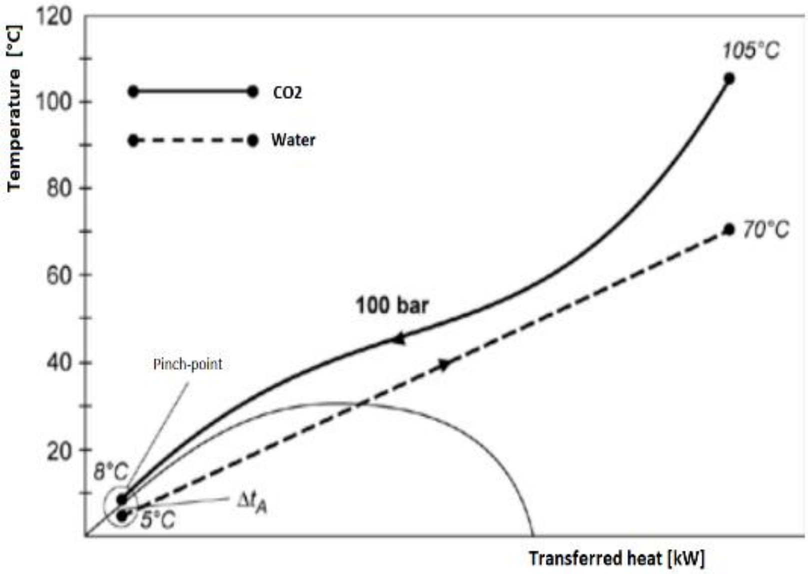

FIGURE 11: EXAMPLE OF CO2- GAS COOLER TEMPERATURE GLIDE WITH PINCH POINT AT THE GAS COOLER OUTLET. (HAUKÅS, 2016)

TABLE 8: OPTIMAL GAS COOLER PRESSURE AT DIFFERENT HOT WATER TEMPERATURES

Hot water temperature [°C]	Optimal gas cooler pressure [bar]	COP
60	90	3.8
70	100	3.58
80	110	3.09

2.4.3.4 Evaporation temperature

The evaporation temperature, t_e, is crucial to the performance of a heat pump. The p-h diagram in Figure 4 shows that t_e decides the pressure on the evaporator side. Assuming a fixed gas cooler pressure, a lower evaporator pressure means a higher pressure ratio. Figure 9 shows that a higher pressure ratio means a lower compressor efficiency. A lower t_e also increases the temperature lift produced by the compressor. An increased temperature lift means a more heat input, W, from the compressor. Equation (7) shows that an increase in W reduces COP. It is reckoned that an increase in t_e of 1 K, increases COP by 2-3% (Stene, Thermodynamikk for varmepumpeprosessen - TEP4260, 2018). However, the evaporation temperature should not be too high. Figure 12 shows that higher t_e restricts the potential evaporation enthalpy.

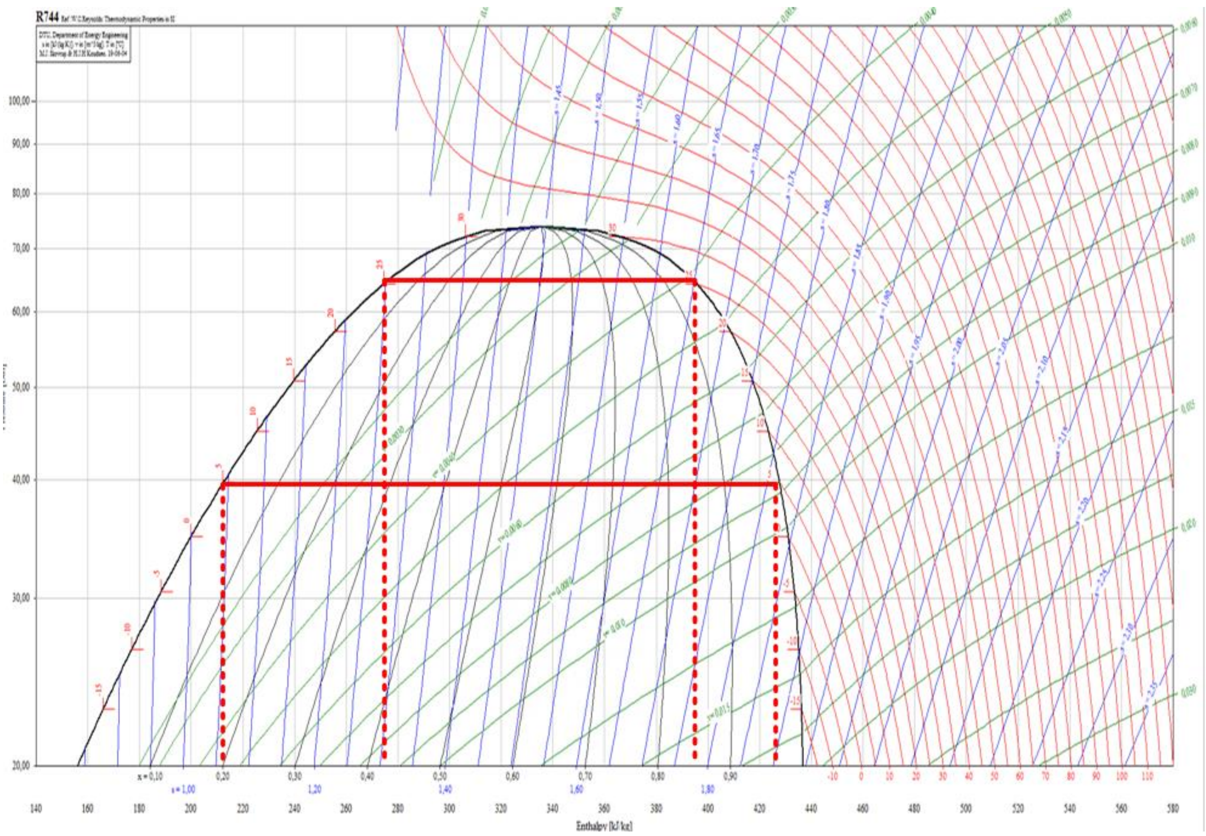

FIGURE 12: EVAPORATION ENTHALPY AT DIFFERENT TEMPERATURES

2.4.5 Ammonia as a refrigerant

Ammonia, NH3, has together with CO2 seen an increase in popularity the last few years, partly because of their excellent thermophysical properties, but also since they are natural with no environmental impact.

2.4.5.1 Thermophysical properties

Ammonia is known for its excellent thermophysical properties. This, combined with its favorable environmental aspects, has made it quite popular in the last few years. To show how ammonia stand out, a few its thermophysical properties is listed in Table 9 together with a few other common refrigerants.

TABLE 9: PROPERTIES OF R717, R744, R-134A AND R290 (STENE, COMPRESSION SYSTEMS WITH NATURAL WORKING FLUIDS)

Property	R717	R744	R-134a	R290
Molar mass [g/mol]	**17,03**	44,01	102	44,0
Evaporation enthalpy [kJ/kg]	**1262,7**	231,6	198,4	377,3
Thermal conductivity, liquid [W/mK]	**0,539**	0,105	0,092	0,105

Thermal conductivity, gas [kJ/kg]	**0,022**	0,023	0,012	0,023
Critical temperature [°C]	**132,4**	31,1	101,06	97
Critical pressure [bar]	**112,8**	73,9	40,6	42,3

With its high critical point of 132,4 °C, ammonia is very applicable. It can used for several different applications, as its range for condensation and evaporation temperature is large. In addition, its low molar mass contributes to a higher efficiency because it leads to a very high evaporation enthalpy. Refrigerants such as ammonia, with a low molar mass, are characterized by a higher specific volume, and thus a relatively low density. A low density means that the tubes can be smaller in diameter as well as smaller valves, which will contribute to a lower pressure loss. The table also shows that ammonia has a high thermal conductivity in liquid form, which leads to superior heat exchange in evaporator and condenser. It is also important to mention that the high liquid thermal conductivity will lead to an effective heat exchange in the condenser and evaporator.

2.4.5.2 Pressure enthalpy process

Ammonia can further be described by looking at the p-h diagram. The p-h diagram for ammonia is shown in Figure 3. The p-h diagram shows that the mixed state area of ammonia is relatively large with a high critical point and a generally large enthalpy difference between the saturation lines. The high critical temperature and pressure grants the possibility for a wide selection of condensation and evaporation temperature. The width of the mixed state area is also an important factor as it increases the potential for accumulated enthalpy through the evaporator, as well as the potential for rejecting heat through the condenser.

2.4.5.3 Safety aspects

Arguments against the installation of ammonia heat pump systems are often tied to its toxicity and potentially flammable tendencies. The gas is irritating to the body's skin and eyes and can at high concentrations be lethal. In liquid form, ammonia is highly corrosive which leads to burns when it is in contact with skin. *Guidelines for Design and operation of Compression Heat Pump, Air Conditioning and Refrigerating Systems with Natural Working Fluids* provide the following safety measures (Stene, Compression Systems with Natural Working Fluids)

- Placing the machine room on the top floor or on top of the building to prevent /minimize dispersion of ammonia vapour to the public.
- Construction of a gas tight machine room that prevents the spread of ammonia to populated parts of the building in case of leakages.
- Sufficient numbers of fireproof and self-closing doors opening outwards as emergency exits.
- Fireproof walls, floor and ceiling.

- Ammonia leakage detectors to detect unusually large amounts of ammonia in the machine room.
- Installation of a failsafe ventilation system that disperse of the ammonia gas.
- Installation of ammonia absorption systems which are based on ammonia being especially prone to absorption in water. These systems are often called scrubbers and are connected to the ventilation duct where sprinklers shower the ammonia filled air to filter out the toxic gasses.
- Remote manual operation and shut down of the plant ant ventilation system from outside the machine room.
- Emergency lighting and availability of fire extinguishers and personal safety equipment. Safety equipment should preferably be placed outside the machine room.

2.5 Hot water storage

Energy can be stored in many ways, shapes and forms such as in a dam, electricity, or as heat. This project will only concern itself with the latter as the objective is a local thermal grid where the goal is an energy efficient solution for the ventilation and heat demands.

As the regulations for buildings are getting more and more strict energy-demand-wise, as does the relevance of energy storage. The reason for this is that the storing of energy makes it possible to not only save energy for when it is really needed, but also “collect” waste heat and re-use it, decreasing the overall need for grid-energy in the first place. A good example of an application where energy storing is extremely valuable is solar energy, whether it is PV-panels or Solar Thermal collectors. The production/collection of solar energy is at its peak during the day. However, this is also when the need for heat and energy is at its lowest. Therefore, if the produced energy is stored and available for use in the evening, when the sun is down and the heat demands increase, the value of the solar application will increases heavily. Energy storage based on the daily demands such as this, is called diurnal storage. It is also possible to think more long-term, and save energy produced in the summer for use in the winter, when the weather is cold. This is called seasonal energy storage.

2.5.1 Stratified TES tank

Because of its low cost and simplicity, the stratified TES tank is one of the most common diurnal TES. Stratification of water is a natural process where the difference in temperature creates a difference in density, which naturally divides the water. Since warm water has a lower density than cold water, warm water will always float to the top, while cold water will stay on the bottom. In order to maximize the effectiveness and functionality of such a tank, it should be constructed according to three main principles. (Ibrahim Dincer, 2011)

1. *The tank should be stratified*. This means that it should be able to separate volumes of water with different temperatures and avoid any mixing between

them. To improve the thermal stratification, it is desirable to have a deep tank. In addition, it is important that the inlet and outlet is positioned in such a manner that the flow between them is uniform, minimizing mixing of the different tempered water layers. To best accomplish this effect, the inlet and outlet should be placed as close as possible to the top and bottom, respectively.
2. *Dead water zones should be minimized.* If the inflowing water is not properly mixed with the tank water, the stratification can cause a separation that renders parts of the tank water still standing, and thus inaccessible. This is a problem, and the solution lies in the placement of the inlet and outlet, as displayed in Figure 13. The last principle is that heat loss/gain should be held to a minimum
3. *Heat loss/gain should be held to a minimum.* (Ibrahim Dincer, 2011) (P. Armstrong, 2013)

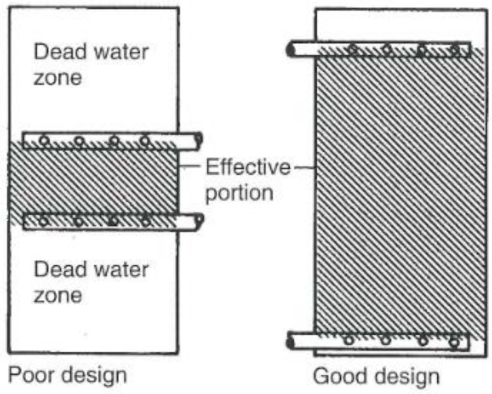

FIGURE 13: A POOR DESIGNED AND A GOOD DESIGNED TES WATER TANK WITH REGARDS TO DEAD WATER.

An important aspect of the storage of hot water in stratified water tanks, is its reputation for occasionally being the breeding ground for pathogens such as legionella. The danger of legionella is that it, if inhaled, can cause Legionaries disease, which is a form of pneumonia. Legionella resides in water with temperatures ranging from 25 to 45°C. Above 49°C sterilization occurs exponentially, and temperatures exceeding 60°C has a definitive bactericidal effect. The World Health Organization (WHO) thus recommend water to be stored at 60°C. However, according to a study conducted in Quebec, even when the thermostat is set at 60°C, approximately 40% of water heaters remain contaminated (Benoît Lèvesque, 2004).This is a result of stratification. Even though the thermostat is set at 60°C the bottom of the tank can still be 30-40°C, leading to the proliferation of legionella. One option is destratification the tank, making sure every part of the tank is at a safe temperature. A study performed at the Oxford University explored this very option. The destratification lead to a homogenous temperature of 54°C which should be sufficient to sterilize Legionella. However, reducing the temperature also lead to a reduction in exergy and a 19% loss in usable hot water. (Benoît Lèvesque, 2004) (P. Armstrong, 2013)

2.5.2 Hot water storage system

In buildings with a large need for hot water storage, hot water tanks are often connected in series. It is important to design such a system in a manner that maintains the natural stratification through the series of tanks. Figure 14 displays a well-designed hot water tank storage system. The figure can be explained by describing the system as operating in two modes: charging and discharging.

Charging:

The heat pump is supplied with cold city water from point 1 in Figure 14. After heating the water, the heat pump sends it to point 2. On its way to point 2 the water passes through coils in the backup re heater. At point 2 the hot water enters the top of the tank furthest to the right. As hot water enters at the top of the tank, cold water exits the tank furthest to the left. As the system keeps charging, the tanks will fill up with hot water from point 2 while cold water exits at point 3. This way, even though the tanks are separate, maintain stratification according to one another is possible.

Discharging:

When discharged, hot water exits at point 2 and flow to point 4 where it is mixed with cold city water to achieve the correct tap water temperature of 55°C. As hot water exits at point 2, cold city water enters at point 3. The black lines at the bottom of each tank are diffusers. The diffusers absorb the pressure and velocity of the water entering at the bottom of the tank and prevent mixing. Since the tanks are stratified, the water entering at the bottom will naturally have a lower temperature than that at the top. If the water enters the tank freely, it might mix with the hot water, distort the stratification and reduce the exergy and the amount of usable hot water.

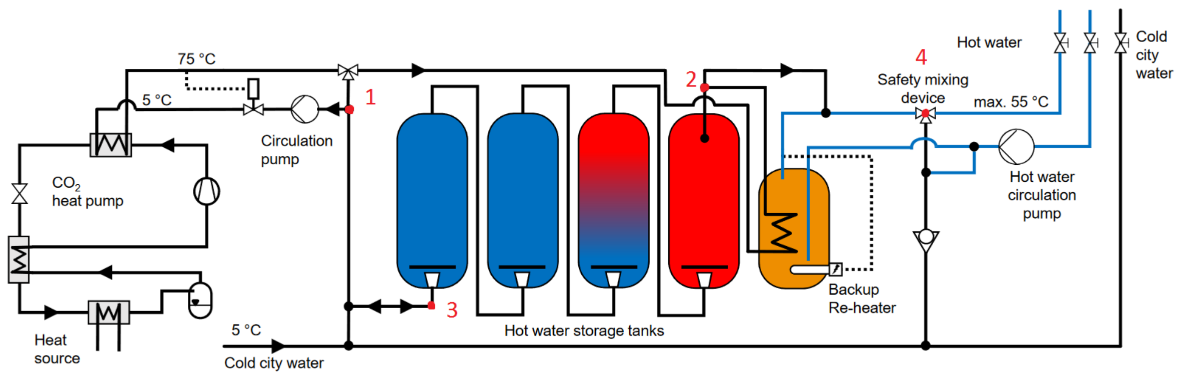

FIGURE 14: HOT WATER STORAGE SYSTEM WITH STRATIFIED WATER TANKS (HAFNER, 2018)

2.1.2 The principle of hot water tank sizing

There are several methods for determining the required capacity of the hot water storage. Geir Eggen, senior engineer at COWI AS, presented a way of sizing the hot water tanks based on the daily consumption profile at Norsk Kjøleteknisk Møte 2019. (Eggen, 2019). Figure 15 is a direct copy from the presentation. The blue line shows the accumulated use of hot water while the red line shows the accumulated production. The minimum required accumulation volume equals the largest difference between the production and consumption. In Figure 15, this point is at 14 where the difference between the production and consumption is $5m^3$. The volume of the accumulation tanks must therefore be at least $5m^3$.

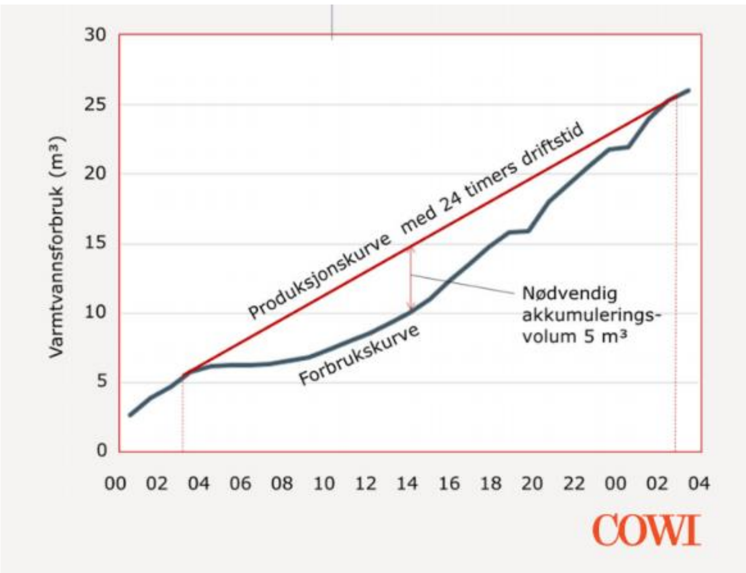

FIGURE 15: SIZING PRINCIPLE OF HOT WATER STORAGE TANKS (EGGEN, 2019)

3. Methodology

3.1 Leangen Sports facility

When researching available waste heat for the residential area, Leangen Sports Facility was recognized as a potential source. Leangen Sports Facility is a large facility containing indoor football and handball courts as well as a few skating rinks. To keep the skating rinks cold the facility uses an ammonia heat pump that removes the excess heat. After speaking to a contact person working with the facility, it was discovered that large portions of this heat are transported to dry coolers on the roof where it is rejected into the ambient. It is this skating rink that lays the foundation as the main source for waste heat in this project. Table 10 displays the monthly and annual waste heat rejected through the dry coolers in 2018.

TABLE 10: WASTE HEAT PRODUCED BY LEANGEN SPORTS FACILITY IN 2018

Month	Dry cooler WM16 [kWh]	Dry cooler WM13 [kWh]	Available waste heat [kWh]
Jan	152375	312182	464557
Feb	138010.51	324387.88	462398.39
Mar	156585.38	229856.45	386441.83
Apr	156246.41	55313.43	211559.84
May	154009.49	0	154009.49
Jun	140228.2	8	140236.2
Jul	0	50.78	50.78
Aug	135447.89	125098.17	260546.06
Sep	158921.3	108428.18	267349.48
Oct	175708.5	74365.66	250074.16
Nov	173865.52	629286.16	803151.68
Des	161523.66	590217.84	751741.5
ANNUAL	**1702921.86**	**2449194.55**	**4152116.41**

3.2 Representative building

To best assess the potential for exploiting the waste heat coming from Leangen Sports facility, it is important to set the parameters. The building description regarding the Leangen area states that there will be buildings of varying sizes ranging from 1 to 8 floors, with a mean apartment size of 70 m^2 (Lund Hagem Arkitekter AS, 2018). Because of the varying building sizes, it has been decided to base the project on a fictional building that represent the assumed average size. This will simplify the project when performing the calculations regarding heating, cooling and hot water demands. A simple description of the building dimensions is listed in Table 11. It is stated in the building description that there will be 1660-1770 apartments, which means that the area will have the equivalent of 86 average buildings. (Lund Hagem Arkitekter AS, 2018)

TABLE 11: DESCRIPTION OF THE AVERAGE BUILDING ON LEANGEN

Average building	
Length [m]	23,7
Width [m]	11,8
Base area [m^2]	280
Area per apartment [m^2]	70
Number of floors	5
Apartments per floor	4
Separating floor thickness [m]	0,6
Inside wall height [m]	2,4
Total heated area [m^2]	1400
Total window/door area [m^2]	280
Inhabitants per building	40

3.2.1 SIMIEN

To get an approximation of the energy demand of the average building, it was decided to use a computer program called SIMIEN. SIMIEN is a simulation tool for calculations of energy demand and evaluations of the indoor climate in buildings. It is often used to perform assessments of buildings against building standards, energy labeling, annual energy demand and the dimensioning of heating, ventilation and cooling appliances. Because of the lack of information regarding the building plans, the base values during the simulation was set according to TEK 17. Table 12 lists the values on which the simulations are based upon (Direktorat for byggkvalitet, 2017). The values are set a bit below that of TEK17. This was done under the assumption that this will be a state-of-the-art project and that the building properties will surpass the standards. For comparison, the minimum demands for passive houses are listed as well. (Myhre, Schild, Pettersen, Blom, & Gullbrekken, 2012)

TABLE 12: BASE VALUES FOR ENERGY-DEMAND-SIMULATIONS IN SIMIEN.

