Master's thesis

NTNU
Norwegian University of Science and Technology
Faculty of Information Technology and Electrical Engineering
Department of Energy and Process Engineering

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Dynamic energy system with a local LTTG at Leangen

Using greywater as a source for a CO_2 heat pump with a flooded evaporator

Master's thesis in Energy and the Environment
Supervisor: Armin Hafner, Hanne Kauko
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Preface

This master thesis is written during the spring of 2020 at the Norwegian University of Science and Technology. It grants 30 credits and concludes my 2-year master's program.

I would like to thank my supervisor, Armin Hafner, and co-supervisor, Hanne Kauko for being helpful, enthusiastic and always available even when it was not possible to meet. This thesis would not have been possible without you.

Trondheim, June 5, 2020
Vegard Skulstad

Task for master thesis

In a few years, a new building area will be built at Leangen in Trondheim. To supply the area with energy in a cost and energy-efficient manner, a local low-temperature grid must be developed. This grid will utilize excess heat from surplus heat sources nearby and the system will provide both heating and cooling for the building mass.

The objective is to make and analyze a dynamic energy system that can cover heating and cooling demands as well as possibly cover the electricity demand for the heat pump compressor. Waste heat, renewable energy sources, heat pumps, thermal energy storage and backup to cover peak loads are potential sources and components in the system and should be evaluated. In addition, a space cooling device and alternative uses for greywater should be investigated to implement into the dynamic system. If other nearby waste heat sources exist, the potential should be mapped and evaluated.

Because the Leangen area has different energy demands during the year the thesis will also focus on developing a dynamic model of the plant in the object-oriented modeling language Modelica. The results should be analyzed, and the cost and profitability of the solutions should be considered to assess the project's viability.

1. Literature review, e.g. Thermal comfort, space cooling, heat pumps
2. Develop model(s) of the Leangen energy system
3. Perform simulations with varying thermal conditions.
4. Analyze results in terms of system performance and energy consumption
5. Make a preliminary cost analysis of the system
6. Summary report (incl. Discussion and conclusion)
7. Propose further work and make a draft version of a scientific paper

Summary

This thesis is a continuation of previous project work and evaluates solutions for using waste heat sources to supply DHW for a building area. The main consideration is a flooded CO_2 evaporator and a DHW heat pump.

Relevant literature on low-temperature district heating, thermal energy storage, and grey-water will be review to give an idea of the concepts in the thesis. Theory and equations for convective and conductive heat transfer as well as heat pumps are also reviewed to understand how the models were developed. In the method section, a system description and explanation of the development of the models in Python and Dymola is included. Heat from the ice skating rink will be used in a natural working fluid heat pump to supply radiant space heating, and during the summer it will supply cooling. During summer mode, when cooling is provided, the heat from the condenser will be transferred to a GW tank. This tank will collect GW from showers, dishwashers, and washing machines which will be used as a source for a DHW CO_2 heat pump. This heat pump has a flooded evaporator with a coiled pipe to increase heat transfer. To avoid fouling due to impurities the GW must have a high velocity and has a backup coil since there is not enough GW to supply all the heat. The second coil uses the outlet from the space heating heat pump evaporator as a source.

The demand for hot water has a maximum daily demand of 1165 kWh and the heat pump is designed to supply 60 kW because it will operate 20 hours every day. The capacity of the evaporator has to be 41,8 kW

The evaporator was tested for a range of velocities, but in order to supply sufficient heat to the heat pump, the velocity was chosen to be $4m/s$. At this velocity, the overall heat transfer coefficient is 3900 W/m^2K and must be $18m$ long. The minimum water outlet temperature is $3°C$ before excessive pipe length is needed to reduce the temperature further. With this temperature drop it can provide 51 kW of heat and the coil experiences a pressure drop of 7,2 bar. The most efficient heat transfer happens when using a single-coil compared to several smaller coils because the decrease in diameter will increase the length of the pipe and the pressure drop. The heat pump model resulted in a COP of 3 and a DHW temperature of $75°C$. This means that the gascooler capacity is 60 kW. The GW in this model transfers 45,8 kW with an outlet temperature of $5{,}8°C$.

A preliminary cost analysis has been made and the saving potential is 8420 kr every month when considering the price of electricity to the DHW heat pump compared to supplying the heat from the district heating network.