Value of base measure	SIMIEN	TEK 17	NS 3700

U-value outer wall $\left[\frac{W}{m^2K}\right]$	**0.15**	≤ 0.18	≤ 0.15
U-value roof $\left[\frac{W}{m^2K}\right]$	**0.13**	≤ 0.13	≤ 0.13
U-value floor $\left[\frac{W}{m^2K}\right]$	**0.09**	≤ 0.10	≤ 0.15
U-value windows and doors $\left[\frac{W}{m^2K}\right]$	**0.60**	≤ 0.80	≤ 0.80
Window/door area compared to heated are [%]	**20**	≤ 25	-
Annual temperature efficiency of ventilation heat exchanger [%]	**80**	≥ 80	≥ 80
Specific fan power in ventilation $\left[\frac{kW}{m^3s}\right]$	**0.5**	≤ 1.5	≤ 1.5
Air leakage per hour at 50 Pa pressure difference	**0.4**	≤ 0.6	≤ 0.6
Normalized thermal bridge value $\left[\frac{W}{m^2K}\right]$	**0.06**	≤ 0.07	≤ 0.03

In addition to the set values in Table 12, there are a few key assumptions regarding the simulation in SIMIEN. The assumptions are as listed below.

Cooling

- The building only requires cooling during the five hottest months of the year which are May, June, July, August and September.
- Cooling will be active at all room temperatures exceeding 22°C.
- Cooling is assumed to be active only between 10:00 – 19:00.

Heating

- The set point temperature will be 21°C during normal hours and 19°C during office hours.
- Office hours are set to be between 07:00 and 16:00.

3.3 Domestic Hot Water load profile

For more thorough calculations concerning the energy demands for hot water, it is crucial to have a daily domestic hot water (DHW) load profile. Since the residential area at Leangen is yet to be built, there is no information regarding the DHW load. The load profile is therefore based on the annual average daily DHW load profile in Germany, which is presented in Figure 16.

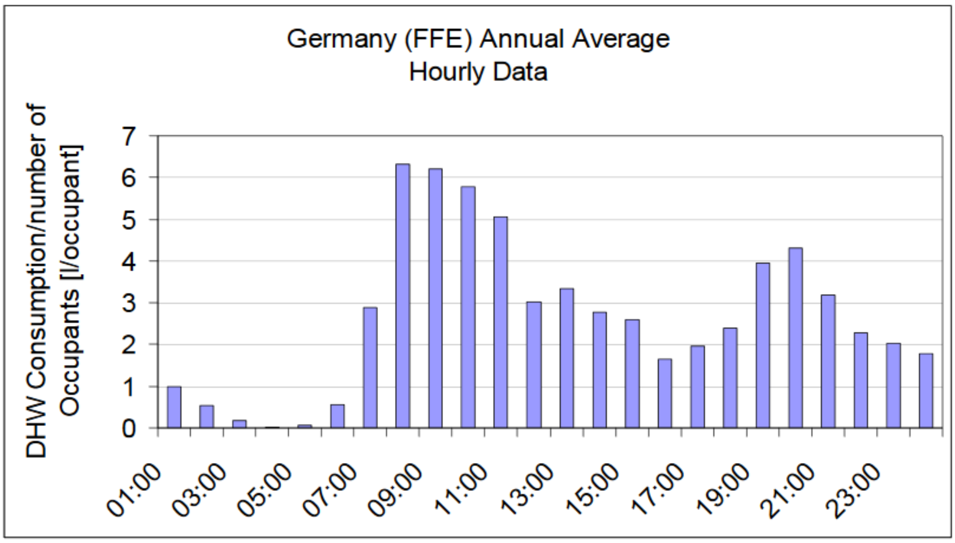

FIGURE 16: AVERAGE DAILY DHW LOAD PROFILE IN GERMANY (KNIGHT, KREUTZER, MANNING, SWINTON , & RIBBERINK, 2007)

3.4 Low Temperature Thermal Grid (LTTG)

After reviewing the data from Leangen Sports Facility it has been decided to use the excess heat as the source for a LTTG, supplying the residential buildings with heat. The LTTG falls under the category of 4GDH and will supply water at $40°C$. Figure 17 shows a sketch of the low temperature thermal grid. The thick red line represents the supply line from the dry coolers, containing hot water. The thick blue line is the return pipes containing the cooled water that is sent back to the sports facility to be reheated. Each building is connected to both the supply line and the return line. This allows for every building to dump the cooled supply water directly into the return line. Making it possible for all buildings to be supplied by an approximately equal supply temperature. If not, the temperature would drop significantly after each building, leaving no heat for the buildings at the end of the line.

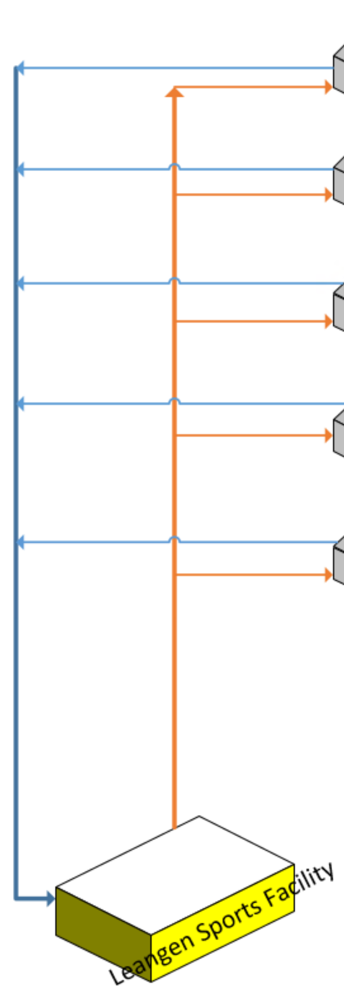

FIGURE 17: LOW TEMPERATURE THERMAL GRID

3.5 Energy distribution system

The project includes a LTTG for distribution of energy from the designated source to the individual building. The LTTG is connected to a local energy distribution system at each building designed to supply the inhabitants with energy to cover their respective energy demand. Primarily, the focus of this project is on this energy distribution system for the individual building. At the moment, the only reliable source of waste heat is the skating rinks. The initial energy distribution system is therefore designed to rely only on the skate rink waste heat for its space heating and domestic hot water. The proposed systems are designed according to two main principles.

- All space heating will be covered by waste heat through the LTTG.
- All DHW is produced by the CO2 heat pump.

3.5.1 Space heating

Since the LTTG supplies hot water, it is decided to link this to a floor heating system in the residential buildings. Chapter *2.3 Hydronic heating and cooling systems* state that the temperature circulating in such systems should have a temperature of $35 - 40°C$. Since the supply water from the skating rink is $40°C$, it is decided to connect the heat system directly to the LTTG.

3.5.2 Domestic hot water heating

One of the main principles of the buildings' energy distribution system is that all DHW is covered by a CO2 heat pump. The heat pump will mainly draw energy from the LTTG, further exploiting the waste heat produced by the skating rink. It is also decided to explore the possibility of connecting the heat pump to the cooling system. The excess heat in the building in the summer time is a heat source that is both available and exploitable. That way the system makes use of the heat pumps COP as it covers both cooling and DHW.

3.5.3 Sizing of the accumulation tanks

The sizing of the hot water accumulation tanks will be performed after the principle presented in chapter *2.1.2 The principle of hot water tank sizing.* This method compares the load profile of the day of largest consumption with the hot water production and gives a minimum requirement of accumulation volume.

3.6 Greywater potential

The basis for this project is to cover the heating demand for Leangen by exploiting waste heat sources. The most prominent source of waste heat is the Leangen Sports Facility. However, if this does not suffice, other sources must be evaluated. One such source is greywater. The average water use is per person is 136-150 litres per day (Mazhar, Liu, & Shukla, 2018). Substantial amounts of the water use end up as wastewater which is dumped into the sewer. Part of this wastewater still holds relatively high temperatures. This means that large parts of the heating required for DHW ends up in the drain. Greywater is thus an untapped source of waste heat which is readily available considering that it resides inside the domestic household. Table 2 shows that the wastewater from showering alone amounts to an average of 54.32 litres per day per person and hold an average temperature of 33.5 $°C$ as it flows down the drain. Equation (1) shows that by collecting this water and extract enough heat to reduce the temperature by $25K$, it can provide 1.58 kWh per day.
The average building has 20 apartments. Assuming 2 inhabitants per apartment, the shower wastewater alone can provide 63.2 kWh per day per building. The downside to greywater is that it is highly unpredictable. One way of circumventing this problem is to collect only the most reliable and consistent greywater sources. Equation (1) shows that a higher temperature of the wastewater means more energy available for extraction. The optimal solution would then be to collect greywater from the sources that are both reliable as well as have a high wastewater temperature. As proposed in chapter *2.2.2 Greywater heat recovery,* the most reliable greywater source when considering use and temperature, is shower-, washing machine- and dishwasher-wastewater. As displayed in

Table 6, these sources amount to an average of 85.4 l per day per person and holds a combined average temperature of $35.32°C$.

The easiest way of extracting the available heat in greywater is to transfer it to a more versatile fluid which can be easily manipulated. The space heating concept in this project is to use waterborne floor heating. One possible solution could be to use the greywater to heat the water circulating in the floor. Greywater production and temperature is unpredictable which means using it for SH is a gamble. Also, the sediment in the wastewater will most likely clog the waterpipes. However, since the LTTG water is a steady $40°C$, using this for floor heating would be easier and more practical. The remaining option is then to use greywater as a heat source for DHW production.

3.7 Dymola

Dymola is a simulation tool that is based on the modelling language Modelica. The software revolves around assembling computerized versions of physical systems by connecting virtual components that interact virtually like they would physically. This allows for a computation and analysis of a system physically having one at hand. Dymola is, in this project, used by way of designing and analysing the CO2 heat pump for DHW heating.

The default library in Dymola was inadequate, so an additional add-on called TIL Suite was installed. TIL Suite is an additional library containing components inhabiting the thermophysical properties relevant for the simulation of a heat pump system.

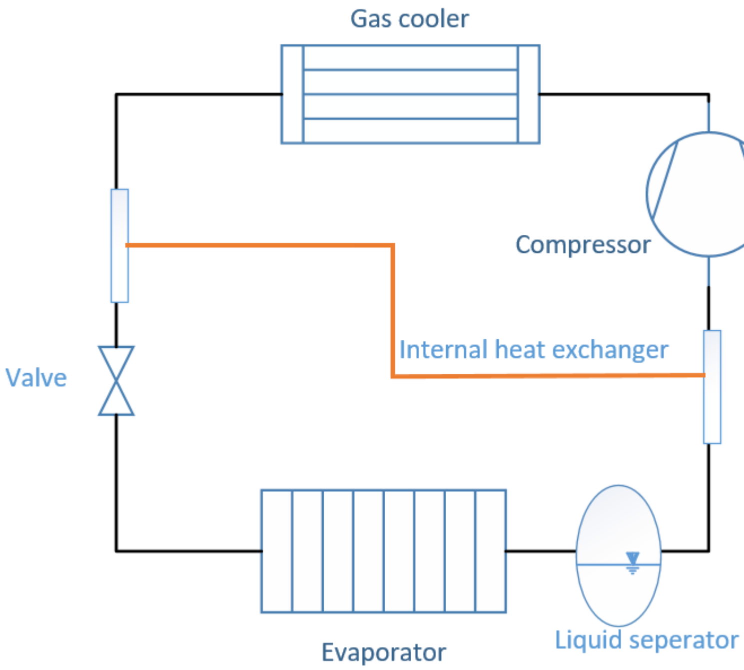

FIGURE 18: CO2 HEAT PUMP FOR DHW PRODUCTION

Figure 18 shows the principle sketch of the CO2 heat pump. The evaporator receives heat from the designated heat sources and transfers it to the CO2. The CO2 exiting the evaporator should ideally be pure gas but will most likely contain traces of liquid. The separator is installed to separate the liquid from the gas, making sure that the coolant entering the compressor is pure gas. The compressor increases the pressure of the coolant as it is sent to the gas cooler. With its high pressure and enthalpy, the CO2 can sufficiently heat the city water to the desired temperature. After exiting the gas cooler, the CO2 passes through a SGHE. This reduces the enthalpy and temperature of the CO2 before it is expanded and is ready for the evaporator again. With a lower temperature and enthalpy, the evaporator has an increased ability the absorb energy. The heat removed from the coolant before the expansion valve is transferred to just before the compressor, increasing the temperature of the CO2 as it enters the compressor. This provides super heat, further reducing the chance for liquid in the compressor. More superheat also increases the gas temperature which reduces the necessary heat input by the compressor, increasing COP.

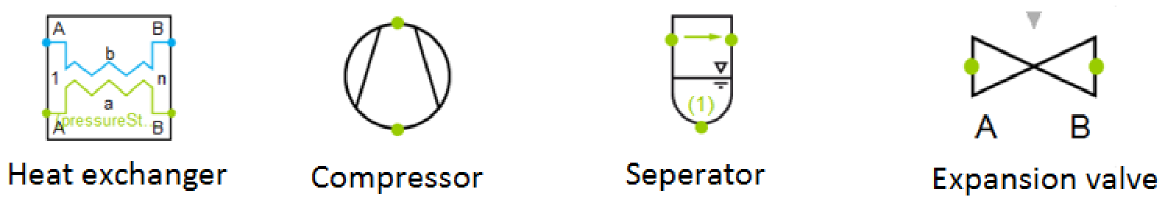

FIGURE 19: DYMOLA ICONS REPRESENTING THE MAIN HEAT PUMP COMPONENTS

Figure 19 shows how TIL-suite displays the main components of a heat pump. The coloured dots on the components shows the connector-points. Green colour means VLE-fluid (Vapour-liquid equilibrium), while blue means liquid. The heat pump in question is a CO2 heat pump that heats DHW, thus green represents CO2 while blue represents water.

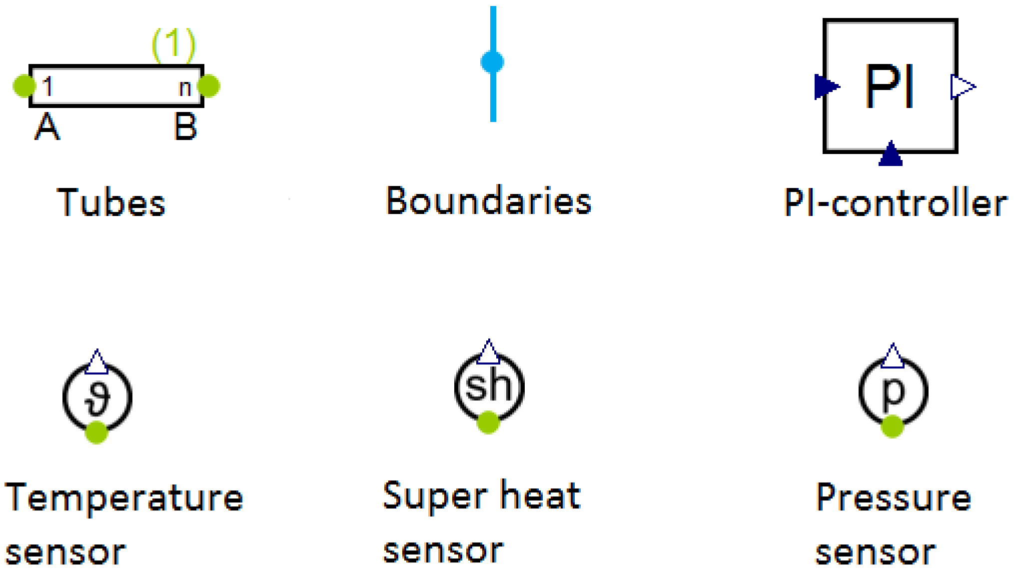

FIGURE 20: DYMOLA ICONS FOR ADDITIONAL COMPONENTS USED IN SIMULATION OF CO2 HEAT PUMP

Tubes
Figure 20 shows additional components necessary for a simulation and analysis of the complete heat pump system. The tube component can be designed to include the pressure drop which occurs in the pipes. In this case the pressure drop is set to zero, as it is expected to be relatively small. Two sets of tubes can also be connected to make an internal heat exchanger, which is their role in the simulated heat pump.

Boundaries
The boundaries are sources and sinks of fluids. The parameters of the boundaries can be instructed to inject and absorb mass flows of a chosen liquid. They are, in this case, the

components that create the water flow through the evaporator and the gas cooler. The temperature of the flowing water is also specified in the parameters of the boundaries.

PI-controller

The PI-controller is an instrument that controls the parameters of a component based on a desired value of another component. A common use of the PI-controller is to manipulate the speed of the compressor until the temperature or pressure at a specific point in the cycle reaches its set value. In this simulation the compressor is controlled relative to the high-pressure side. If the pressure on the high-pressure side is too low, the PI-controller increases the compressor speed, increasing the mass flow of CO2. An increased flow of CO2 means an increased pressure build up on the high-pressure side of the expansion valve. Likewise, it reduces the compressor speed if the pressure is too high. Thus, the PI-controller always aims to keep the pressure at its setpoint. For the PI-controller to achieve the desired pressure, it is connected to a pressure sensor on the high-pressure side. Figure 21 displays the model for the simulation in Dymola.

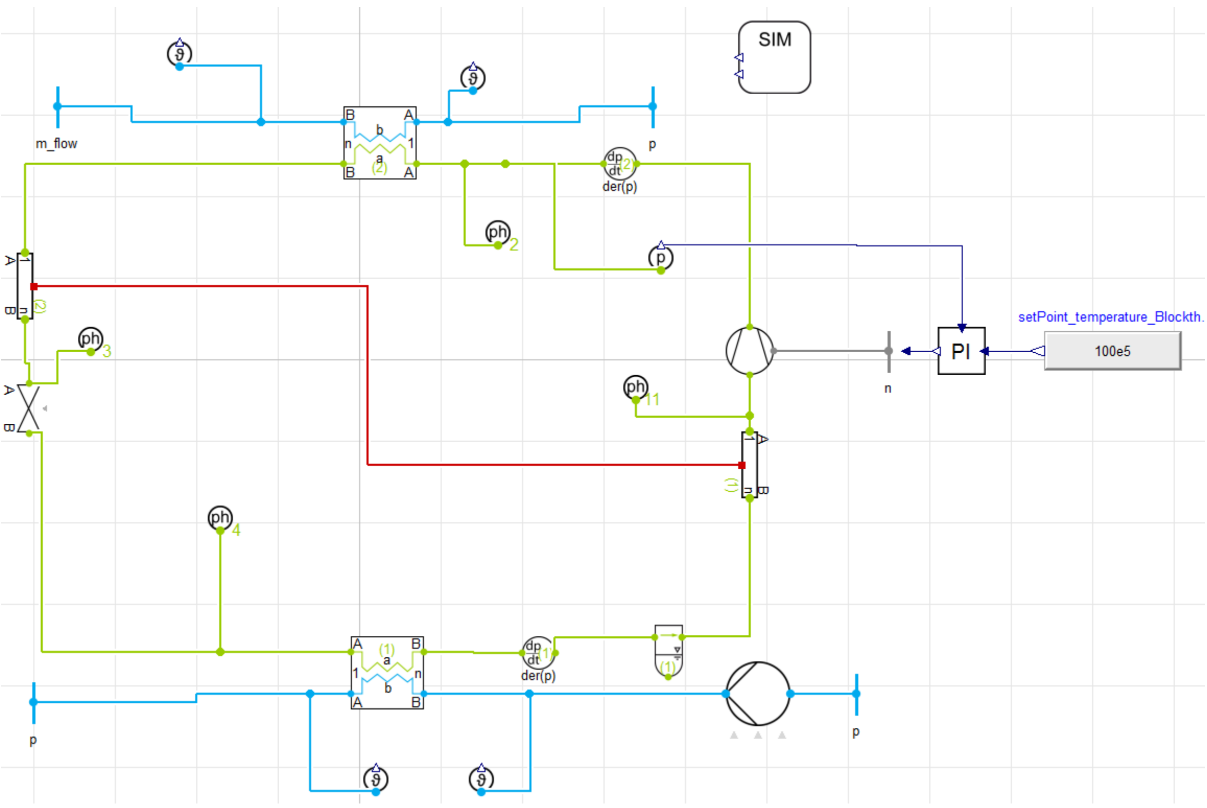

FIGURE 21: A DISPLAY OF THE SIMULATED HEAT PUMP IN DYMOLA

3.7.2 Design

Gas cooler

The priority when designing this heat pump is to make sure it produces enough hot water. This is achieved by calculating the necessary water flow through the gas cooler. The mass flow is calculated by equation (1) where Q is the daily hot water demand and ΔT is the temperature lift provided by the heat pump.

Next step is the dimensioning and sizing of the gas cooler. It is decided to go with the example geometry of the heat exchanger shown in Figure 22.

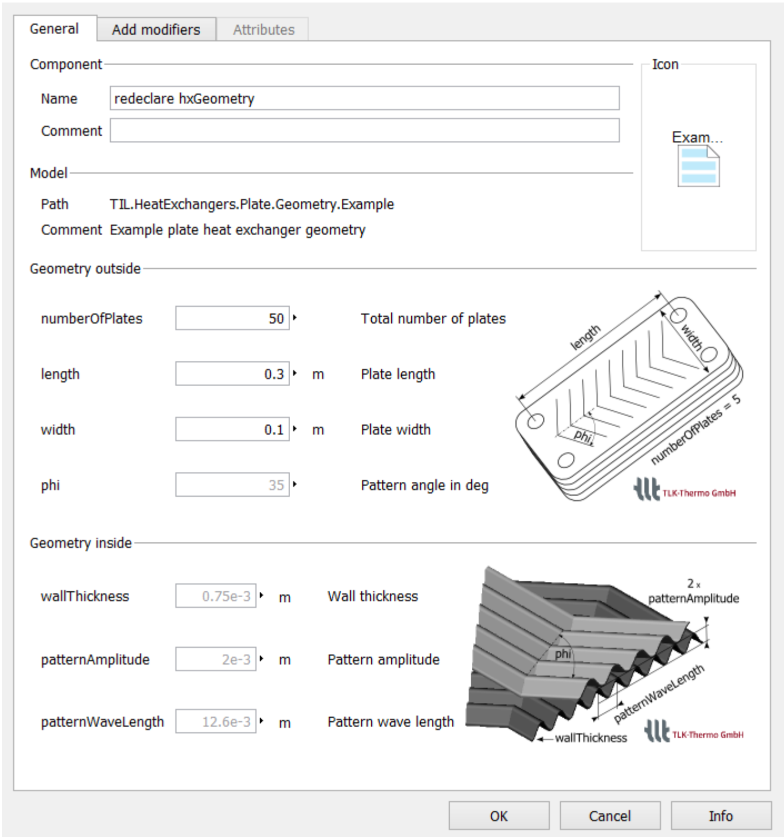

FIGURE 22: GAS COOLER GEOMETRY

Figure 22 shows that the heat exchanger has 50 plates with the size of 0.09 m^2. The total heat exchanger surface area is then $4.5m^2$.

It is decided to neglect the pressure drop in the gas cooler and it is therefore set to zero. The thermal transmittance of the gas cooler is based on a research rapport by Jørn Stene concerning a CO2 DHW heater. During the modelling of the system, U-values ranging from 1800-2400 W/K was used (Stene, High-Efficiency CO2 Heat Pump Water Heater Systems for Residential and Non-Residential Buildings). The gas cooler therefore has a U-value of 2000 W/K.

Evaporator

The geometry and thermal transmittance of the evaporator is chosen to be the same as the gas cooler, a surface area of $4.5m^2$ and a U-value of 2000 W/K. The water inlet temperature of the evaporator is set according to the chosen heat source. The water flow through the evaporator is likewise adjusted to create a temperature drop sufficient to maintain a constant exit temperature. The exit temperature of the evaporator water is important as it greatly influences t_e.

PI- regulator

As presented in Figure 21, the PI- regulator is set to control the compressor according to the high-pressure side. Table 8 shows that the optimal gas cooler pressure at $70°C$ DHW is 100 bar while optimal pressure for $80°C$ is 110 bar. Optimal pressure for $75°C$ is therefore assumed between 100 and 110 bar.

4. Results and discussion

4.1 Energy demand

4.1.1 SIMIEN

Table 13 shows the simulated monthly need for space heating (SH), space cooling (SC) and DHW heating for the average building.

TABLE 13: SIMULATED ENERGY DEMAND

Month	Heating demand [kWh/building]	Cooling demand per building [kWh/building]	DHW demand [kWh/building]	Total heating, cooling and DHW [kWh/building]
Jan	6160	0	3542.6	9636
Feb	4928	0	3119.7	8404
Mar	3850	0	3542.6	7326
Apr	462	0	3428.3	3938
May	0	1560	3542.6	5036
Jun	0	3080	3428.3	6556
Jul	0	3696	3542.6	7172
Aug	0	2926	3542.6	6402
Sep	154	616	3428.3	4246
Okt	2618	0	3542.6	6094
Nov	4870	0	3428.3	8346
Des	6545	0	3542.6	10021
TOTAL	**29587**	**11878**	**41711**	**83177**

The average building is mainly designed according to TEK 17, but Table 12 shows that most of the building properties also fit the minimum requirements for passive houses. (Myhre, Schild, Pettersen, Blom, & Gullbrekken, 2012). For a better comparison, the minimum heating demand for passive houses are listed in Table 14, while the simulated heating demands are listed in Table 15.

TABLE 14: MINIMUM REQUIREMENTS FOR PASSIVE HOUSES (MYHRE, SCHILD, PETTERSEN, BLOM, & GULLBREKKEN, 2012)

Parameters	Values
Space heating demand [kWh/m^2]	≤ 15
Hot water demand [kWh/m^2]	≤ 30
Total need for SH and DHW [kWh/m^2]	≤45

TABLE 15: HEATING DEMAND FROM SIMIEN

Parameters	Values
Space heating demand [kWh/m^2]	21.13
Hot water demand [kWh/m^2]	29.79
Total need for SH and DHW [kWh/m^2]	50.92

The tables show that even though the specifications fit that of a passive house, the SH demand is too large. However, the passive house requirements are derived from the German standards in the 1990s and do not take into consideration the difference between the Norwegian and German climate (Myhre, Schild, Pettersen, Blom, & Gullbrekken, 2012). Since the average Norwegian climate is considerably colder than the German climate, expecting the annual heating demand to fulfill these requirements is not realistic. In addition to the difference in climate, the individual habits of the inhabitants affect the heating demand tremendously.

4.1.2 Heating demand of a passive house in Trondheim

In 2013, measurements were performed on an actual passive house building in Trondheim. The load profile was a result of hourly measures of SH- and DHW delivered by district heating throughout the year. The monthly and annual SH- and DHW demand is listed in Table 16.

TABLE 16: DISTRICT HEATING SUPPLY FOR SPACE HEATING AND DOMESTIC HOT WATER IN A PASSIVE HOUSE IN TRONDHEIM

	DHW demand [kWh/m2]	SH demand [kWh/m2]	Total SH and DHW demand [kWh/m2]
Jan	2.60	5.95	8.54
Feb	2.35	4.49	6.84
Mar	2.60	4.14	6.74
Apr	2.51	1.70	4.21
May	2.60	0.45	3.05
Jun	2.51	0.15	2.66
Jul	2.60	0.15	2.75
Aug	2.60	0.10	2.70
Sep	2.51	0.37	2.88
Okt	2.60	1.86	4.46
Nov	2.51	3.41	5.92
Des	2.60	3.40	5.99
Annual	**30.57**	**26.17**	**56.75**

The measurements show that the actual energy demand surpassed the requirements for passive houses, even though the building in fact is classified as a passive house. Comparing the measured energy demand with the simulation, it appears that the DHW demand is very similar. The SH, on the other hand, is almost $5\,\frac{kWh}{m^2}$ lower in the simulation. This might indicate that the simulated values are better than what can be realistically expected. On the other hand, the building project will not start until 2023. This means that by the time the construction work starts, the building, in which the measurements were made, will be 10 years old. It is therefore reasonable to assume that the new buildings will be a result of newer and better technology, and a lower energy demand can be expected. Thus, it would be reasonable to use the simulated values as a basis for the calculations of this project. However, the simulations claim no need for heating between May and August. The measurements show this to be an unrealistic expectation. It is therefore decided to use the measured heating demand of the passive house form 2013 as a basis for the calculations in this project. By the absence of data concerning the need for cooling, the cooling demand from SIMIEN is used.

4.1.2 Domestic hot water load

Chapter *2.1.1 Stratified TES tank* explains that to reduce the chances of proliferation of legionella, the minimum temperature in a stratified hot water tank should be $60\ °C$. However, the hot water temperature is not only important because of the dangers of Legionella. The storage temperature of hot water greatly affects the volume of the hot water load. A higher temperature means more energy stored per kilogram of water. Thus, a higher temperature allows for a reduced storage capacity. Table 17 presents the average demand for hot water production. Figure 23 presents the hourly production volume of the daily DHW demand from Table 17 at storage temperatures ranging from 60 to $85°C$. The values presented in the figure is calculated using equation (2), where ΔT is the temperature difference between the city water and the hot water storage.

TABLE 17: MONTHLY AND ANNUAL AVERAGE OF THE DAILY DHW DEMAND

	Daily average DHW demand per building [kWh]
Jan	117.21
Feb	117.27
Mar	117.31
Apr	117.32
May	117.26
Jun	117.40
Jul	117.28
Aug	117.31
Sep	117.26
Okt	117.21
Nov	117.36
Des	117.27
Annual daily average	**117.29**

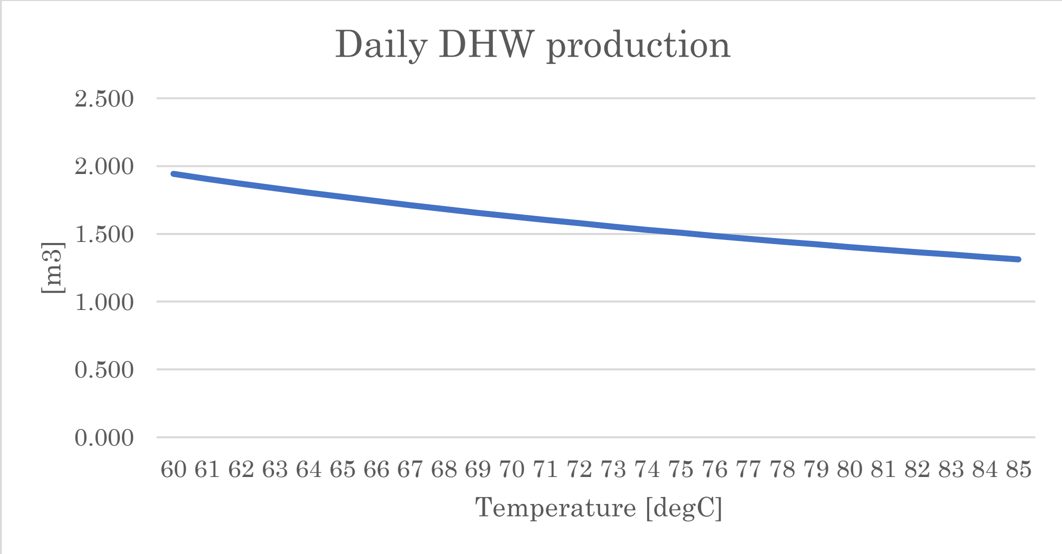

FIGURE 23: AVERAGE DAILY DHW PRODUCTION AT DIFFERENT STORAGE TEMPERATURES

It is decided to have a hot water storage temperature of $75°C$. This allows for 500 litres less storage than at $60°C$, while providing more safety regarding legionella.

The DHW load profile is based on the average daily consumption in Germany which is shown in Figure 16 (Knight, Kreutzer, Manning, Swinton , & Ribberink, 2007). The graph displays a daily consumption of 64.9 litres per day per person. From the data concerning the passive house in Trondheim, the average consumption requires a daily production of 1.507 m^3 hot water per building. Assuming 40 inhabitant per building, this equals 37.68 litres per day per person. The hourly values from the German load profile is multiplied with $\frac{37.68l}{64.9l} = 0.583$. This scales the consumption down to fit the expected DHW demand in Trondheim while fully adopting the hourly load profile. The resulting load profile concerning the average building at Leangen is presented in Figure 24.

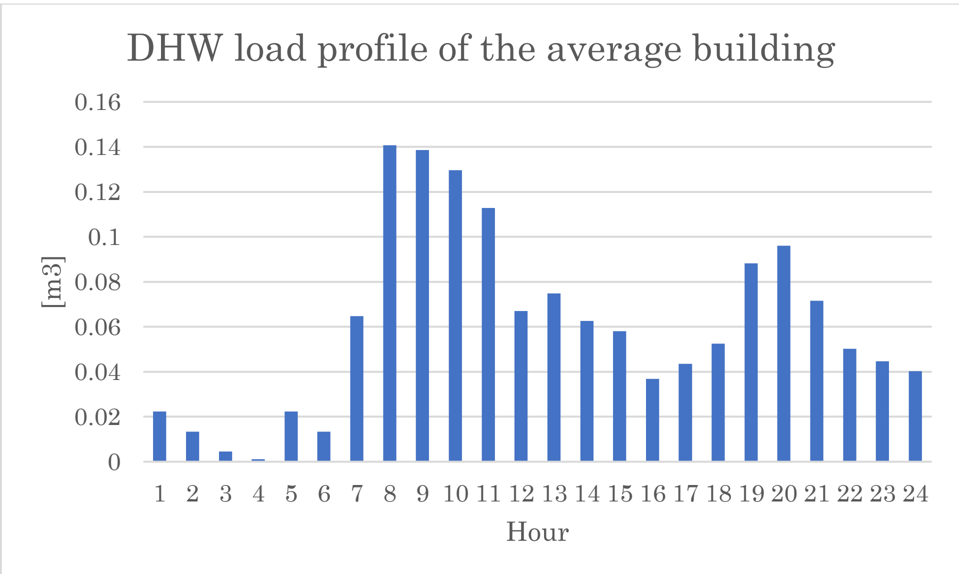

FIGURE 24: DHW LOAD PROFILE OF THE AVERAGE BUILDING

4.2 Initial distribution system

Figure 25 on the next page displays the energy distribution system on a building level. The different components that make up the system are explained and discussed in the upcoming subchapters. Chapter *3.5 Energy distribution system* explains that the system is designed with an emphasis on simplicity. The figure therefore shows that the system only has two connections to the LTTG. One to the supply line and one to the return line. The idea is to create kind of plug-and-play system so that when the building is finished, the system can simply plug into the LTTG and be ready to run.

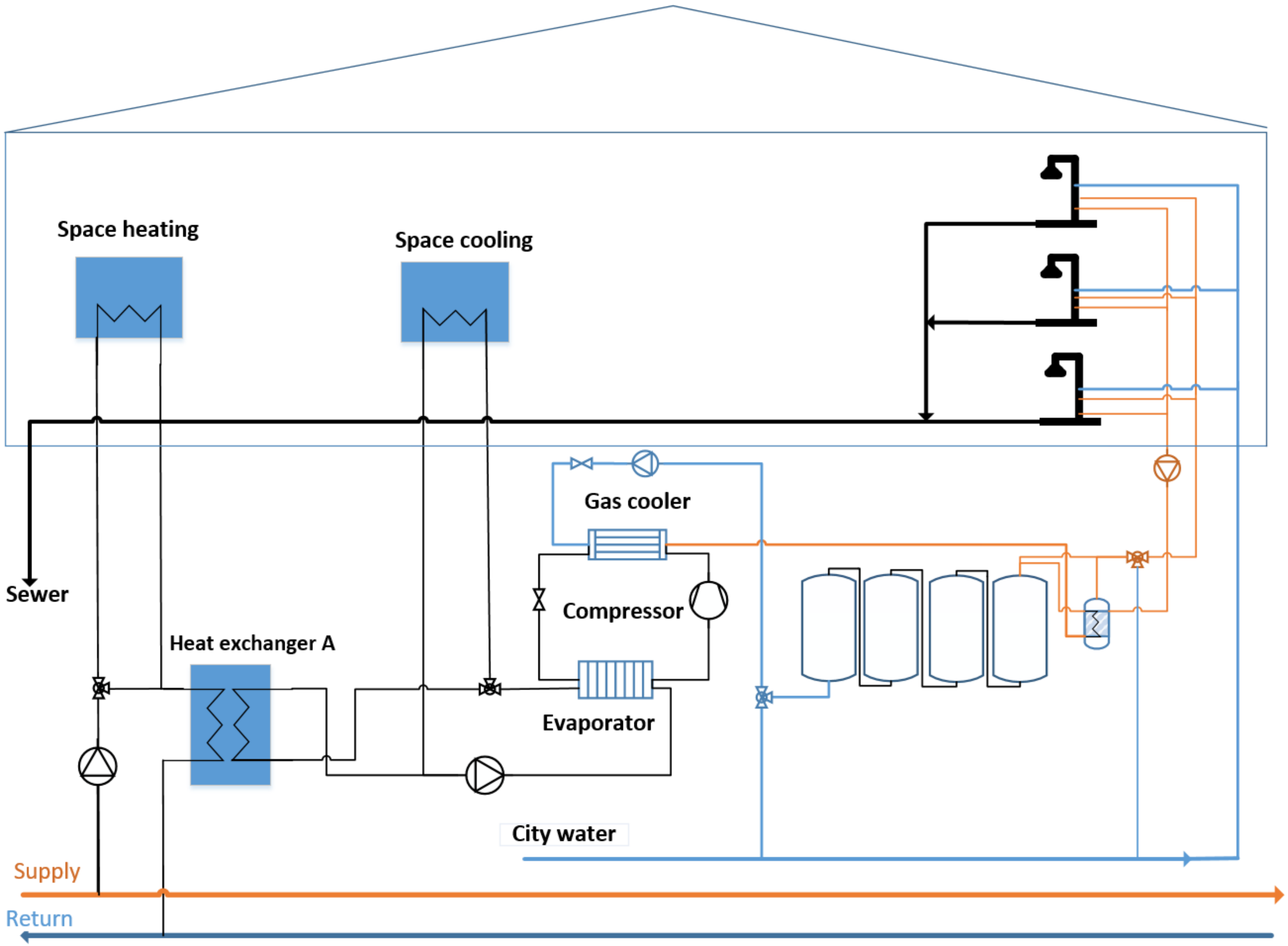

FIGURE 25: INITIAL ENERGY DISTRIBUTION SYSTEM

4.2.1 Space heating system

Figure 25 displays the energy distribution system on a building level. The figure displays the model in its most basic form, including only the most essential components. From the LTTG supply, water is pumped into the system for space heating. The space heating system is a floor heating system, in which hot water circulates in the floors giving off heat to the designated area. Chapter *2.3.2 Heating systems* states that $40°C$ is an acceptable water temperature for floor heating systems. It also states that the expected temperature drop of the water is 5K. This indicates that the average temperature of the exit water is $35°C$. Figure 26 is a display of the space heating system from Figure 25. It shows a more detailed view of how the supply water interacts with the floor plans in the different stories of the building.

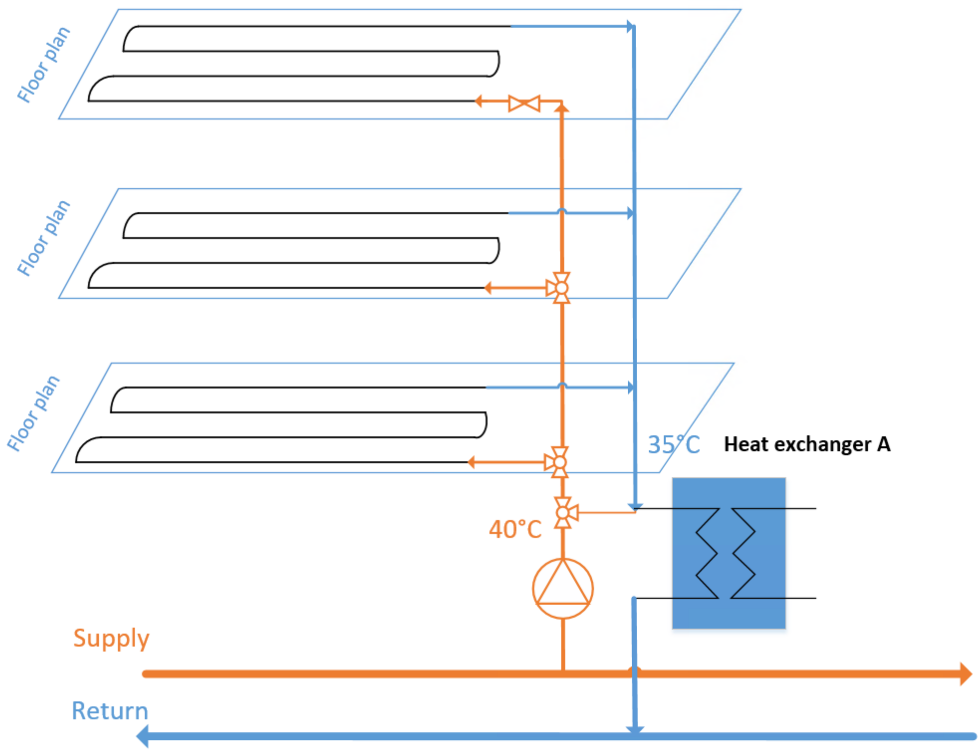

FIGURE 26: FLOOR HEATING SYSTEM

4.2.2 Cooling system

The operation of the cooling system of Leangen Sports Facility has a down period in the middle of summer. The total available waste heat from the skating rink is only 50.78 kWh in July, as shown by Table 10. However, Table 16 shows that the total need for SH and DHW in July is 3848 kWh. Thus, the waste heat from Leangen Sports Facility is not enough. As the LTTG cannot deliver enough energy, an alternative source for waste heat must be considered.