Oppsummering

Denne masteroppgaven er en fortsettelse av en tidligere prosjektoppgave og evaluerer bruken av spill-varmekilder for å lage varmtvann til et leilighetsområde. Hovedfokuset er på en oversvømmet fordamper og en CO_2 varmepumpe for varmtvannsberedning.

Relevant litteratur om lav-temperaturs fjernvarme, termisk energilagring og gråvann vil bli gjennomgått som en innføring i prinsippene bak oppgaven. Teori om konvektiv og konduktiv varmeovergang og varmepumper vil også bli gjennomgått for å gi en forståelse for hvordan modellene ble laget. I metodedelen vil det være en systembeskrivelse og forklaring av modellutviklingen i Python og Dymola. Varme fra en nærliggende skøytehall vil bli brukt i en varmepumpe med et naturlig arbeidsmedium som gir romoppvarming til leiligheten på vinteren og kjøler dem ned på sommeren. Om sommeren vil varmen fra denne varmepumpen bli overført til en gråvannstank som samler opp gråvann fra dusj, vaskemaskiner og oppvaskmaskiner. Gråvannet vil bli brukt som varmekilde til en CO_2 varmepumpe som har en oversvømmet fordamper. Denne fordamperen er valgt for å øke varme brukt til fordampning og røret er spoleformet for å øke varmeoverføringen mer. Et ekstra rør vil også være i fordamperen der vann fra utløpet fra romoppvarmings varmepumpen er brukt som kilde.

Det daglige behovet for varmtvann er 1165 kWh og varmepumpen er derfor designet for å tilføre 60 kW siden den er på 20 timer om dagen. Fordamperen må derfor tilføre 41,8 kW.

Fordamperen ble testet for en rekke hastigheter og må holde en minimums hastighet på 4 m/s for å tilføre nok varme. Med denne hastigheten er det totale varmeovergangstallet 3900 W/m^2K og må være 18m langt. Utløpstemperaturen er da 3$°C$ fordi lengden øker betydelig hvis det skal kjøles ned ytterligere. Med dette temperaturfallet overføres 51 kW med varme og trykktapet vil være på 7,2 bar. Det er mer effektivt med et enkelt rør enn å dele det opp i flere små. Dette er fordi rørlengden blir lenger for mindre rør siden diameteren minker. Det fører også til høyere totalt trykktap som betyr at mer pumpekraft trengs. Varmepumpemodellen i Dymola resulterte i en COP på 3 og varmtvannet når en temperatur på 75 $°C$. Gasskjøleren har dermed en kapasitet på 60 kW. Gråvannet i denne modellen overfører 45,8 kW og har en utløpstemperatur på 5,8$°C$.

En grov kostnadsanalyse ble gjennomført og det er mulig å spare 8420 kr hver måned hvis en sammenligner strøm til varmepumpen med å overføre all varmen fra fjernvarmenettet.

Nomenclature

Abbreviation

4GDH	4th Generation District Heating
COP	Coefficient of Performance
DH	District Heating
DHW	Domestic Hot Water
GW	Greywater
GWHR	Greywater Heat Recovery
GWP	Global Warming Potential
IHX	Internal Heat Exchanger
RES	Renewable Energy Sources
TES	Thermal Energy Storage

List of Symbols

α	Thermal diffusivity $[m^2/s]$
ΔT	Change in temperature [K]
δ	Boundary layer thickness
$\dot{m}$	Mass flow [kg/s]
$\dot{V}$	Volumetric flow rate $[m^3/s]$
μ	Dynamic viscosity $[Pa \cdot s]$

ν	Kinematic viscosity $[m^2/s]$
ρ	Density [kg/m^3]
c_p	Specific heat capacity [J/kg$\cdot$ K]
e_{pi}	Error [-]
f/ζ	Friction factor [-]
h	Convection coefficient $[W/m^2K]$
h_n	Enthalpy $[kJ/kgK]$
k	Thermal conductivity $[W/mK]$
K_p	Proportional gain [-]
Nu_x	Nusselt number [-]
P	Pressure [Pa]
Pr	Prandtl number [-]
Q	Energy potential [W][J]
$q_s^"$	Heat flux $[W/m^2]$
R	Thermal resistance $[K/W]$
r_{pi}	Reference value [-]
Re_x	Reynolds number [-]
T_i	Integration time [-]
U	Overall heat transfer coefficient $[W/m^2K]$
u	Velocity $[m/s]$
u_{pi}	Controller output [-]
W	Work $[kW]$
V	Volume [m^3]

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Chapter 1

Introduction

1.1 Background

A lot of energy is today wasted by releasing heat made by industrial processes into the air. Utilizing this waste heat can reduce energy consumption, increase energy efficiency, and reduce the environmental impact efficiency by utilizing energy that would otherwise be wasted.