Table 13 show that, per average building, the need for cooling in July is 3693 kWh. The surplus of heat inside the building in the summer is thus an untapped source of heat and readily available. By using a waterborne ceiling system for cooling, the cooling system can be directly attached to the evaporator circuit. Figure 25 shows that the water leaving the evaporator passes through a valve in which it is directed either into the cooling system or onto the heat exchanger where it interacts with the LTTG. Figure 27 shows the path of the water when the cooling system is used as the heat pumps heat source.

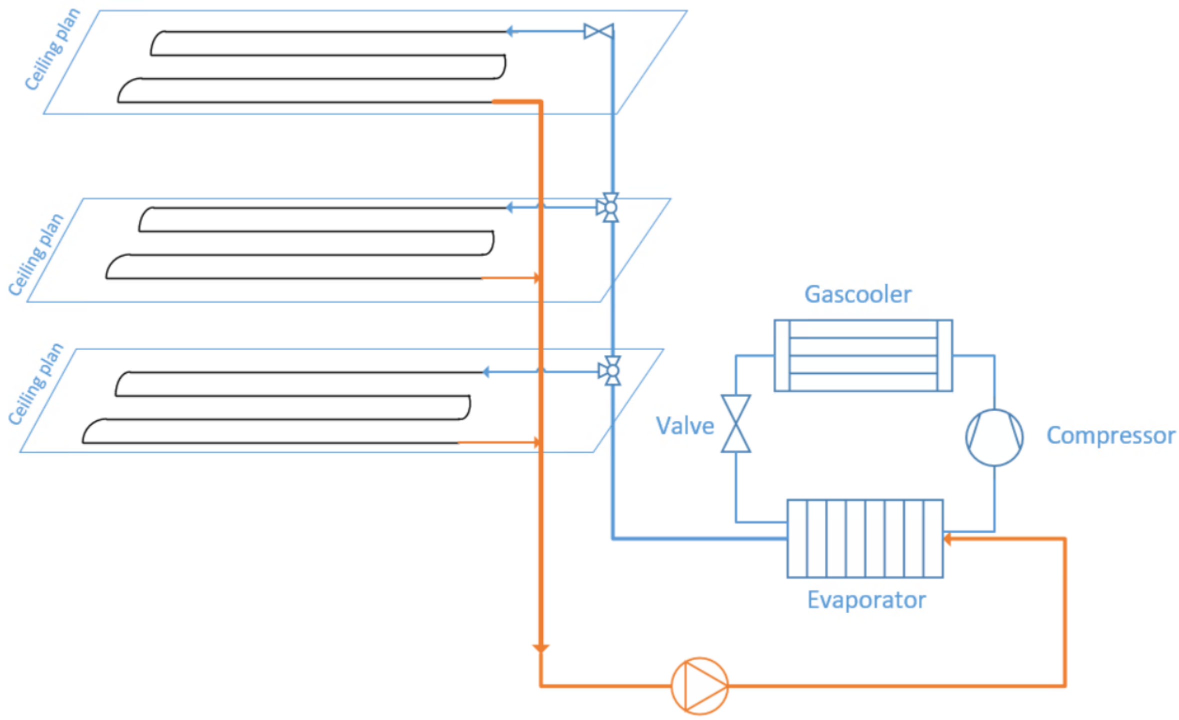

FIGURE 27: WATER BOURN CEILING COOLING SYSTEM

The figure shows the cold exit water from the evaporator being led through the waterborne ceiling system. This is represented by the blue lines. As the water flows through the ceiling system it picks up the excess heat in the building. The heated water is then pumped through the evaporator for heat rejection. By using the cooling system as a heat source for DHW production, the power needed for SC becomes the electrical energy needed to run the compressor in the heat pump. The system therefore benefits from the heat pumps COP twice, as the electricity need for DHW simultaneously covers both DHW and space cooling.

It is important to note that cooling mode is only assumed to be active between 10:00-19:00 since that is when the sun mostly affects the room temperature. The DHW production is designed to run continuously and as SC is active only 9h per day, the heat pump will need an additional source for the remaining hours of production.

4.2.3 DHW distribution system

After absorbing heat through the evaporator, the heat pump transfers it to the water flowing through the gas cooler. Cold city water at $8°C$ is pumped through the gas cooler where it is heated to approximately $75°C$. The heated water is then pumped into the hot water storage tanks. The actual hot water storage system is displayed in Figure 28. When in demand, DHW is discharged from the hot water storage tanks and sent to the hot water load, which in the sketches is symbolized by the showers. The model is based on Figure 14 where the system is built with an emphasis on maintaining proper stratification. Point 1-4 in the figure is meant to represent point 1-4 in Figure 14.

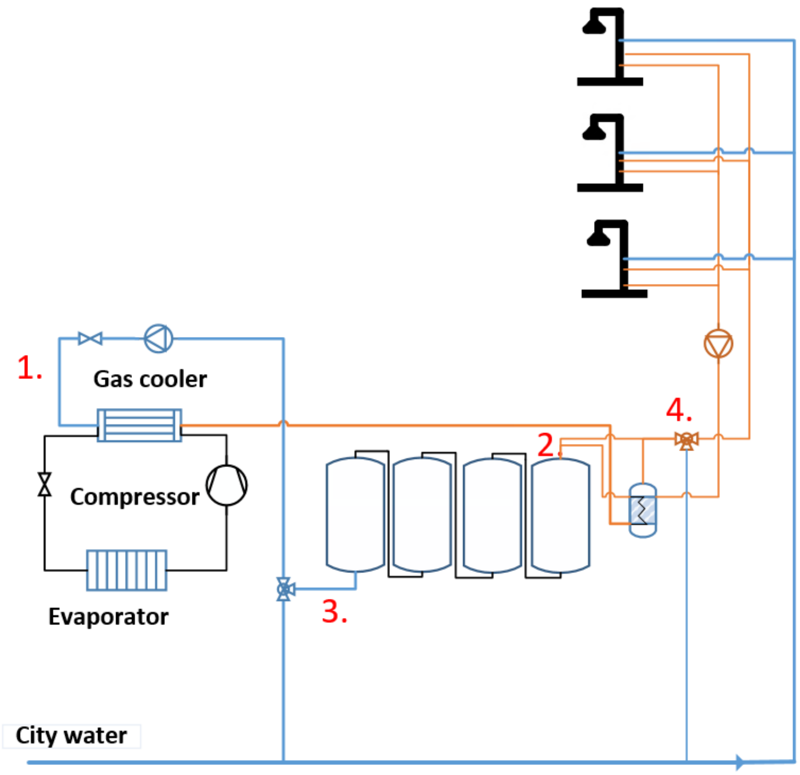

FIGURE 28: HOT WATER STORAGE- AND DISTRIBUTION SYSTEM

4.2.4 Sizing of the hot water storage system

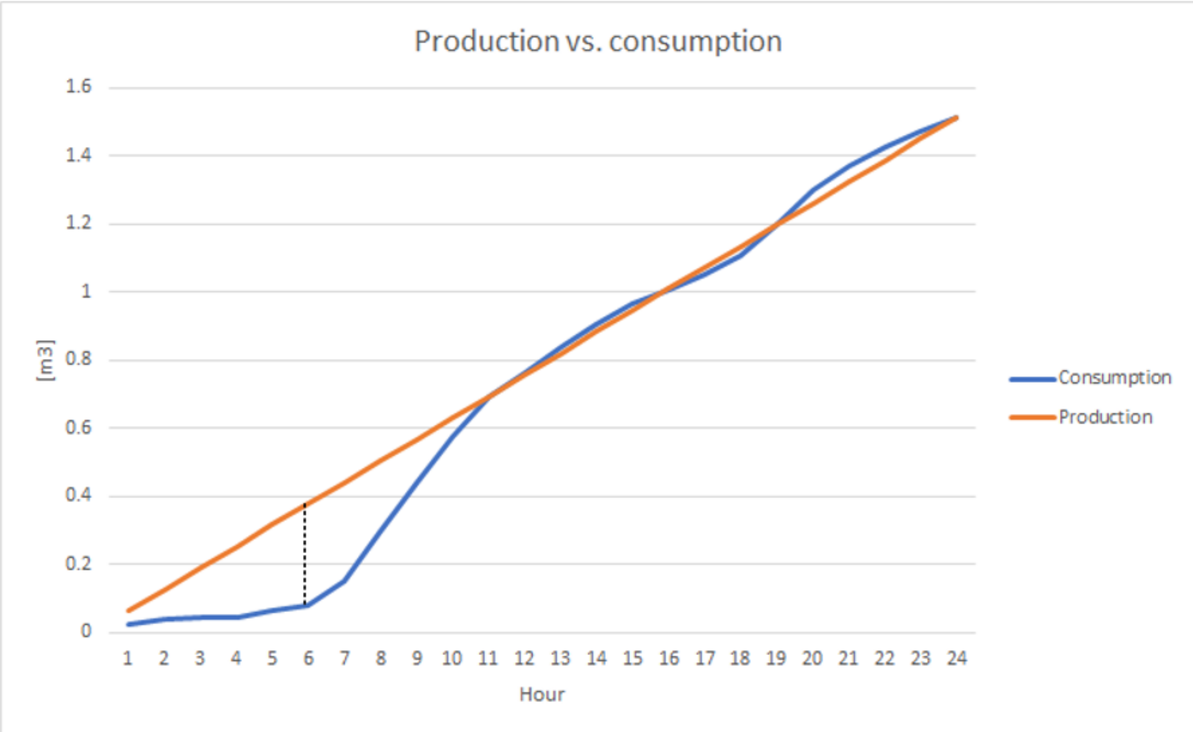

FIGURE 29: PRODUCTION VS. CONSUMPTION BASED ON A 24H PRODUCTION OF DHW

The production presented in Figure 29, is drawn under the assumption that the heat pump is in operation 24h per day. The point where the difference between production and consumption is largest is at 6:00. At this point the production exceeds consumption by 0.298 m^3. This means that the accumulation tanks must have at least a capacity of 0.298 m^3. Considering that the building is expected to house 40 inhabitants, a hot water tank of 298 litres seems rather small. If the heat pump fails and the system requires maintenance, a reservoir of less than 10 litres per person will not last long.

A solution might be to run the heat pump at a larger load but for a shorter time. If the heat pump only operates 20h per day, production will further exceed consumption and the tanks must be larger. This will not only provide security in the form of a larger DHW reservoir, but also by way of emergency operation. If a situation should arise where the DHW consumption is much larger than expected, the heat pump now has the option of running for 4 more hours. This way, the sudden need for DHW could be met by emptying the reservoir as well as extending the heat pumps operating time. Figure 30 compares the DHW consumption to production ranging from 12 to 22h of daily operation. Table 18 lists the minimum requirement of the storage tanks based on Figure 30.

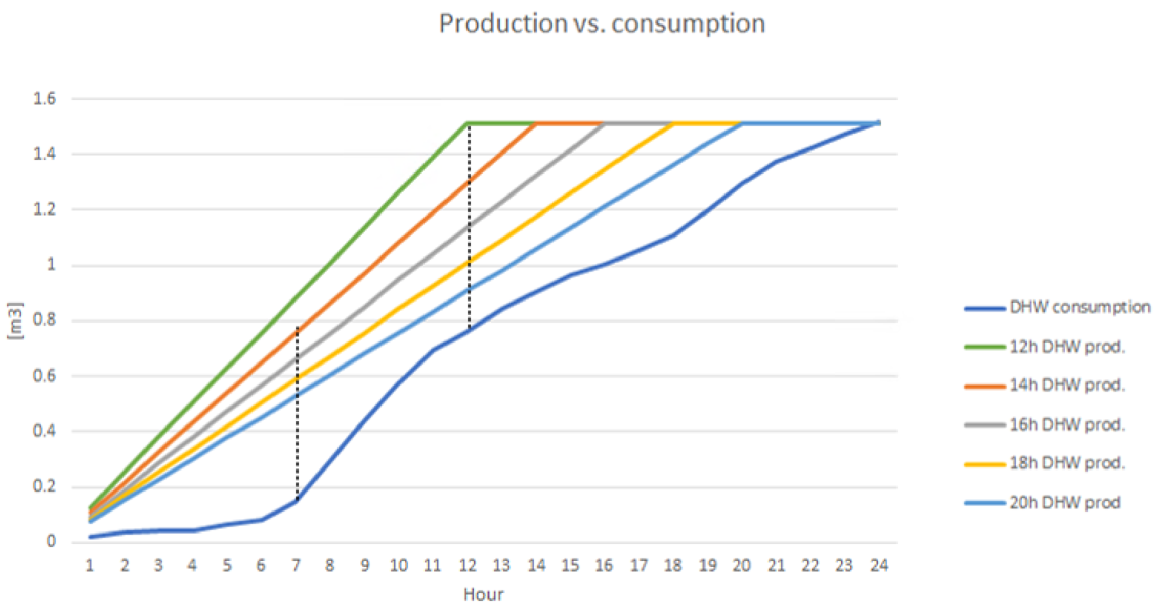

Figure 30: Production vs consumption with 12-22h of daily operation

Table 18: Volume flow and minimum storage capacity at 12-22h of daily DHW production

Runtime	Minimum storage tank capacity $[m^3]$	Production volume flow $[m^3/\text{h}]$
22h	0.319	0.066
20h	0.366	0.073
18h	0.422	0.081
16h	0.492	0.091
14h	0.583	0.104
12h	0.719	0.121

According to Figure 30 and Table 18, by reducing the daily run time of the heat pump to 12h, the daily accumulation of DHW is 719 litres. 719 litres is still relatively small considering that it is less than half a day's worth of DHW consumption.

As an extended safety it is decided to always have a storage equivalent to one day's consumption of DHW at hand. Figure 31 includes this, which is why the graphs displaying the accumulated production starts at $1.514m^3$.

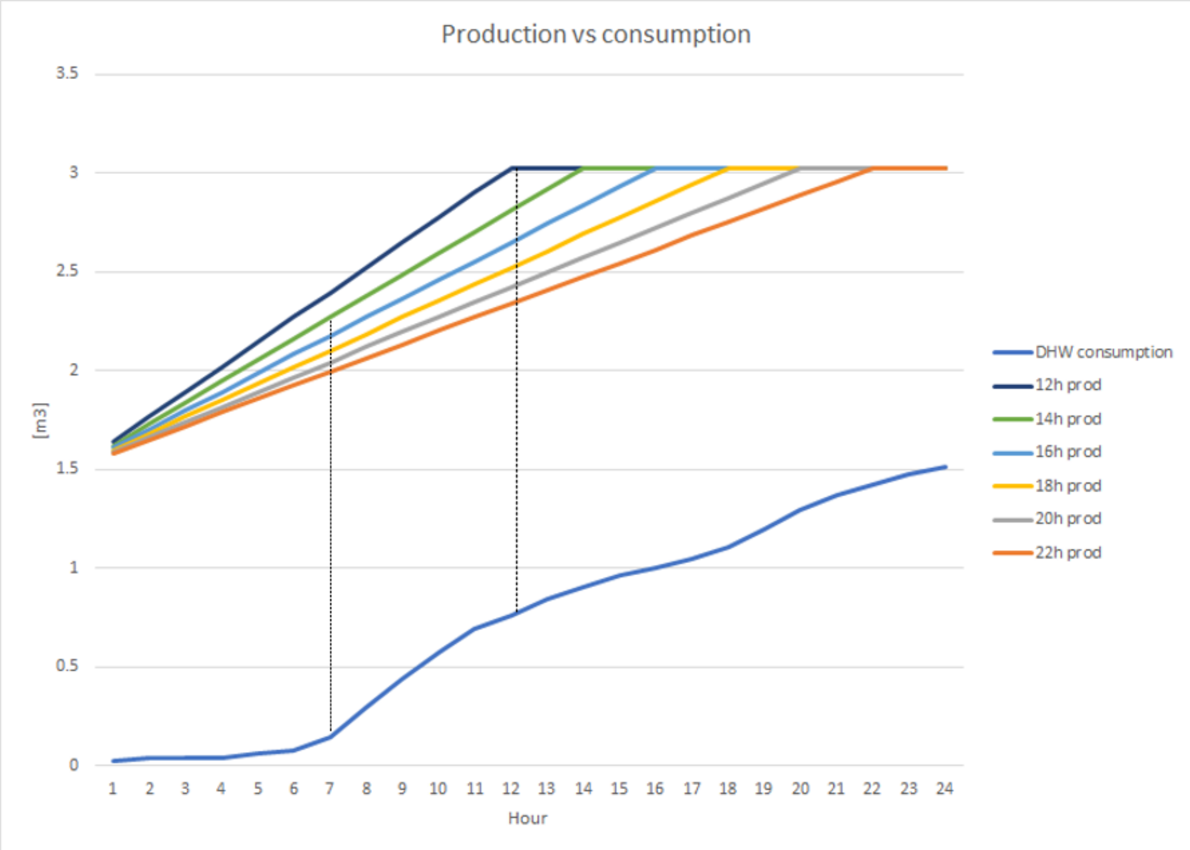

FIGURE 31: PRODUCTION VS CONSUMPTION AT 12-22H OF PRODUCTION WITH A CONSTANT RESERVOIR OF ONE DAY'S WORTH OF HOT WATER.

The minimum requirement for hot water storage capacity is greatly increased. For 14-22h runtime the largest difference between production and consumption occur at 07:00. For 12h runtime the point of largest difference is at 12:00. Table 19 lists the minimum required capacity for each scenario.

TABLE 19: MINIMUM REQUIRED STORAGE CAPACITY AT 12-22H OF DAILY DHW PRODUCTION

Runtime	Minimum storage capacity [m^3]
22h	1.847
20h	1.896
18h	1.955
16h	2.028
14h	2.123
12h	2.265

It is decided to go for 20h. This grants a room for 4h of extended operation. Table 18 shows that this means a daily additional production of almost 300 litres. Anything exceeding this seems redundant considering that the tanks already contain a safety reservoir of 1.514 m^3. Each time the storage tanks are drained to less than 1 514 litres of hot water, the heat pump has the option of running a few extra hours to fill in the missing amount.

Another aspect of reduced runtime is deciding when to insert the break in operation. Having the stop time in the middle of the day or in the morning might change the minimum required water tank capacity. Figure 32 compares daily production and consumption with varying stop times. Table 20 shows the minimum required storage capacity of each scenario.

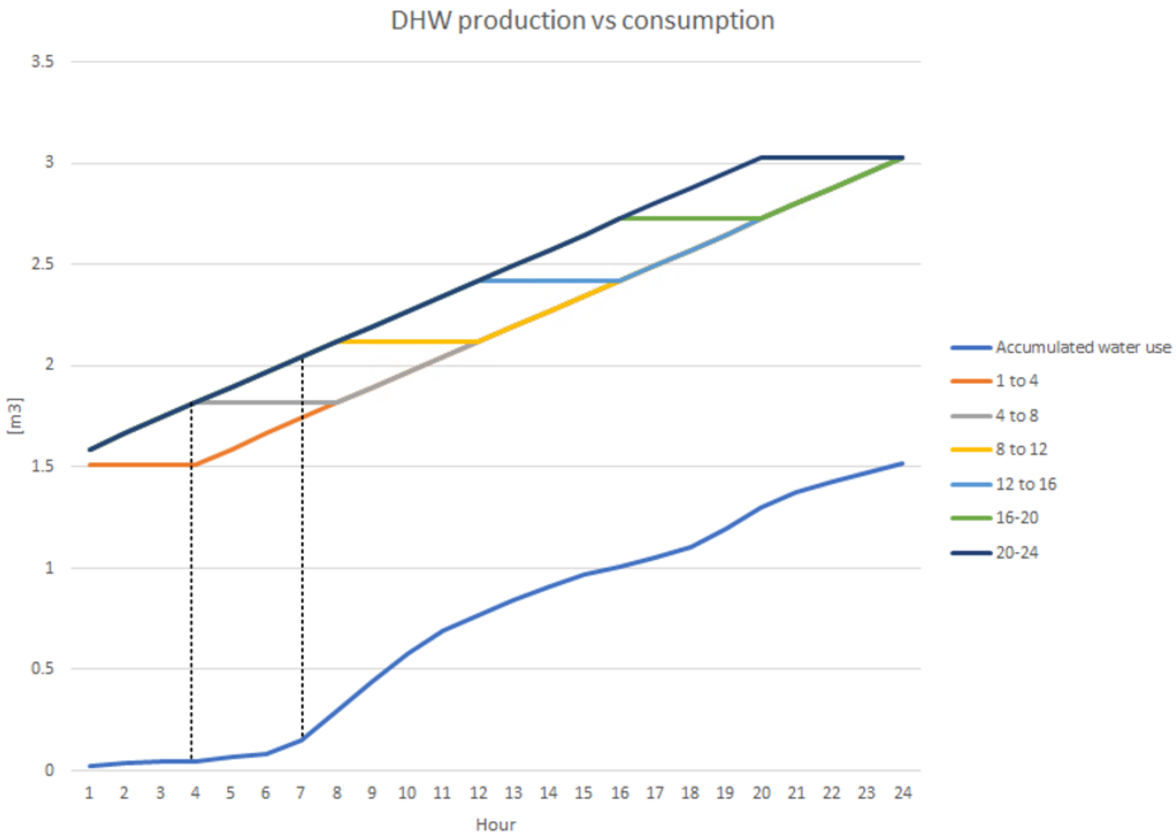

FIGURE 32: DHW PRODUCTION VS CONSUMPTION WITH 20H PRODUCTION TIME FOR DIFFERENT STOP TIMES

TABLE 20: MINIMUM REQUIRED STORAGE CAPACITY WITH DIFFERENT HEAT PUMP STOP TIMES

Stop time [time of day]	Minimum required storage capacity [m^3]
00 - 04	1.593
04 - 08	1.774
08 - 12	1.896
12 - 16	1.896
16 - 20	1.896
20 - 24	1.896

Table 20 shows that the minimum required capacity decreases by having the daily downtime between 00:00-08:00. Since cooling is meant to be used as a heat source for DHW it is important to keep the heat pump running while there is a need for cooling. Chapter *3.2 Representative building* shows that cooling demand is assumed between 10:00 and 19:00. The heat pumps downtime must therefore be set outside of this timeframe. It is therefore decided to let the heat pump run from 00:00 to 20:00, with the downtime from 20:00 to 00:00.