1.2 Structure and goals

The primary goal of this thesis is to evaluate a CO_2 heat pump with a flooded evaporator with GW as its heat source to produce DHW for the building area. Heat from a nearby ice skating rink, transported with a low-temperature thermal network will be used to provide space heating and as a backup to the DHW heat pump. To answer evaluate this the thesis has been structured in the following way:

1. Review of relevant literature and theory
2. Developing models for the flooded evaporator and heat pump in Python and Dymola
3. Analyze the results and do a preliminary cost analysis
4. Propose further work

The thesis consists of 7 chapters where chapter 7 is further work and the literature review begins in chapter 2.

1.3 Limitations

The limitations of this thesis are mainly the uncertainty in the system solution chosen. The total cost of the system is unknown and this impacts the ability to accurately determine the economic benefits or drawbacks.

Chapter 2

Literature review and theory

This thesis is a continuation of previous project work and some of the relevant literature is taken from there. It will consider general literature about waste heat harnessing and TES to introduce relevant concepts used in the system description. It will also include relevant theory and equations used in the calculations.

2.1 4th Generation district heating (4GDH)

To meet the challenge of more energy-efficient buildings, 4GDH is a part of the solution [15]. Present district heating (DH) uses water as a carrier and often has a supply temperature below 100°C. The general development trend is to lower the supply temperature that in turn will decrease the grid losses. An advantage of 4GDH is its ability to connect with other sources like geothermal and solar thermal. Another possibility is waste heat from industrial processes and cooling such as supermarkets or refrigeration facilities. This will make the system more complex and needs detailed planning.

New buildings have better insulation and require less heating which means more buildings can be connected to the DH network. According to Lund et al. it is possible to reduce the space heating demand to a level equivalent to hot water heating. Because of this, there will be a better balance between energy use during summer and wintertime. Renewable energy sources (RES) or heat recycling can then deliver the energy needed at a lower cost. A low-temperature grid can operate with a supply temperature of 50°C and return of 20°C that can, in combination with lower dimension twin pipes and lower peak flow, reduce heat losses by a factor of 4.

Low-temperature DHW is also a possibility despite the Legionella growth in this temperature range. By using substations without storage of DHW at the end-user and small water volume between the heat exchanger and the taps the bacteria growth can be limited. For space heating, a supply temperature of 40°C and a return of 20-22°C can be used to get a floor or wall temperature slightly higher than room temperature. If the supply temperature is lower than 30°C, it can be used to preheat DHW before the temperature is further lifted by for example a heat pump.

Implementation methods of low-temperature district heating to residential area

In a paper from 2012 written by Hongwei Li and Svend Svendsen, a low-temperature district heating network is simulated [12]. The supply and return temperatures are 55°C and 25°C respectively and the hypothetical network is supplying 30 low energy detached buildings. A temperature drop of 5°C is assumed and since the lowest required tapping temperature in the kitchen is 45°C the supply temperature is almost at the lowest limit. Floor heating is used in the bathroom while low-temperature radiators are used for the rest of the buildings.

The paper investigates two types of in-house substations. Instantaneous heat exchanger and district heating storage tank. To avoid Legionella the storage tank is on the primary side of the network. In figure 2.1 it is possible to see that the heat exchanger prepares the DHW directly and has the advantage of being smaller in size compared to the tank. It is important to design it for maximum draw-off from the taps. This will result in increased heat losses because it will, for the most part, be over-dimensioned. The left side of the figure shows the schematic of the tank system. It functions similarly and uses a heat exchanger since the storage tank is on the primary side of the network. Because of the storage, it can shave the peaks in DHW demand and therefore reduce the branch pipe diameter.

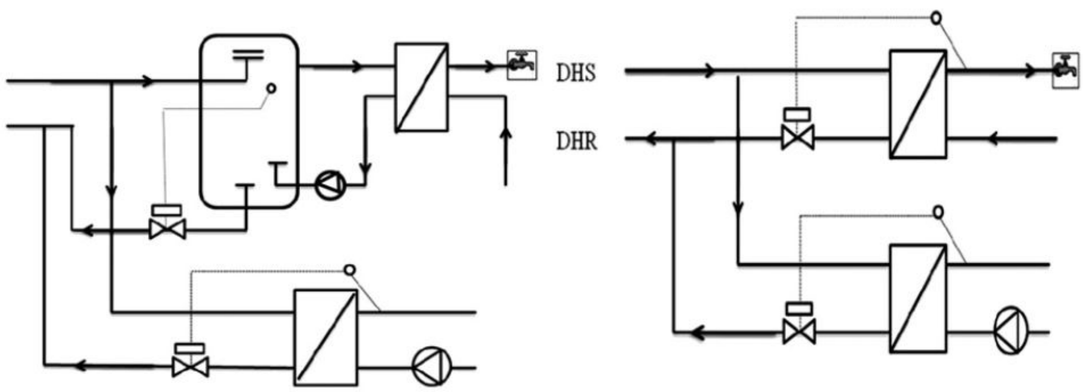


Figure 2.1: Schematic of district heating storage tank on the left and instantaneous heat exchanger on the right [12]

2.2 Heat sources

A waste heat source is a by-product of a process. This can be in the form of gases discharged to the atmosphere, such as hot air from industrial refrigeration or a combustion process. it can also be hot water released into the environment [39].

Greywater heat harnessing

An article from 2018 has done a review of GW heat harnessing for non-industrial purposes [16]. GW enters the sewers with high temperatures and is, therefore, a major source of inefficiency for energy usage in a building. Harnessing this heat can reduce the carbon footprint, increase the share of renewable energy, and revive the concept of ultra-low energy houses. GW is divided into two types with different contents:

- Light or low-load GW (Showers, WC basins, bathtubs)
- Heavy/Dark-high load GW (Dishwasher, kitchen basins, washing machine)

To utilize heavy GW, it is necessary to include grease traps and sludge removal since its contents include oil and grease. The third group of drain water is blackwater. This is toilet water and is not good for heat harnessing due to its low temperature and high contamination.

The usage of water in a building is usually consistent throughout the year but can vary depending on age, gender, and features of the building residents. Some estimates have been done on the average water usage which is 140-150 L/day per person. The average water temperature in the UK is 51,9°C before the drain and GW temperature is usually 5-10°C lower than this. In table 2.1 it is listed how much is used for the different applications per person.

Table 2.1: Average distribution of water in a conventional household if the usage is 150 liter per day per person.

Source	**Amount [%]**	**Temperature [°C]**	**[kg/day]**
Shower	25	40-50	37,50
Bath	8	40-50	12,00
Washing machine	9	30-60	13,50
Dish washer	1	60-85	1,50
Hand wash dishes	4	50-60	6,00
Bathroom hot tap	7	50-60	10,50
Toilet	22	-	33,00
Other (cold taps)	22	-	33,00
Garden	1	-	1,5
Car	1	-	1,5
Total	**100**	-	**150**

The daily user profile of hot water usage is seen in figure 2.2. It is based on a statistical prediction model of the American Water Works Association [19]. For comparison the energy need for an apartment building per m^2 is shown in figure 2.3 [29]. There is a peak in the morning and in the evening for all the different GW sources.

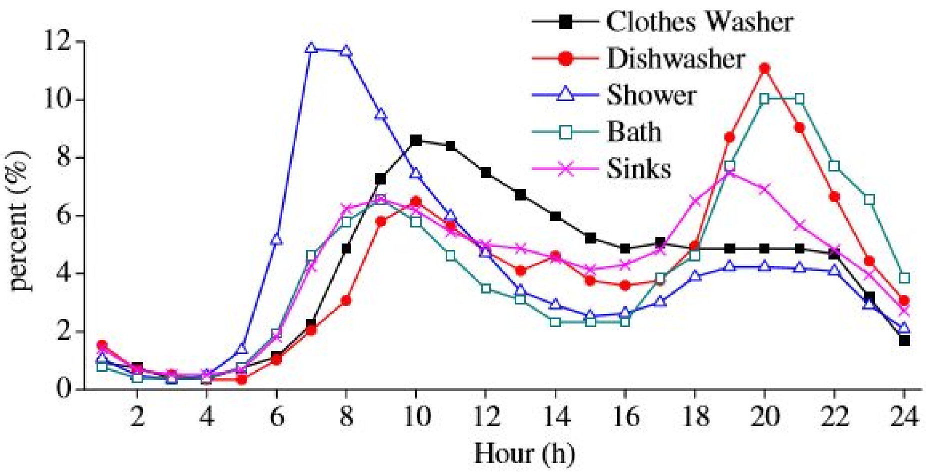


Figure 2.2: User profile of hot water usage [19]