One of the main suppliers of hot water storage tanks are OSO AS. The sizes of the standard hot water storage tanks are 600 or 1000 litres (OSO Hotwater AS , 2018). With 20h production time the minimum required storage capacity is 1890 l. The storage capacity must therefore be two 1000 litres tanks. However, having a capacity of 2000 litres with a minimum requirement of 1890 litres gives a wiggle room of only 110 litres. If all 40 inhabitants one day consume 10 litres less than expected, the storage will require 400 litres of extra storage. It is therefore decided to install an extra tank that compensates for potential unforeseen scenarios. The result is three 1000 litre tanks with a combined hot water storage capacity of 3000 litres.

Figure 33 displays the three tanks. The first two tanks are the main storage holding the required capacity of 1890 litres. The last tank offers extra storage capacity in case of extraordinarily low DHW consumption. The object in the middle of tank nr. 2 is a temperature sensor. The sensor measures the temperature of the water at the point of $1.514m^3$. Since the tanks should always contain $1.514m^3$ of hot water, the temperature at this point should always be $75°C$. If, at the end of the daily production, the sensor measures a temperature lower than $75°C$ it instructs the heat pump to extend its runtime until the sensor is satisfied.

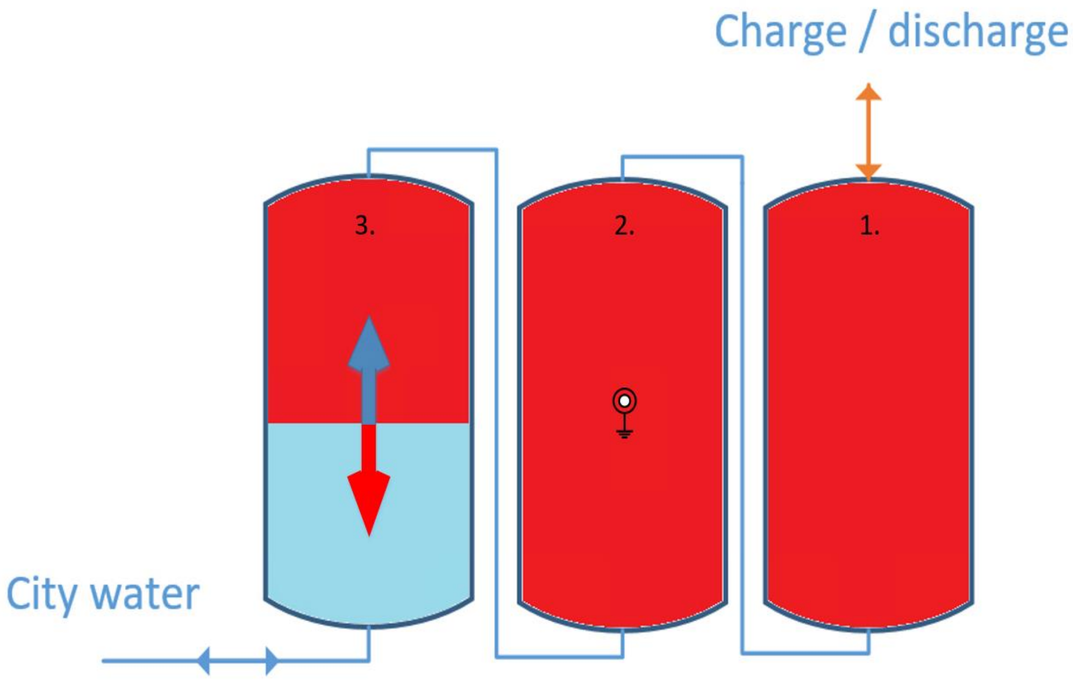

FIGURE 33: HOT WATER STORAGE TANK SYSTEM

4.2.5 Simulated heat pump

4.2.5.1 Normal mode

Chapter *2.3.3 Optimal gas cooler pressure* explains that a heat pumps performance can be evaluated by comparing the temperature glide of the CO2 and water in the gas cooler. Figure 34 shows the heat pumps performance in a temperature-enthalpy diagram with 100 bar gas cooler pressure. The blue line is the temperature glide of the heated water through the gas cooler.

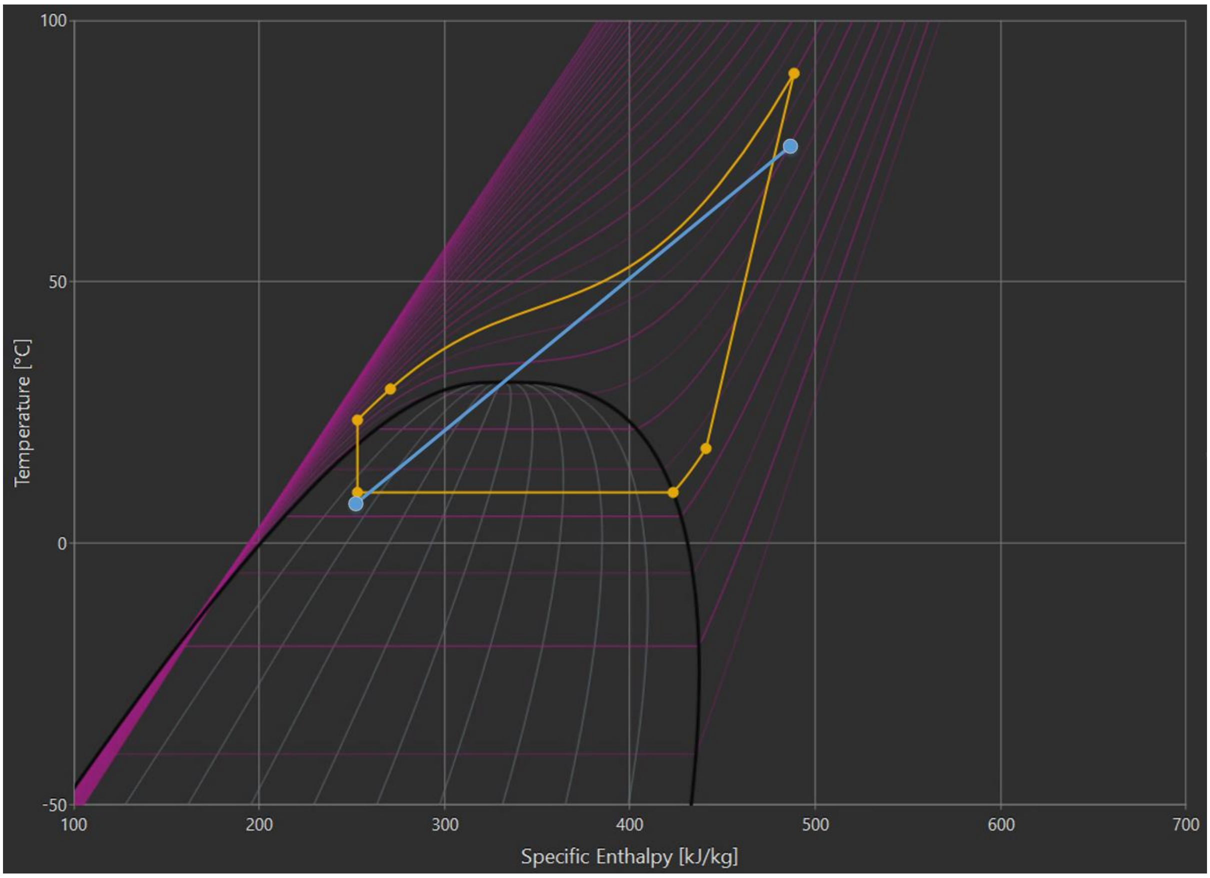

FIGURE 34: TEMPERATURE-ENTHALPY DIAGRAM FOR THE HEAT PUMP AT 100BAR

The figure shows that the pinch point is located inside the gas cooler, similar to Figure 10. This indicates a low gas cooler pressure. A low gas cooler pressure causes a high CO2 temperature at the gas cooler outlet, resulting in a high Δt_a. High Δt_a causes high expansion losses and reduces COP. Figure 11 shows that by sufficiently raising the gas cooler pressure, the pinch point can be moved to the gas cooler outlet. This reduces Δt_a, increasing the performance of the heat pump. To improve the temperature glide of the CO2 the gas cooler pressure is increased. Gradually increasing the pressure reveals that optimal temperature glide is achieved at 115 bar. Figure 35 shows the t-h diagram presenting the heat pumps performance with a gas cooler pressure of 115 bar.

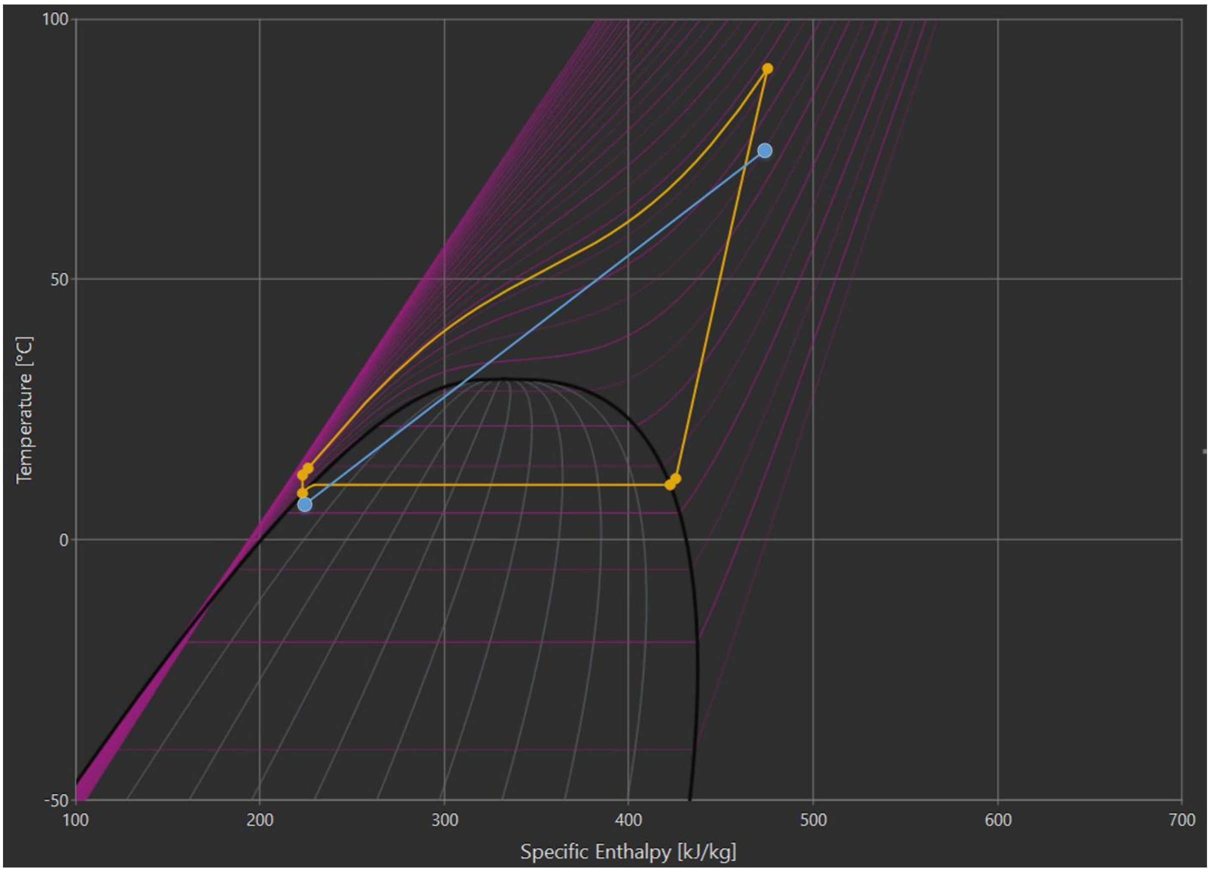

FIGURE 35: TEMPERATURE-ENTHALPY DIAGRAM FOR THE HEAT PUMP AT 115BAR

At 115 bar the pinch point is located at the gas cooler outlet with a Δt_a of 3.92K. Table 8 show that 115 bar is higher than the expected optimal gas cooler pressure. However, the information in Table 8 is the result of realistic conditions which might not be accurately reflected in the simulation.

Since the heat pump is set to start and stop once every day, it is important that it reaches steady state quickly. Figure 36 shows the time it takes for the PI-regulator to stabilise the compressor speed.

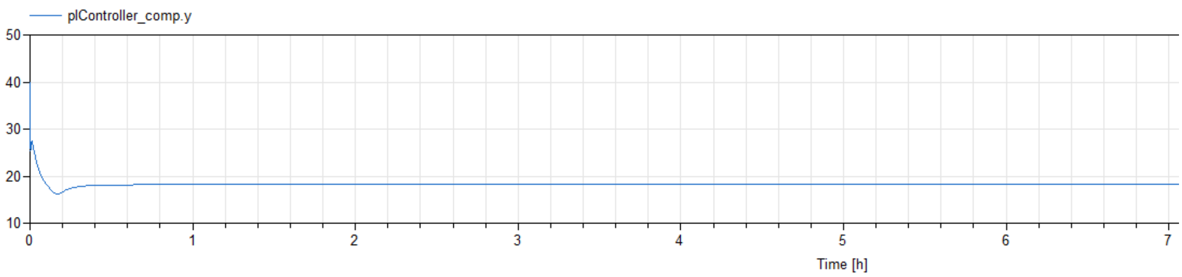

FIGURE 36: OUTPUT SIGNAL FROM THE PI-CONTROLLER TO THE COMPRESSOR

The figure shows that the heat pump reaches steady state after approximately 24 minutes. This is deemed acceptable.

Figure 37 shows the heat pumps performance with a gas cooler pressure of 115 bar depicted in a p-h diagram. The COP for the heat pump is 4.74 and is calculated based on the enthalpy displayed in the p-h diagram.

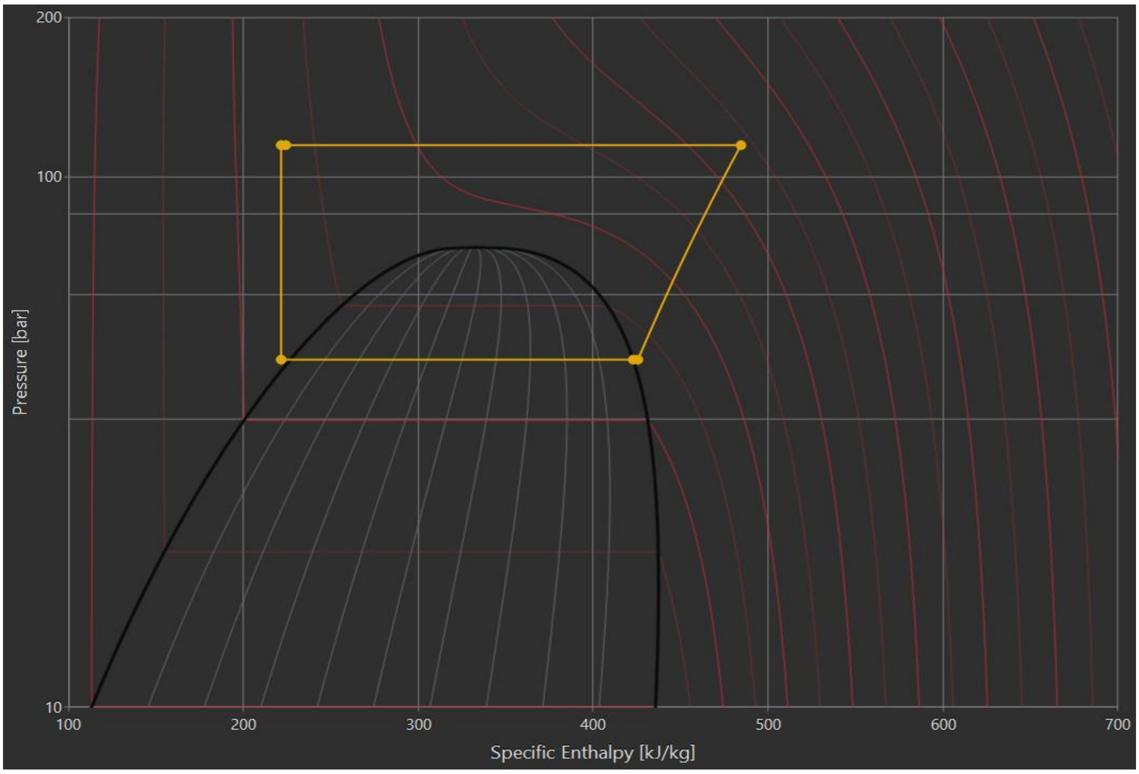

FIGURE 37: P-H DIAGRAM OF THE HEAT PUMP IN STEADY STATE WITH 115 BAR GAS COOLER PRESSURE

4.2.5.2 Cooling mode

Chapter *2.3.2 Cooling systems* explains that the mean water temperature in the cooling system should not be lower than 14-16$°C$ because of the risk of condensation. Chapter *2.3 Hydronic heating and cooling systems* states that the expected temperature change in such systems is 5K. The evaporator inlet temperature is therefore set to be 19$°C$. With an inlet temperature of 19$°C$ the water flow is adjusted until the outlet temperature is 14$°C$.

The only parameters changed from Normal mode to Cooling mode is the evaporator inlet/outlet temperature and water flow. By increasing the outlet temperature from 10 to 14$°C$, the evaporator temperature is increased to 12$°C$. The increased evaporator temperature lowers the temperature lift by the compressor. The reduced heat input from the compressor causes an increase in COP. The COP in cooling mode is therefore 4.93 while the COP in normal mode is 4.74.

Table 21 displays the heat pump characteristics. The gas cooler capacity is calculated from equation (2) where $\dot{m}$ is the gas cooler water flow. The water flow through the gas cooler is derived from Table 18 which states that the hourly produced DHW at 20h runtime is 0.073m^3/h. This is equivalent to a mass flow of 0.0201kg/s. The evaporator capacity is thus calculated from equation (6) and (12).

TABLE 21: HEAT PUMP CHARACTERISTICS IN NORMAL MODE AND COOLING MODE

	Characteristics	Value
Normal mode	Evaporator temperature [$°C$]	10
	Evaporator capacity [kW]	4.44
	Gas cooler capacity [kW]	5.63
	COP	**4.74**
Cooling mode	Evaporator temperature [$°C$]	12
	Evaporator capacity [kW]	4.44
	Gas cooler capacity [kW]	5.63
	COP	**4.93**

Table 8 shows that the expected COP of a CO2 heat pump producing 75$°C$ DHW, is between 3.09 and 3.58. The COP of the simulated heat pump is 4.74 and 4.93. This can be a result of the difference in evaporation temperature. The information in Table 8 is based on a t_e of -5$°C$ while the simulated heat pump has t_e of 10 and 12$°C$. Chapter *2.4.3.4 Evaporation temperature* explains that a higher evaporation temperature means higher COP. This might be the reason for the difference between expected and simulated COP.

4.2.6 Waste heat consumption

As described in chapter *3.1 Leangen Sports Facility*, the basic system revolves around using waste heat from Leangen as the soul heat source for SH and DHW production. Table 22 shows the need for heating, cooling and DHW in one building.