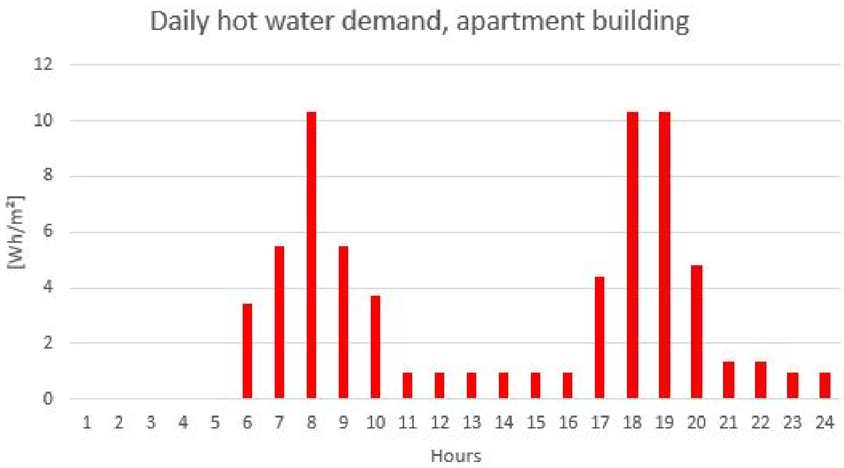


Figure 2.3: Hot water demand for a apartment building [29]

Greywater heat recovery (GWHR) in an American army base

A study has been done on the viability of GWHR in army barracks and dining facilities where DHW is more than 60% of the annual heating demand [38]. The system, shown in figure 2.4, consists of a counter-flow heat exchanger where the GW flows through a vertical pipe wrapped in a copper coil to extract heat.

The drain pipe is designed as a vertical pipe where both sinks and showers are connected. In addition, it is connected to other drains from the rooms below it. The study concludes that the solution is economically feasible for this configuration if the number of floors is more than 3 and the cost of energy is high. When the electricity price is low it would take about 20-30 years to pay back, while if the price was average it would take 5-8 years. For a dishwasher solution, the payback time is considerably lower ranging from below 1 year to about 9 if the electricity cost is low.

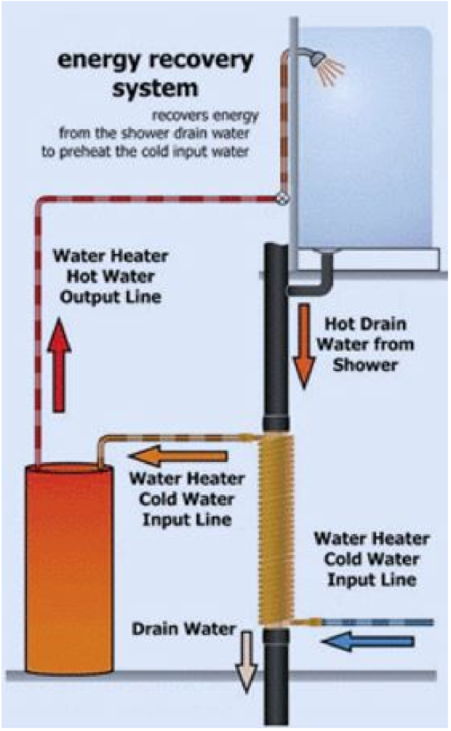


Figure 2.4: A simple energy recovery system installed in a shower [38]

Waste water heat recovery using heat pumps

A washing facility produces 5220 kg/h of water at a temperature of 50-55°C [21]. To find the amount of energy needed to heat the city water from 13,8°C to 50-55°C it is necessary to use the following equation

$$Q = c_p \dot{m} \Delta T \tag{2.1}$$

Where Q is the energy potential in [W], $\dot{m}$ is the mass flow in [kg/s] and c_p is the specific heat capacity in [J/kg · K]. Inserting relevant values will result in 220-248 kW of energy potential.