TABLE 22: MONTHLY ENERGY DEMAND OF THE AVERAGE BUILDING

	SH demand [kWh]	DHW demand [kWh]	SC demand [kWh]	Total energy demand for heating and cooling [kWh]
Jan	8328.06	3633.43	0	11961.49
Feb	6292.54	3283.68	0	9576.22
Mar	5797.68	3636.53	0	9434.21
Apr	2375.72	3519.62	0	5895.34
Mai	631.51	3635.004	1560	4266.514
Jun	211.59	3522	3080	6793.59
Jul	212.73	3635.78	3696	7496.51
Aug	137.75	3636.69	2926	6539.44
Sep	518.78	3517.82	616	4036.6
Okt	2609.94	3633.43	0	6243.37
Nov	4770.21	3520.72	0	8290.93
Des	4757.27	3635.43	0	8392.7
Annual	**36643.78**	**42810.134**	**11878**	**91331.91**

The numbers listed in the table are derived from the values in Table 16, where the $\left[\frac{kWh}{m^2}\right]$ values are multiplied with 1400 m^2 (area per building). As the system is based on relying on the waste heat from the LTTG, it is necessary to find out how much waste heat is required to support the system. The SH system draws energy directly from the LTTG. The need for LTTG energy to cover space heating will therefore be the same as listed in Table 22. Figure 26 shows that the heat pump uses the leftover heat from SH as its heat source. The need for LTTG energy will therefore be the need for SH plus the required heat to cover DHW.

Cooling mode

The heat pump also has the option to use SC as a heat source. The cooling system is active during the day, from 10:00-19:00. Since the need for cooling is unpredictable and very much reliant on the weather, it is, for simplicity's sake, decided to assume a steady need for cooling between 10 and 19. The hourly need for cooling is therefore the product of daily SC demand divided by 9 *(*the hours during which cooling is needed*)*. Figure 38 compares the average daily need for SC and DHW in May per building.

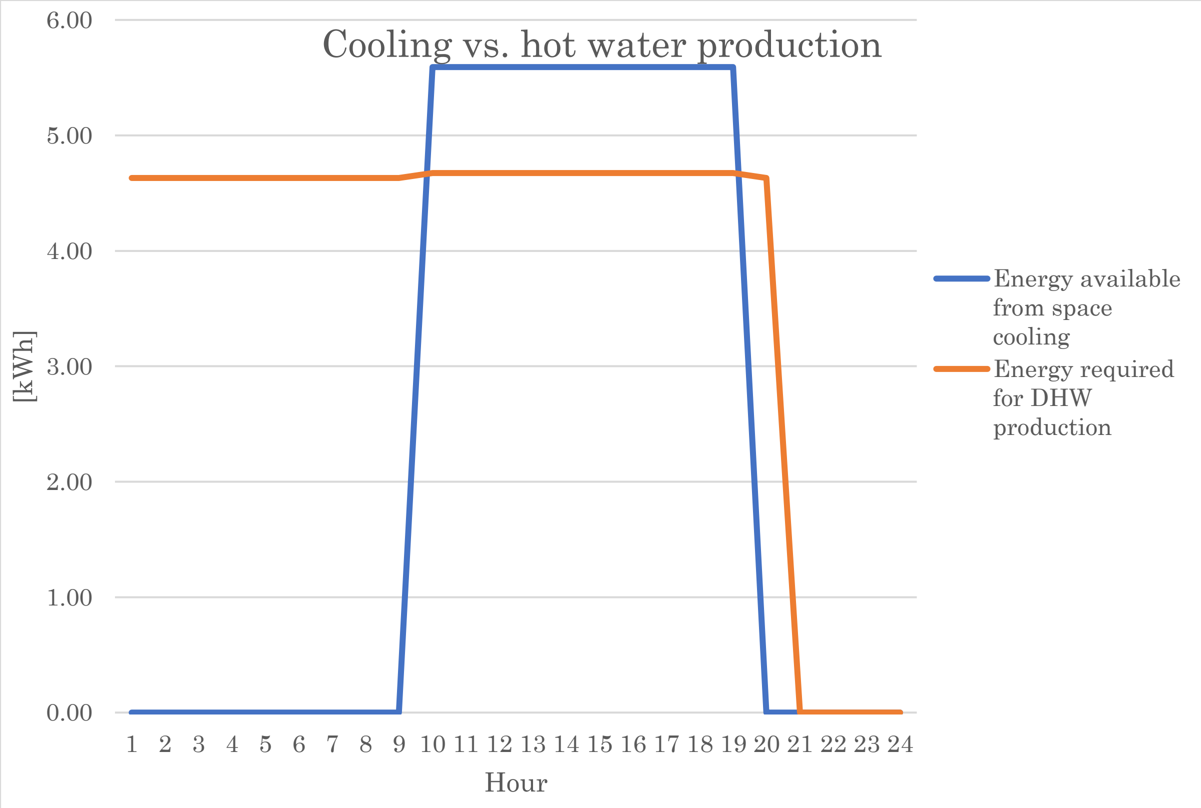

FIGURE 38: HOURLY AVAILABLE ENERGY FOR COOLING COMPARED TO ENERGY REQUIRED FOR DHW PRODUCTION

The production of DHW is set to be in constant operation for 20h a day. *Energy required to cover DHW production* displays the required energy supply to the evaporator. The slight increase in the required energy from 10:00 to 19:00 is a result of the increased COP during cooling mode. The figure shows that the hourly need for DHW is too low to entirely cover the need for cooling. This might be solved by installing dry coolers to the system. Parts of SC can thus be used for DHW production while the rest is released out into the air. However, that would require a total redesign of the heat pump and a complete analysis of the cooling profile as to size the evaporator and dry cooler large enough to cover the cooling peaks. That way, the need for cooling would benefit from the heat pumps COP, assuming it would otherwise be covered by electricity.

Figure 38 also shows that when SC is active, it can cover 50% of the required energy supply to the evaporator. However, this only applies when the hourly need for SC is larger than the required evaporator energy supply. This is not the case for September when the monthly need for cooling is 616kWh. The average daily need for cooling in September is 20.5 kWh. Since cooling is only needed 9 hours a day, this gives an hourly need of 2.28 kWh. The hourly energy need for the heat pump is 4.74 kWh. This means that the heat pump must simultaneously draw energy from the LTTG and cooling system to cover the DHW demand. This is rather complicated. In normal mode the heat pump uses the LTTG as a heat source. The water inlet temperature of the evaporator is $35°C$ with a mass flow of 0.045 kg/s. When using SC as a heat source, the inlet water temperature is $19°C$ with a mass flow of 0.21 kg/s. By using both at once, the resulting temperature and mass flow would be hard to predict. Wrongly adjusting the two mass flows can lead to an evaporator energy supply insufficient to produce the desired DHW temperature. It is therefore decided to neglect SC as a heat source in September.

TABLE 23: REQUIRED ENERGY SUPPLY FORM THE LTTG FOR SPACE HEATING AND DOMESTIC HOT WATER

	LTTG energy for SH [kWh]	LTTG energy for DHW [kWh]	Total need for LTTG energy per building [kWh]
Jan	8328.06	2866.88	11194.94
Feb	6292.54	2590.92	8883.46
Mar	5797.68	2869.33	8667.01
Apr	2375.72	2777.08	5152.80
Mai	631.51	1434.06	2065.57
Jun	211.59	1389.48	1601.07
Jul	212.73	1434.37	1647.10
Aug	137.75	1434.73	1572.48
Sep	518.78	2775.66	3294.44
Okt	2609.94	2866.88	5476.82
Nov	4770.21	2777.95	7548.16
Des	4757.27	2868.46	7625.73
Annual	**36643.78**	**28085.82**	**64729.60**

Table 23 shows the need for waste heat supply to each building throughout the year. The column showing *LTTG energy for DHW* shows the total need for energy supply to the evaporator minus the 50% of the daily energy contribution from SC during May-August. *Total need for LTTG energy per building* is the sum of energy delivered to SH and DHW production by the LTTG.

Figure 39 shows the monthly waste heat consumption compared to the monthly waste heat production from the sports facility during the three building steps. When the construction is complete, there is going to be a total of 86 average buildings at Leangen. Therefore, the columns representing building step 1 show the consumption of $(86 \cdot \frac{1}{3})$ buildings. The columns representing building step 2 show the consumption of $(86 \cdot \frac{2}{3})$ buildings. The columns representing building step 3 show the consumption of 86 buildings.

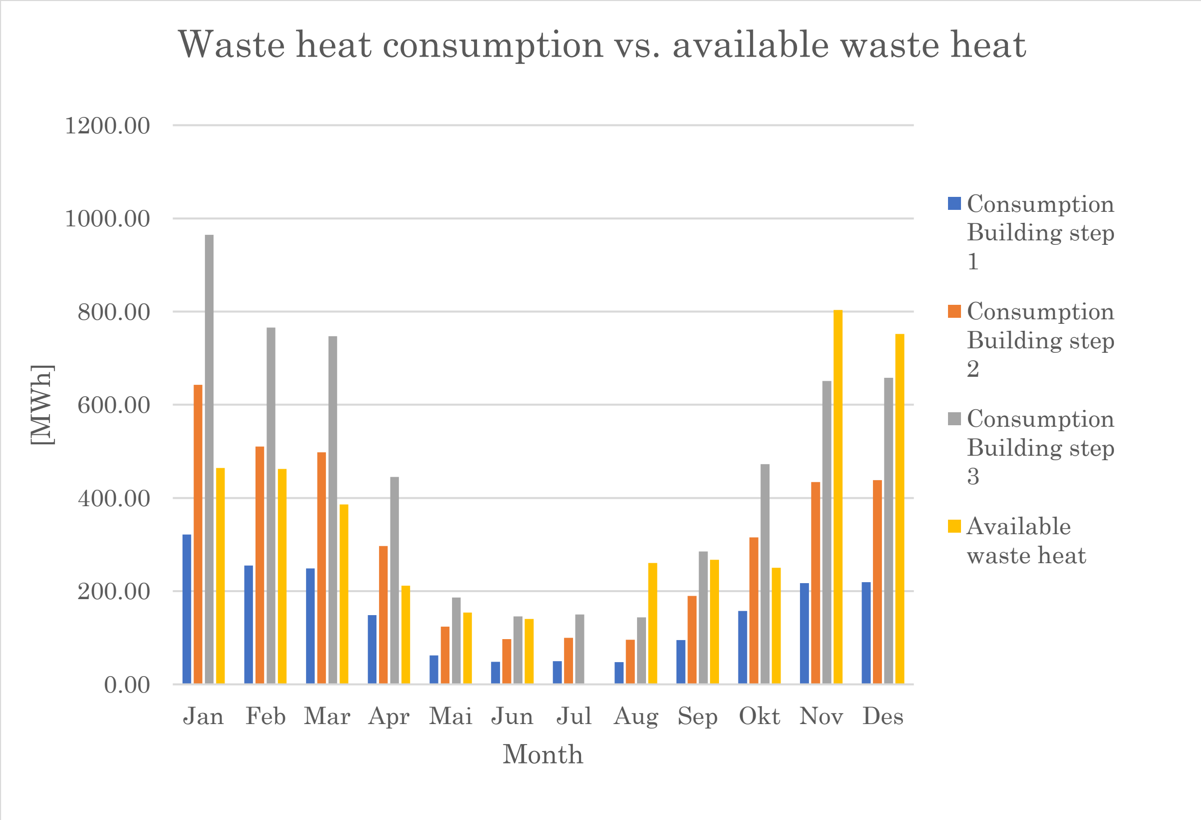

FIGURE 39: WASTE HEAT CONSUMPTION DURING THE DIFFERENT BUILDING STEPS AND THE AVAILABLE WASTE HEAT

In building step 1, the energy supplied by the sports facility is enough to cover the demand. The only exception is July, since that is off season and the skating rinks will not be in operation. In building step 2 however, 6 out of 12 months have an energy demand that exceeds the available waste heat. The heating demand in building step 3 can only be covered by waste heat in August, November and December. Thus, the energy supplied by the skating rink is not enough to supply Leangen with its expected energy need.

4.3 Greywater heat recovery

4.3.1 Greywater accumulation at each building

One way of using GW as a heat source for DHW could be by collecting the greywater in a large tank underneath each building. The tank can then be connected to the evaporator circuit where the water can pass through the greywater tank and collect heat.

Chapter *2.2.2 Greywater heat recovery* presents the option of using greywater as an alternative waste heat source. It also states that the most reliable sources with the highest temperatures are showers, baths, washing machines and dishwashers. Table 24 shows the expected daily greywater production from these select sources per building. Table 25 shows its heat harnessing potential with a ΔT of 25 K. Since the outlet temperature of the evaporator is 10°C, the greywater temperature is assumed to have a lower limit of $10°C$. With an average temperature of $35.32°C$ and a lower limit of $10°C$, ΔT is set to 25 K.

TABLE 24: DAILY GREYWATER PRODUCTION PER BUILDING FROM SHOWERING, BATHS, DISHWASHERS AND WASHING MACHINES

Parameters	Value
Apartments per building	20
Residents per apartment	2
Wastewater per day per person [kg]	85.40
Wastewater per building $[\frac{kg}{day}]$	3416

TABLE 25: ENERGY POTENTIAL OF DAILY GREYWATER PRODUCTION FROM SHOWERS, BATHS, DISHWASHERS AND WASHING MACHINES.

Parameter	Value
Specific heating capacity of water Cp$[\frac{kJ}{kg \cdot K}]$	4.18
Wastewater mass m [kg]	3416
Temperature lift dT[K]	25
Available energy Q[kJ]	356972
Available energy Q[kWh]	99.16

Per building, as Table 25 shows, the greywater from showers, washing machines and dishwashers can potentially provide 99.16 kWh of heat per day.

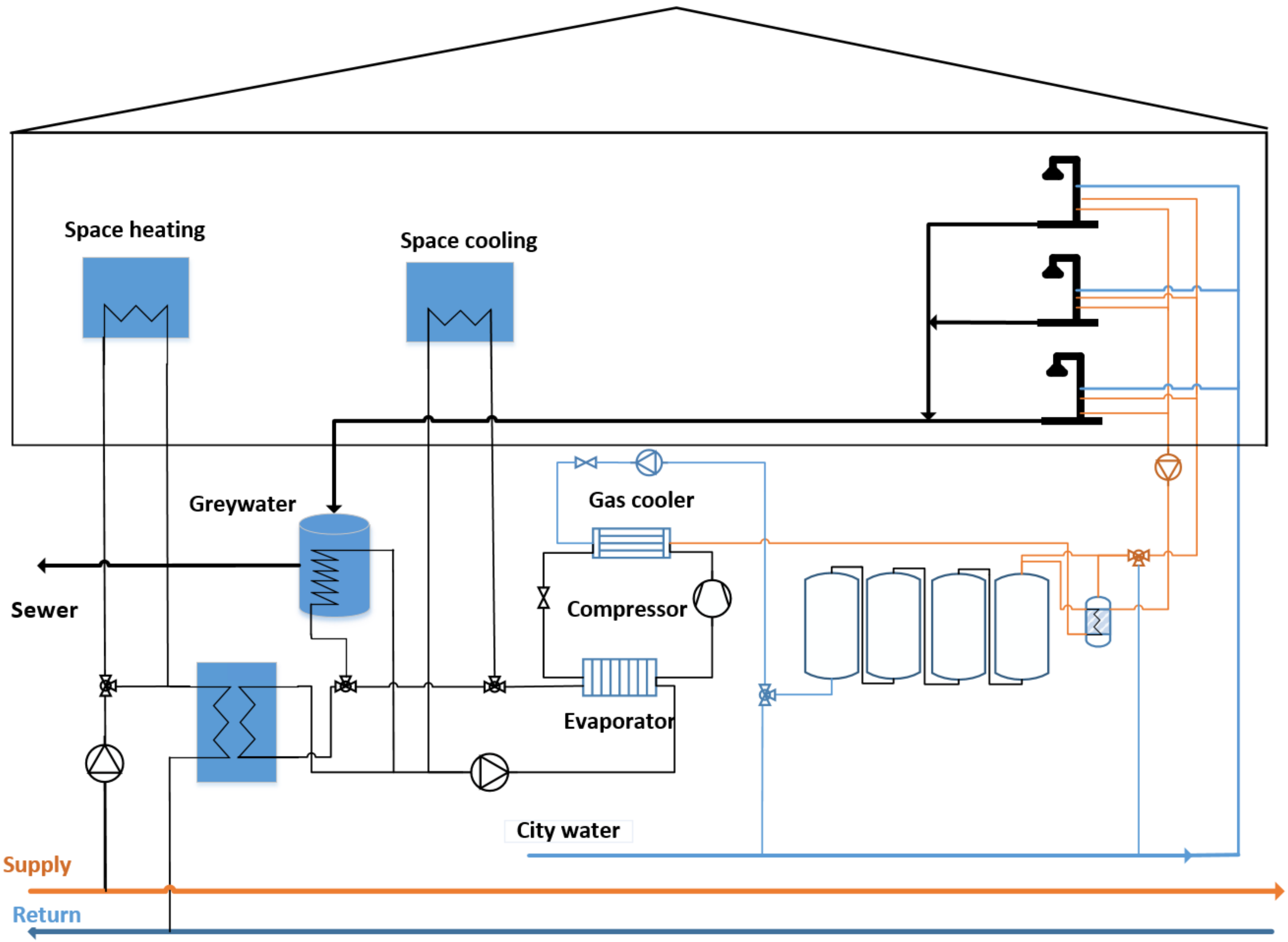

FIGURE 40: ENERGY DISTRIBUTION SYSTEM WITH GREYWATER

4.3.1.1 Greywater as independent heat source

Two ways of connecting the greywater tank to the heat pump have been explored. The first one is connecting the grey water tank to the heat pump as an independent heat source. Figure 40 shows the entire system sketch including the greywater tank. The exit water from the evaporator first enters a valve in which it can be directed into the cooling system. If the water is not directed into the cooling system, it is directed on to the next valve. This valve decides if the water is sent to the Heat exchanger A or to the greywater tank. Figure 41 shows an up-close-view of how the greywater tank is attached to the evaporator. The water enters the greywater tank from the bottom and exits at the top. As presented in chapter *2.6 Hot water storage,* water naturally stratifies. If the water in the greywater tank stays still, the temperature at the bottom will naturally be colder than that at the top. When the water from the evaporator circuit enters at the bottom, it is met by the coldest greywater first. Thus, allowing for the water to absorb heat all the way through the tank since the surrounding temperature constantly increases.

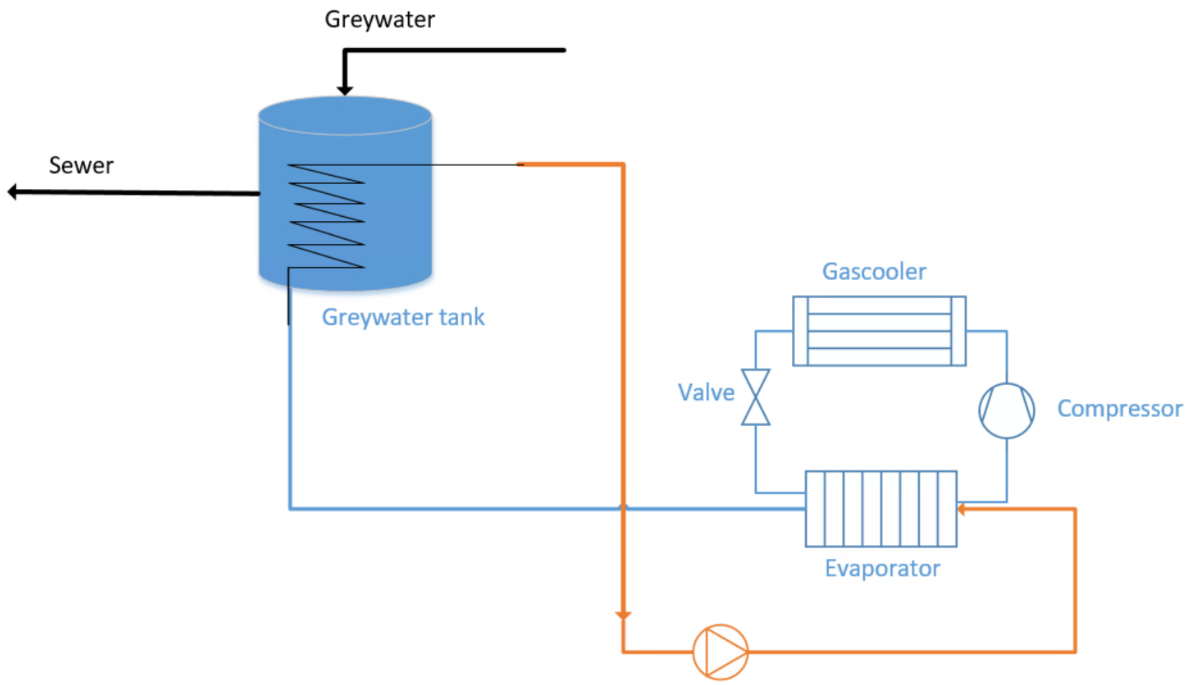

FIGURE 41: DISPLAY OF THE CONNECTION BETWEEN EVAPORATOR AND GREYWATER TANK AS INDIVIDUAL HEAT SOURCE

Since the temperature and production is unpredictable, the temperature lift supplied by greywater will vary greatly. A lower temperature lift means less absorbed energy. To supply the evaporator with enough energy, the speed of the pump in the evaporator circuit must increase proportionally to the temperature decrease in the greywater tank. Table 26 shows the change in mass flow through the evaporator at different temperatures to maintain an exit temperature of $10°C$. It also presents the resulting COP.

TABLE 26: CHANGE IN MASS FLOW WITH CHANGING GREYWATER TEMPERATURES

GW temperature [$°C$]	Mass flow [kg/s]	COP
35	0.0427	4.74
30	0.054	4.72
25	0.072	4.63
20	0.109	4.53
15	0.21	4.39

The table shows that even the lowest COP is higher than 4. This shows that using a greywater tank as an independent heat source is a viable option. The pump controlling the mass flow can be connected to a temperature sensor in the grey water tank and adjust the mass flow accordingly. Figure 42 presents the graph showing the increasing mass flow with a decreasing greywater temperature.

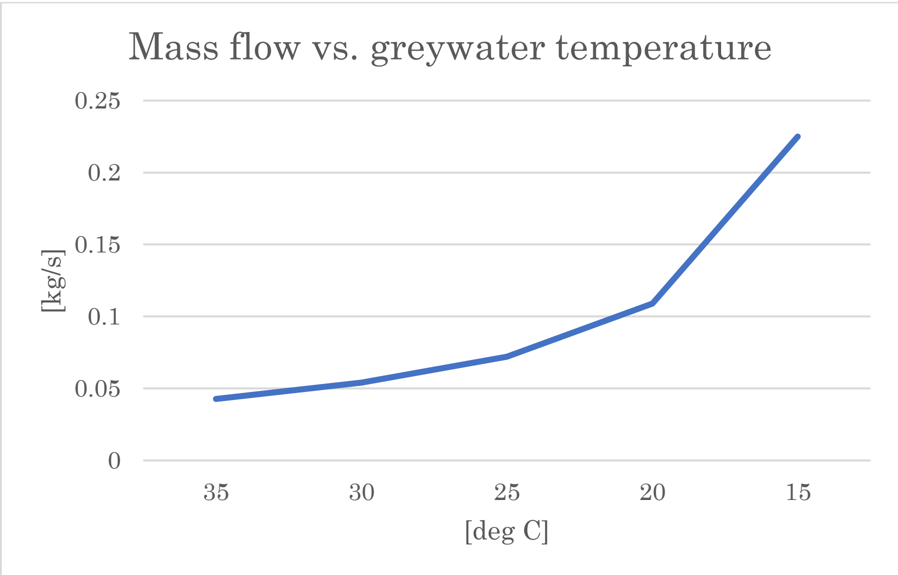

FIGURE 42: MASS FLOW AT CHANGING GREYWATER TEMPERATURES

The graph shows that the closer the evaporator supply temperature is to the evaporation temperature, the larger the mass flow needs to be. To avoid very large mass flows, it is decided not to use greywater as a heat source if the temperature drops below $20°C$. With an average temperature of $35°C$ and a lower limit of $20\ °C$, the average available temperature lift from greywater is 15K. Table 27 shows the potential for heat recovery from greywater with a lower temperature limit of $20°C$.

TABLE 27: GREYWATER POTENTIAL WITH A LOWER TEMPERATURE LIMIT OF 20°C

Parameter	Value
Specific heating capacity of water Cp$[\frac{kJ}{kg \cdot K}]$	4.18
Wastewater mass m [kg]	3416
Temperature lift dT[K]	15
Available energy Q[kJ]	356972
Available energy Q[kWh]	59.49

With waste heat from the skate rink, space cooling and greywater, the heat pump has three available heat sources. Figure 39 shows that waste heat from the skate rink alone is not enough to cover the heating demand in building step 2 and 3. It is therefore decided to prioritize greywater and space cooling as heat sources for DHW. Prioritizing greywater and space cooling leads to a reduced need for energy from the LTTG. Table 28 presents the monthly and annual consumption from the three sources.

TABLE 28: ENERGY DRAWN FROM GREYWATER, COOLING AND LTTG FOR DHW PRODUCTION

	Energy supply evaporator [kWh]	Available greywater energy [kWh]	Available energy from cooling [kWh]	LTTG DHW energy supply [kWh]
Jan	2888.87	1844.19	0	1044.68
Feb	2610.79	1665.72	0	945.07
Mar	2891.34	1844.19	0	1047.15
Apr	2798.39	1784.7	0	1013.69
Mai	2890.13	1844.19	1445.06	0.00
Jun	2800.28	1784.7	1400.14	0.00
Jul	2890.74	1844.19	1445.37	0.00
Aug	2891.47	1844.19	1445.73	0.00
Sep	2796.96	1784.7	0	1012.26
Okt	2888.87	1844.19	0	1044.68
Nov	2799.26	1784.7	0	1014.56
Des	2890.46	1844.19	0	1046.27
Annual	**34037.57**	**21713.85**	**5736.31**	**8168.37**

Table 23 shows that the heat pump consumes an annual amount of 28086 kWh of waste heat from Leangen Sports Facility. By including grey water as a heat source this is reduced to 8168 kWh. Figure 43 compares the consumption and production of waste heat from the skating rink when greywater is used.

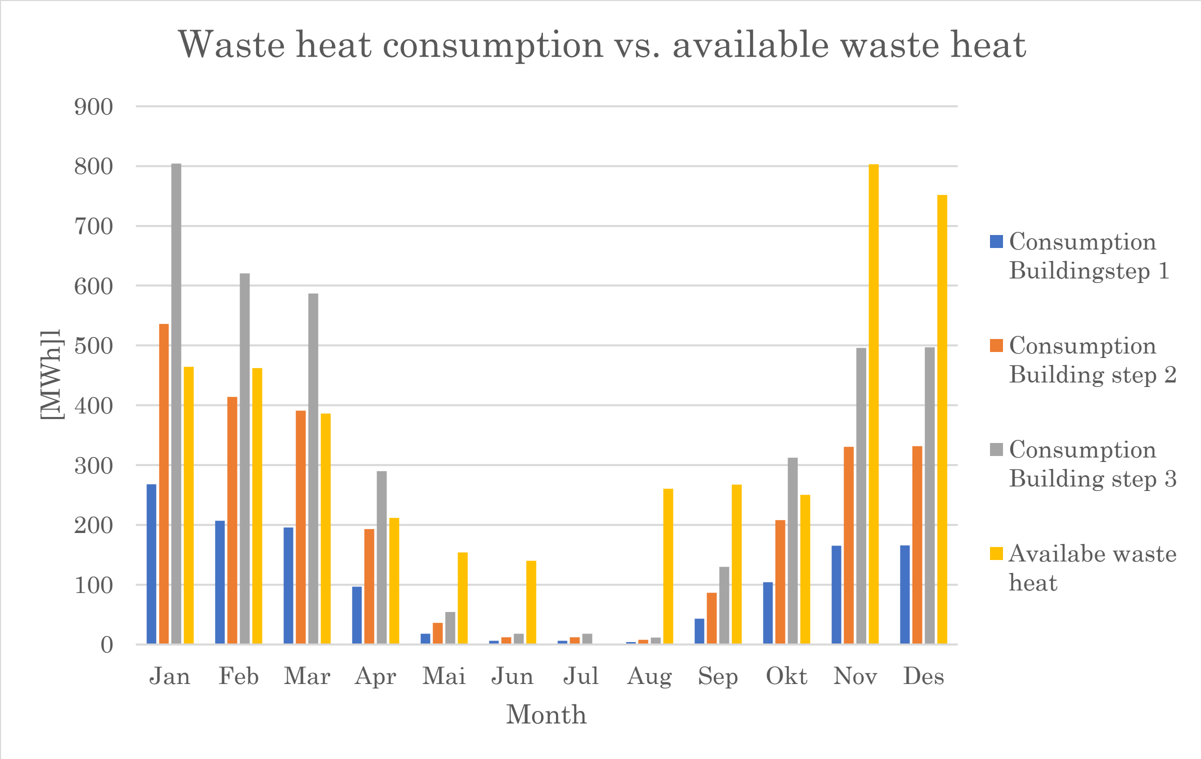

FIGURE 43: WASTE HEAT CONSUMPTION COMPARED TO AVAILABLE WASTE HEAT WHEN GREYWATER IS USED AS INDEPENDENT HEAT SOURCE FOR CO2 HEAT PUMP.

Figure 43 shows that by including greywater the waste heat from the skate rink is enough to cover the demands of all the buildings in building step 1 except for July. It is also enough to cover most of the heating demands of building step 2, except January, March and July. In building step 3 however, there are 6 months during which the consumption exceeds the available waste heat.

The downside to using greywater as a heat source while having need for SH is that the return temperature in the LTTG will sometimes be too high. If no energy is rejected through Heat exchanger A, the average return temperature will be the same as the average exit temperature from the floor heating system, $35°C$. It is therefore probably best to use greywater as an alternative heat source in the summer when there is a low need for SH and no available waste heat. This is further supported by Table 28 showing greywater and space cooling together is enough to cover DHW production from May to August. However, cooling demand and greywater production is not predictable. It is possible that the production of greywater coincides with the need for cooling, and since they are individual sources the heat pump cannot extract heat from both at once. Another possibility is that there are times during the day when there is no available greywater nor cooling demand, which means that the heat pump must use waste heat from Leangen Sports Facility. It is therefore unrealistic to expect both sources to be used to their full potential. Thus, despite the information given in Table 28, it should be assumed that there will be need for waste heat for DHW from the skate rink during May-August.

An alternative could be to simultaneously use LTTG and greywater. The valve controlling the water flow through the greywater tank could be switched to a bypass valve. The bypass valve can split the water flow sending some of the water to collect heat from greywater and the rest to absorb heat from the LTTG. Figure 44 presents a sketch of this proposed solution.

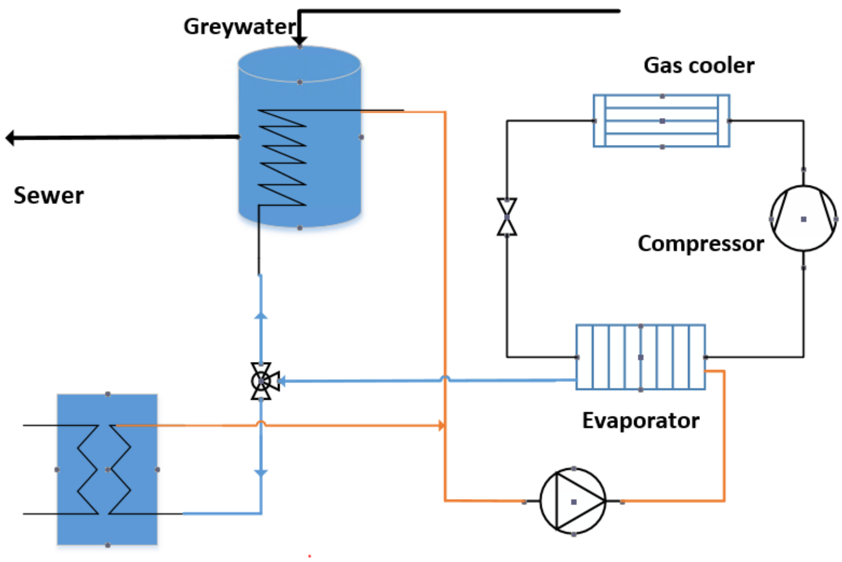

FIGURE 44: SIMULTANEOUSLY DRAWING ENERGY FROM LTTG AND GREYWATER FOR DHW PRODUCTION

The system shows that the water flow splits at the 3-way-valve. After the valve, the two streams go separate ways to collect heat before being reintroduced and flow together to feed the evaporator. The temperature of the greywater varies, making it hard to predict the resulting temperature when the two streams meet. Thus, making it hard to adjust the mass flow to fit the evaporation temperature.

4.3.1.2 Greywater connected in series with LTTG

The second option regarding the connection of the greywater tank is to connect it in series with Heat exchanger A. This option takes into consideration the unpredictable temperature of the greywater while working with a constant speed of the pump circulating the evaporator water. In normal mode, the evaporator requires an inlet temperature of $35°C$. If the temperature in the tank is low, it won't be able to offer a high enough temperature lift. A lower temperature lift means less absorbed energy. Thus, the energy supplied to the evaporator by the greywater alone will not be enough. The greywater tank is therefore connected in series with the Heat exchanger A, supplying heat from the LTTG.

Figure 45 shows that the average expected exit temperature from Heat exchanger A is $35°C$. Since Heat exchange A is the main heat source for DHW, the heat pump is designed for an evaporator inlet temperature of $35°C$. If the greywater cannot lift the water temperature all the way to $35°C$, Heat exchanger A will cover the rest. If the tank is drained and greywater does not provide any heat, everything will be covered by Heat exchanger A. If greywater alone manages lift the water to $35°C$ the ΔT in Heat exchanger A will be zero and no waste heat will be absorbed. Thus, the system will always collect the available energy from greywater and when that is not enough, the LTTG will cover the rest.

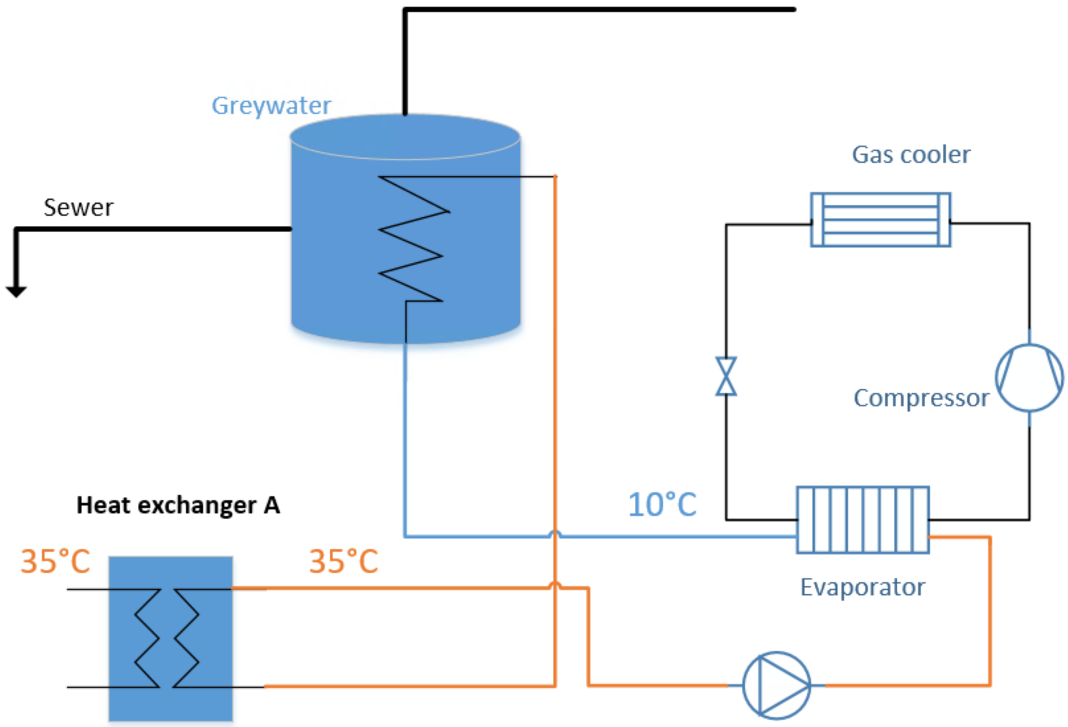

FIGURE 45: GREYWATER TANK CONNECTED IN SERIES WITH HEAT EXCHANGER A

To explore the full potential of this system it is assumed a scenario in which the greywater tank is closed off, containing the equivalent of an average day's greywater production. The initial temperature in the tank is the average greywater temperature, $35.33°C$, as presented in Table 6. It is assumed that the exit temperature from Heat exchanger A is always $35°C$. Thus, the evaporator always has an inlet temperature of $35°C$.

Figure 46 shows the energy supply distributed amongst greywater and the LTTG. Initially the greywater possesses high enough energy to provide a temperature of $35°C$. This is illustrated in the figure by the blue graph providing all the energy during the first hour. After draining the greywater of energy, the temperature decreases. The lower the temperature of the greywater becomes, the more energy is automatically drawn from the LTTG. This is illustrated in the figure by the proportional increase of the yellow graph to the decrease of the blue graph.

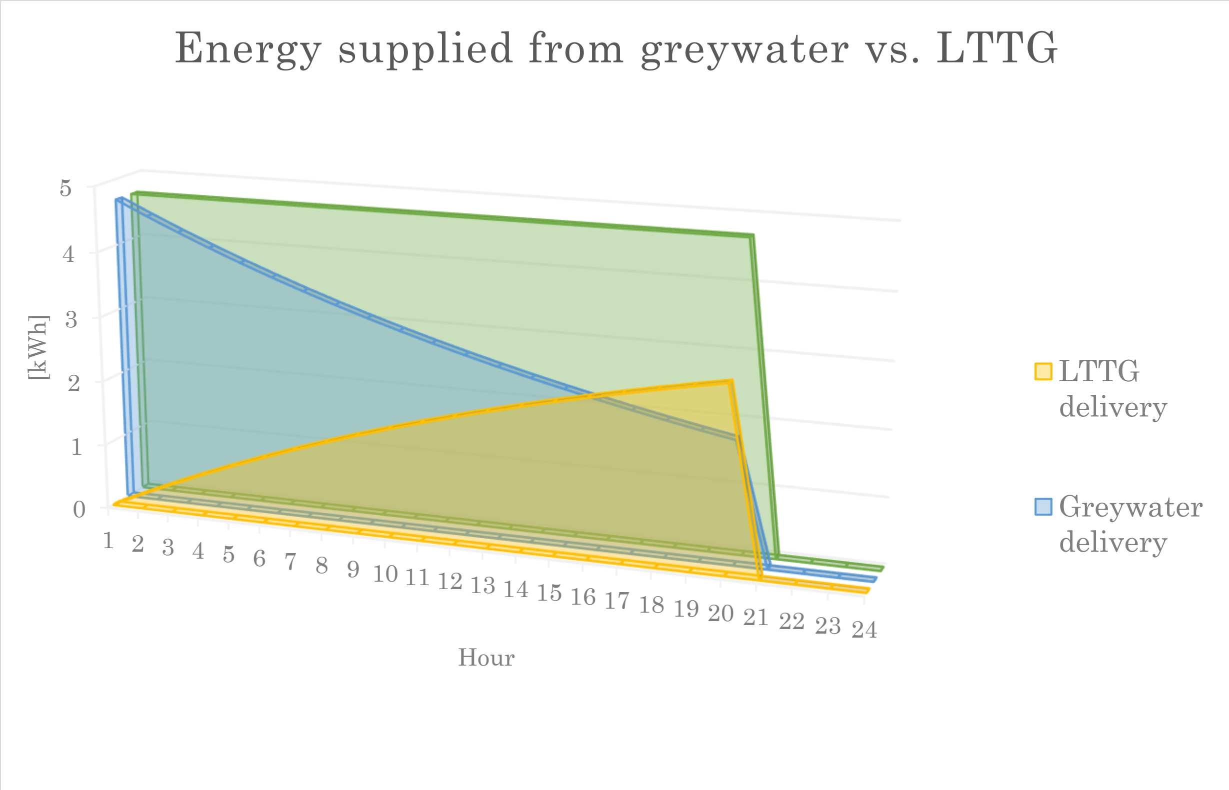

FIGURE 46: ENERGY SUPPLY DISTRIBUTION BETWEEN LTTG AND GREYWATER

Calculations regarding the energy supply by greywater in such a system is presented in appendix A. With this approach, when the hot water demand is covered, the temperature in the greywater tank is reduced to 19.5$°C$. Table 29 shows the energy delivered by greywater and LTTG.

TABLE 29: TOTAL DAILY ENERGY SUPPLY FROM GREYWATER AND LTTG

Parameter	Value
Energy delivered by greywater [kWh]	61.54
Energy delivered by the LTTG [kWh]	32.27
Total delivered energy [kWh]	**93.80**

Figure 47 presents the consumed waste heat compared to the available waste heat when greywater is installed in series with Heat exchanger A.