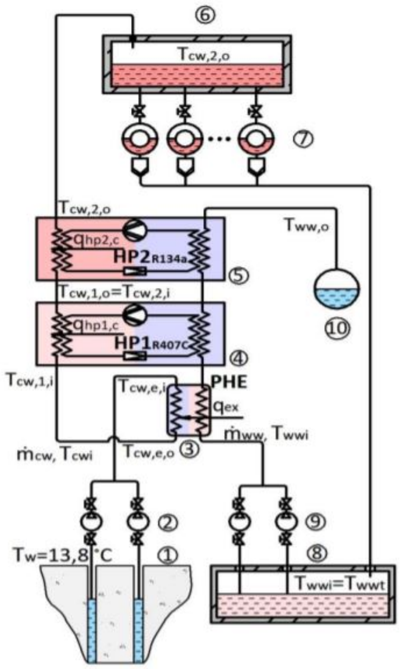


Figure 2.5: Schematic of the heat recovery system using heat pumps in two stages [21]

As shown in figure 2.5 the system has three stages of heat recovery. A plate heat exchanger, HP_1 using higher source temperature and HP_2 using lower source temperature. The first stage is the plate heat exchanger that manages to lift the temperature from 13,8°C to 33,9°C. This gives a COP of 3,19. The second stage with HP_1 further raises the temperature to 46,2°C resulting in a COP of 1,95 for this HP. Water then arrives at the third step and HP_3 and the temperature increases to 54,7°C giving a COP of 1,35.

The total COP of this system is 6,03-6,5. Approximately 85% of the energy demand can be covered by recovering the GW where the remaining power is electric energy input to the system.

Heat recovery in industrial refrigeration

In industrial refrigeration, there are several ways to recover heat. It can be recovered from oil cooling or heat rejection from a fluid-cooled screw compressor, high-stage discharge gas stream from either screw or reciprocation compressors, or an air-cooled condenser [26].

Heat recovery can be divided into high-grade and low-grade recovery. In this example, high grade uses a de-superheater on the refrigerant between the compressor and the condenser. The temperature is usually between 60 and 90°C. The de-superheater is a heat exchanger that heats the fluid by using the heat from the refrigerant. It also reduces the need for cooling water or air by the condenser. Figure 2.6 shows a configuration of such a system. A system like this is often designed to meet hot water demands and if the demand varies a bypass valve is placed before the de-superheater.

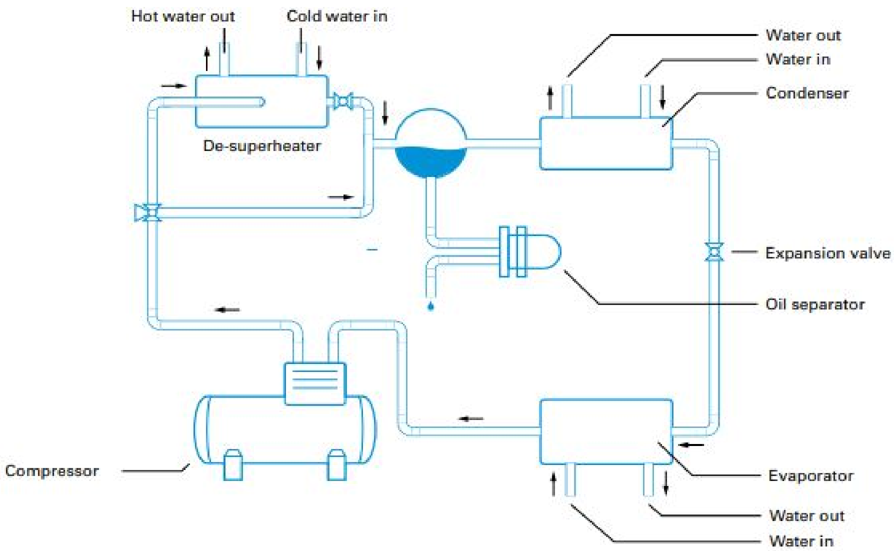


Figure 2.6: An example of how the de-superheater can be placed in between the compressor and the condenser [26]

Low-grade heat recovery is from the condensing of the refrigerant and is usually between 20 to 40°C [26]. The grade of heat will depend on the type of installation and refrigerant used. Heat recovery can be applied to almost all sizes and can recover up to 30% of the cooling capacity if applied to an existing plant. It is not viable to use if the compressor is smaller than 30 kW. In new units, it can recover 100% of the waste heat [5].

2.3 Thermal energy storage (TES)

TES is helpful to increase efficiency and provide thermal energy if there is a mismatch between energy generated and demand. It can be used for several different purposes such as space and water heating, cooling, air-conditioning, etc. [42]. There are two main categories of TES.

- Sensible heat storage
- Latent heat storage

Sensible heat storage store heat by raising or lowering the temperature in a medium. It is common to use rocks, ground, or water as a storage medium. The energy amount put into a sensible TES is proportional to the difference between the storage final and initial temperature, the storage medium mass, and its heat capacity. The heat stored can be calculated by the following formula

$$Q = c_p m \Delta T = \rho c_p V \Delta T \tag{2.2}$$

Q is given in joules, and as shown in section 2.2 it can easily be recalculated to watts by using mass flow instead of mass. Latent heat is the heat released by a phase-changing material, often this change is from solid to liquid. Both of these categories use the basic three processes: Charging, storing, and discharging, see figure 2.7.

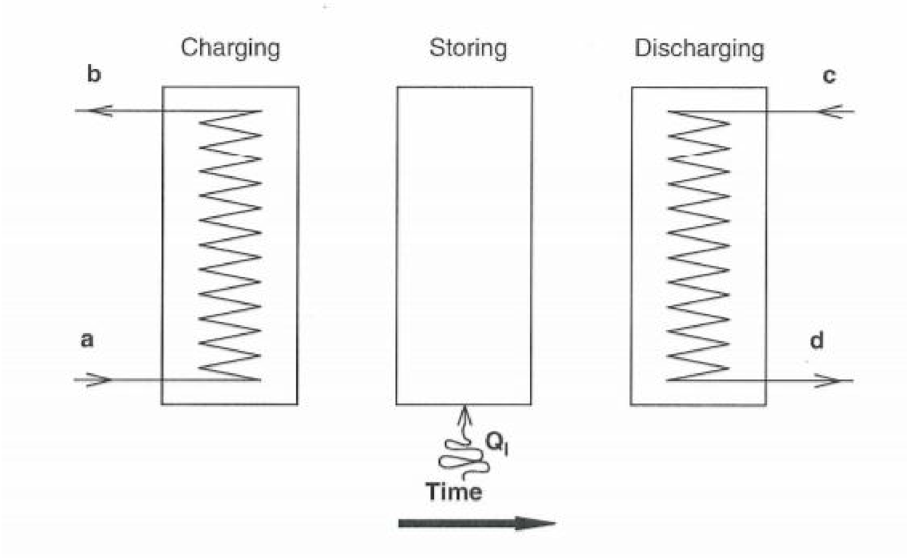


Figure 2.7: Three TES processes: Charging (Left), storing (middle), discharging (right). Q_I is the heat flow which will be positive when it is charging. It will be negative when discharging and releasing to the surroundings [42]

DHW tanks

In the temperature range needed for buildings, it is common to use water as a storage medium. It has a high c_p value and since it is liquid it can easily be pumped and allows a good heat transfer rate. When storing water it is common to use thermally stratified tanks. Stratified means there is a temperature gradient across the tank and the warm water has a lower density than cold water and will float to the top while the cooler water remains below. When designing the tank it is necessary to consider this. If the inlet and outlet are improperly placed it will have a large dead water zone. This is a zone where the water cannot be utilized.

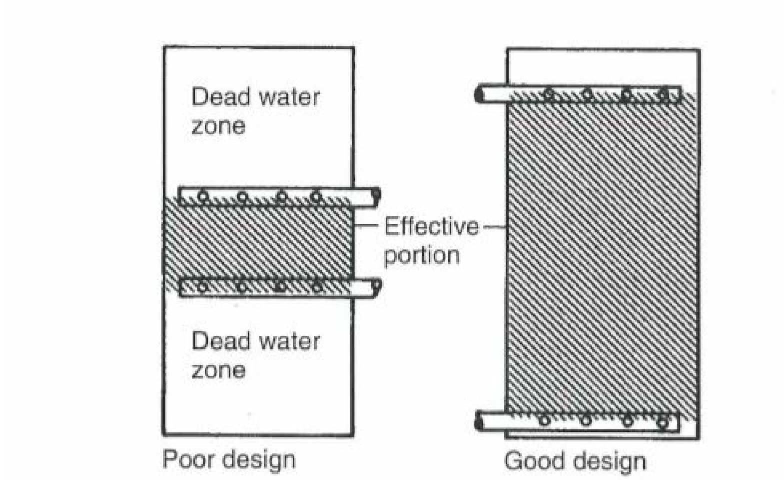


Figure 2.8: Illustration of proper (right) and improper (left) design of a thermally stratified tank