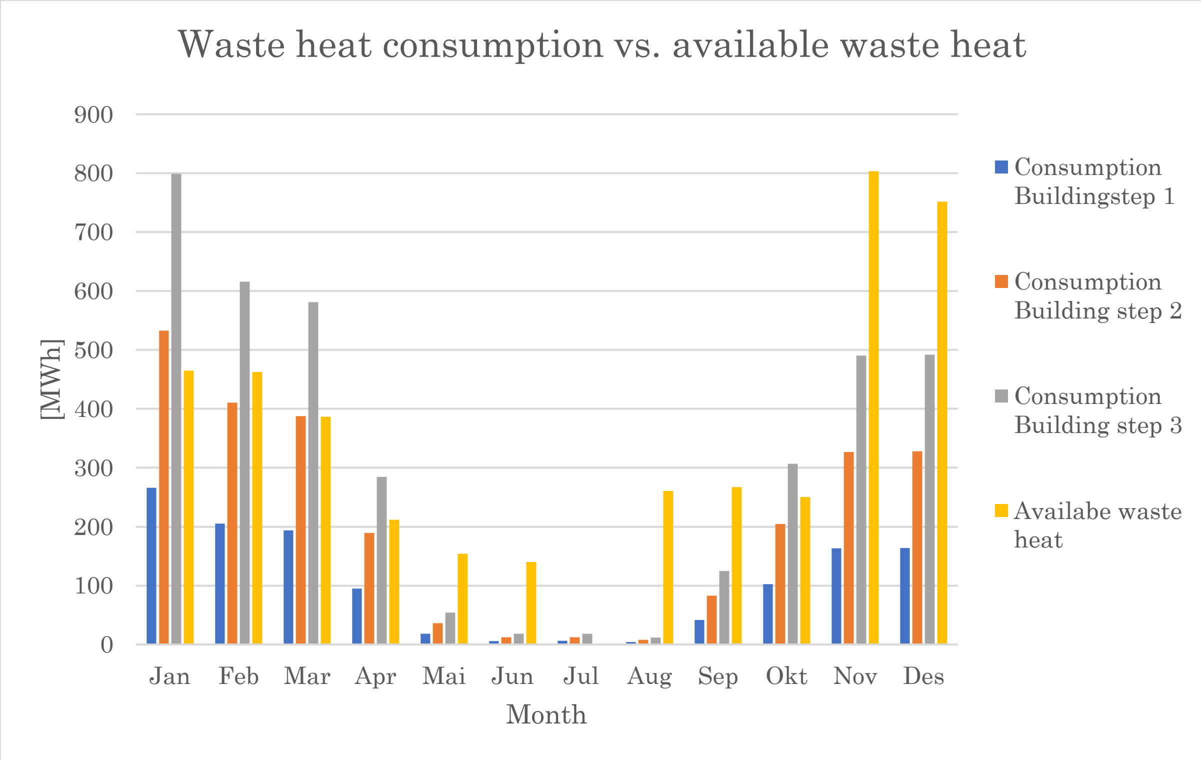

FIGURE 47: WASTE HEAT CONSUMPTION COMPARED TO AVAILABLE WASTE HEAT WHEN GREYWATER IS CONNECTED IN SERIES WITH LTTG AS HEAT SOURCES FOR CO2 HEAT PUMP.

By comparing the results from Figure 47 and Figure 43, it can be concluded that the potential for energy recovery in the two systems is very similar. This is further supported by comparing Table 27 and Table 29 showing that connecting greywater in series only provide 2kWh extra per day compared to if greywater is installed independently. The downside to using such a system is much the same as when installing greywater as an individual heat source. When the majority of DHW energy is supplied by greywater the return temperature of the LTTG is at risk of being too high. If the return temperature to the skating rink is too high, the supply temperature increases and grows too high.

4.3.2 Central accumulation of greywater

Instead of collecting the greywater at each individual building and connecting the greywater tank directly to the heat pump, it might be possible to collect it largescale in a centralized tank. By collecting Leangens collective production of greywater in a large centralized tank and connecting it to the LTTG via a heat pump, the heat pump can extract the greywater energy and feed it directly into the LTTG. Thus, when there is a shortage of waste heat available from Leangen Sports Facility the LTTG imports energy from the accumulated greywater. Figure 48 displays the principle of this approach.

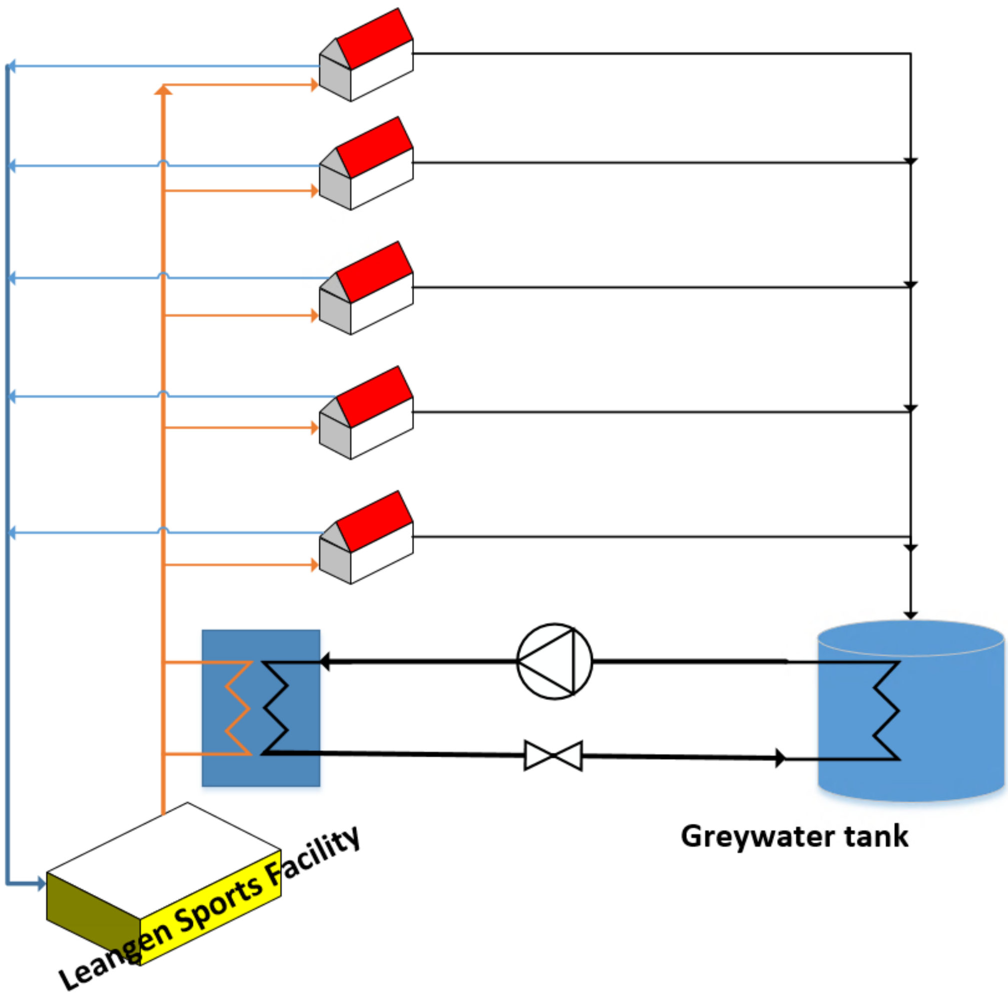

FIGURE 48: CENTRAL ACCUMULATION OF GREYWATER AT LEANGEN

The potential energy extracted from a central pool of greywater depends on how much one can expect to reduce the greywaters temperature. It is therefore assumed that the lower limit of the greywater temperature is equal to the city water temperature, $8°C$. An average temperature of $35.33\ °C$ and a lower limit of $8°C$ means a ΔT of 27.33 K. Table 30 presents the resulting energy potential as well as the parameters from which the potential is calculated. Since the energy contribution from greywater is imported by way of a heat pump, the total energy supply will be even larger than what is presented in Table 30. Equation (12) shows that the total energy output from a heat pump is the sum of the compressor input plus the evaporator input. The energy of the greywater itself equals the potential energy supplied to the evaporator. Assuming the heat pump has an average COP of 4, the potential energy contribution from greywater is presented in Table 31.

TABLE 30: ENERGY POTENTIAL IN GREYWATER ACCUMULATED IN A CENTRAL TANK AT LEANGEN.

Parameter	Value
Greywater mass per building [kg]	3416
Number of buildings	86
$C_p\ [\frac{kJ}{kgK}]$	4.18
dT [K]	27.33
Daily energy potential [kWh]	**9322.44**

TABLE 31: TOTAL ENERGY CONTRIBUTION POTENTIAL FROM GREYWATER ACCUMULATED IN CENTRAL TANK AT LEANGEN, INCLUDING COMPRESSOR INPUT

Parameter	Value
Daily energy potential [kWh]	9322.44
COP	4.00
Total potential energy contribution [kWh]	**11653.05**

By importing the greywater energy into the thermal grid, it becomes part of the LTTG waste heat. Since the greywater is accumulated in a central pool, the energy distribution system at each individual building can remain the same as in the initial system. The energy consumption from the LTTG at each building will thus be the same as in Table 22. Figure 49 compares the waste heat consumption with the available waste heat when greywater is included.

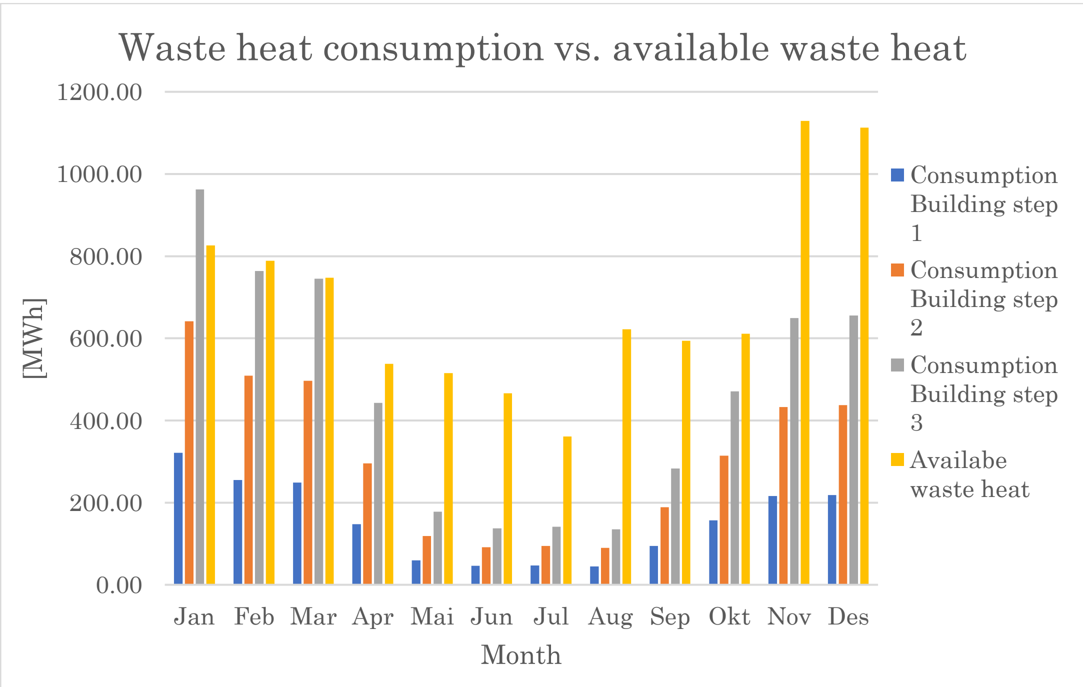

FIGURE 49: WASTE HEAT CONSUMPTION COMPARED TO AVAILABLE WASTE HEAT WHEN GREYWATER ENERGY IS IMPORTED INTO THE LTTG

The figure shows that by supplying the LTTG with energy form both greywater and the skate rink, the only month in building step 3 with an energy deficit is January.

The downside to accumulating the collective greywater at Leangen is the vast infrastructure required to accomplish it. It requires a specialised drainage system in which the desired greywater is separated from the unwanted greywater. It also needs a distribution system which transports the desired greywater form each building to the accumulation tank/pool. Such a system has not been thoroughly explored and the science behind it is scarce. The results in this chapter shows that the potential is vast. However, the results are products of assumptions and are shallowly based on energy calculations which might prove to be false if explored deeper. The waste energy within greywater is nevertheless a large source of domestic energy loss and should be subject to further research.

4.4 Additional energy sources for the LTTG

The energy sources explored in this project are Leangen Sports Facility and greywater. Even though these show great promise, there is not enough available energy to securely cover the expected heating demand. This means that more sources should be included.

The construction at Leangen is expected to last 20+ years. The building plans include three building steps, each expected to last approximately 7 years. The results presented the previous subchapters show that if the use of greywater proves a reliable source, the heating demand during the first two building steps is more than accounted for. If the infrastructure for LTTG is in place, connecting additional energy sources is a simple matter. Since the heating demand for building step 1 and 2 is already in place, the system is secure for the next 14 years. This gives ample time to explore the possibility of other sources nearby that can be connected to the LTTG.

Another possibility is to connect the LTTG to the DH system. Since DH has a typical operational temperature of $70°C$, the energy transfer from DH to the LTTG is easily accomplished. However, the main goal for this project is to find ways of covering the heating demand at Leangen with waste heat. Importing DH is therefore deemed counterproductive and should only be used in emergencies. Figure 50 shows how the LTTG might look with additional heat sources.

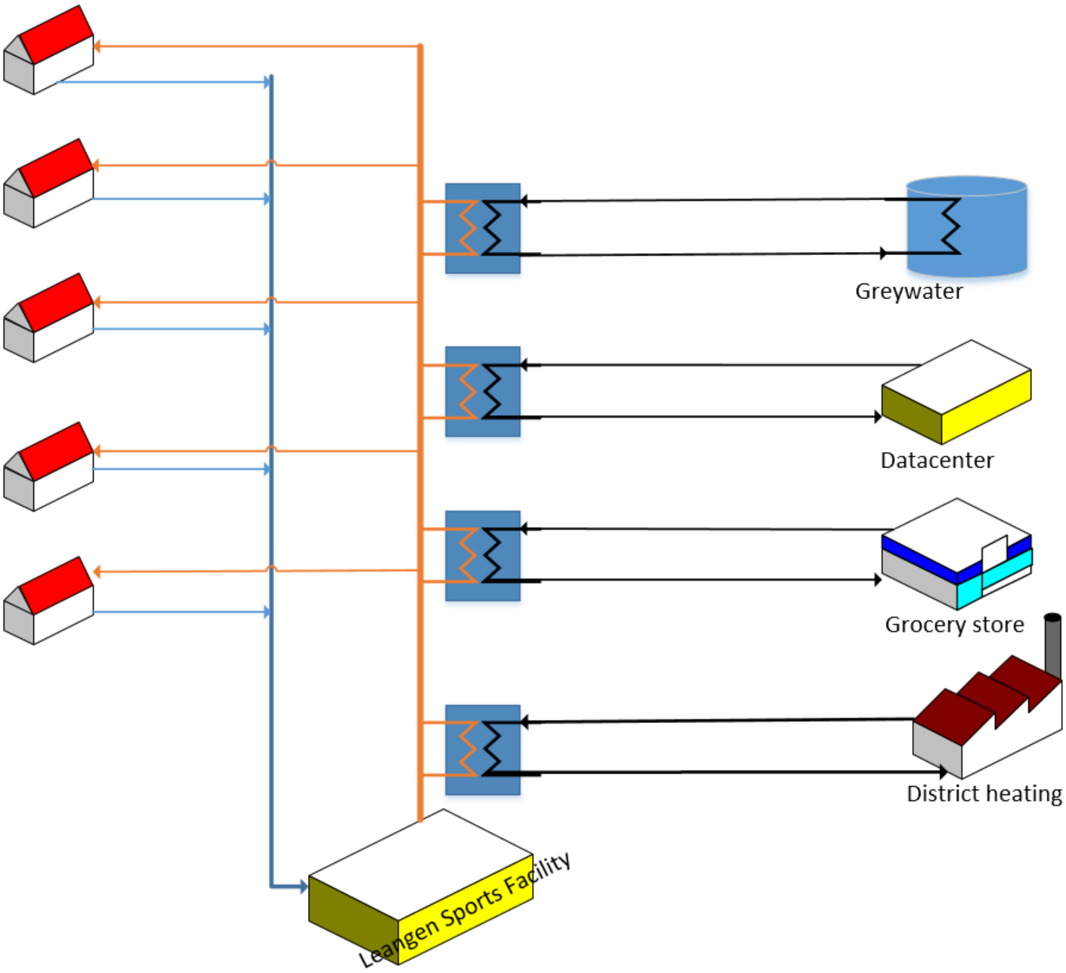

FIGURE 50: LTTG CONNECTED TO DIFFERENT WASTE HEAT SOURCES

5. Conclusion

With the initial system, the energy from the skating rink alone is not enough to cover the heating demands at the future residential area of Leangen. The need for waste heat from the skating rink can be greatly reduced by collecting the individual buildings greywater and using it as an additional heat source for DHW. However, even with the combined energy from the skating rink and greywater for DHW, the heating demands are not covered. Applying greywater in these manners restrict the lower limit of the grey water temperature, severely reducing the potential for energy recovery. When applying greywater as an independent heat source for DHW, the lower temperature limit is set to $20°C$. The temperature is reduced to $19.5°C$ by applying greywater as a heat source in series with the LTTG. By importing greywater energy directly into the LTTG however, the heat pump linking the greywater to the LTTG is assumed to be able to drain the greywater temperature as low as $8°C$. This greatly increases the greywater energy recovery potential. It also allows greywater to be used for both SH and DHW, while the other applications only serve to cover DHW. It is therefore concluded that a centralizes accumulation of greywater is the most desirable option.

Even with greywater as an additional source for the LTTG, the heating demands are not fully covered. There is still one month where the needs outweigh the supply. This implies that energy occasionally must be imported from DH. The overall aim is for the heating demands to be fully covered by waste heat. Therefore, the surrounding area should be further explored for additional sources.

The results show that when applying greywater directly into the LTTG and assuming a lower greywater temperature of $8°C$, the system has a large excess of waste heat from April to December. This allows for the possibility of storing the excess heat in a thermal storage. If this proves to be a realistic endeavour, the excess energy in December can be stored and used during the expected waste heat deficit in January.

6. Further work

One suggestion for further work is to further explore the option of greywater. The presented calculations regarding greywater has been mere energy calculations. To better assess greywater as a realistic heat source, it should be subject to closer examination.

Another suggestion is to examine the potential for thermal storage. The results show that, if applying greywater to the LTTG, there is a vast excess of waste heat from April to December. If this can be stored and recovered at a later point, the system would theoretically be self-sufficient, relying only on waste heat for its heating demand.

The models constructed in this assignment are mostly based on assumptions and monthly averages. Looking at the data with a day-to-day approach and seeing the deviation from the current results can be interesting. Also, applying an hourly load profile to the use of space heating and cooling will produce more realistic results regarding the applicability and profitability of the proposed models.

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Appendix A

Calculations regarding the installment of greywater in series with Heat exchanger A.

The calculations regarding greywater in series are based on a hypothetical situation in which the greywater is in a closed room where nothing pours out and nothing is added throughout the day. This closed tank contains the average daily greywater production per building which is 3416 l. The initial temperature of the greywater is the average greywater temperature from Table 6, which is 35.32$°C$. For simplicity the temperature is set to 35$°C$. The calculations are performed in excel, but the step by step method will be explained here. The calculations are largely based on temperature lift. Assumptions are listed below

- The lower temperature limit of the greywater is equal to the outlet water of the evaporator which is 10$°C$.
- The temperature inlet of the evaporator is always 35$°C$
- Average greywater temperature is 35$°C$, while lower limit is 10$°C$.
- As the temperature of the greywater is reduced and it is no longer able to provide 35$°C$, Heat exchanger A steps in and makes sure the water entering the evaporator is always 35$°C$. The total temperature lift provided to the water in the evaporator circuit will therefore always be 25K.

1. The greywater tank contains 3416 l(kg) of greywater, and has a temperature of 35$°C$ at the start of the day (00:00). With a lower limit of 10$°C$ this equals 356 972 kJ of available energy.

   During the first hour (from 00:00 to 01:00), the DHW demand is 17 414 kJ. This is subtracted from the available energy in the greywater tank. Therefore, at the start of the second hour of the day, the greywater tank only contains 340 091kJ of available energy.

2. By reducing the energy in the greywater tank, the temperature decreases. From equation (1), where $m = 3416kg$ and $C_p = 4.18\frac{kj}{kg*K}$, the new ΔT is now 23.81K. This shows that after the first hour greywater can no longer provide the 25K temperature lift.

   $\frac{23.81K}{25K}$ = 0.95 → Greywater can now only provide 95% of the DHW demand.

   The DHW demand during the second hour is 16 832kJ. The energy supplied by greywater is then $0.95 \cdot 16\ 832kJ = 16\ 031\ kJ$. The available energy in the greywater tank at the start of the third hour is therefore 324 060kJ. The remaining 802 kJ is supplied by Heat exchanger A.

3. Equation (1) shows that ΔT at the start of the third hour is 22.69K. Therefore, greywater can only provide $\frac{22.69}{25} = 0.91$ of the DHW demand of the third hour.

By using this approach, the energy delivered by greywater is calculated by equation (14), while the energy drawn from LTTG is calculated by equation (15).

$$(\boldsymbol{DHW\ demand}) \cdot \frac{\boldsymbol{\Delta T}}{\boldsymbol{25K}} \tag{14}$$

$$(\boldsymbol{DHW\ demand}) \cdot (\boldsymbol{1} - \frac{\boldsymbol{\Delta T}}{\boldsymbol{25K}}) \tag{15}$$

Thus, as the day progresses, and more energy is drawn from the greywater, its temperature decreases. As the greywater temperature decreases, the share of DHW demand covered by greywater is lowered while the share covered by Heat exchanger A increases.

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