2.4 Radiant floor heating

Radiant floor heating is a method mostly used to heat residential buildings [22]. Today the system consists of usually plastic pipes under the floor made to distribute the heat in the room. The reason for having the pipes in the floor is the angle factor between the person and the radiant heat source. The floor usually has the highest angle factor of the surfaces in the room.

To calculate the acceptable thermal conditions, operative temperature is an important parameter. This temperature is determined by the combination of air temperature and mean radiant temperature. If the air velocity is low, the operative temperature can be calculated as the average of these two. When taking the angle factor into account, the floor heating will need lower radiant temperature than the ceiling to raise the air temperature to the same level. When comparing to a convective heating system that only heats the air, the floor heating can reach the same operative temperature with lower air temperature.

Figure 2.9 shows the vertical air temperature for different systems with a heat flux of 50 W/m^2. The floor heating has an even temperature distribution while the ceiling system has large differences. The basis for measurement was an operative temperature of 22°C.

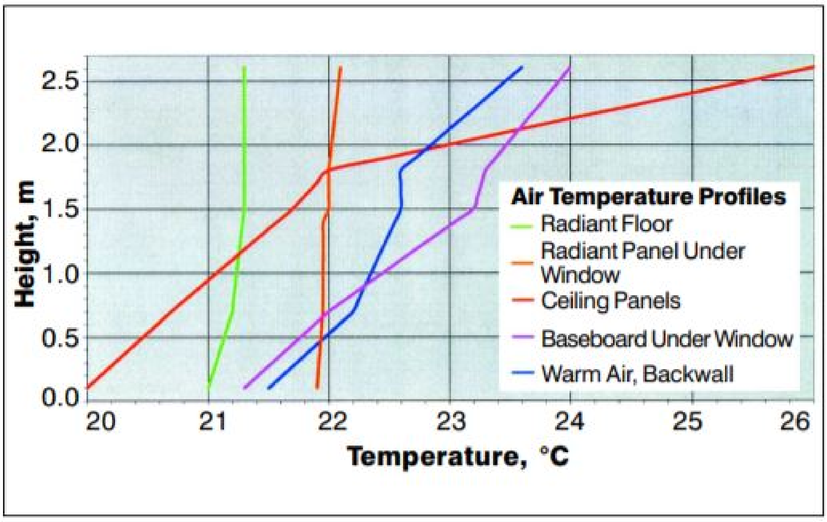


Figure 2.9: Vertical air temperature difference for different heating systems [22]

The floor temperature can be in the range of 19°C minimum for cooling to 29°C maximum for heating. This range will depend on the floor material. The radiant system can also prevent condensation due to higher surface temperatures and it reduces the transportation of dust compared to air heating. When using the system for cooling it can be done without noise and creating drafts.

2.5 Controller

P-controller

A proportional-controller (P-controller) is changing the output proportional to the error [4].

$$u_{pi} = K_p \cdot e_{pi} + u_0$$

$$e_{pi} = r_{pi} - y$$

Where u_{pi} is the output, K_p is the proportional gain, u_0 is the initial output, r_{pi} is the reference, and y is the actual value. For standard controllers, the initial output is set to 50% of maximum. The output can be for example the speed of a motor or the opening of a valve.

PI-controller

A proportional integral controllers (PI-controller) function is to make an additional signal to the output to help make the error become zero. The mathematical formula for the output of the PI-controller looks as follows:

$$u_{pi} = K_p(e_{pi} + \frac{1}{T_i}\int_0^t e_{pi}dt) + u_0$$

Where T_i is the integration time but it does not represent the time it takes to integrate. The integration takes place from when the I-part is connected until it is disconnected which usually is when the regulator is disconnected.

A few rules of thumb about how the I-part and P-part work together:

- If the output is unsteady the K_p should be decreased or the T_i should be increased
- If the dynamic error is large, increase K_p
- If the time to reach 0 error is large, reduce T_i

The dynamic error is the larges error to occur in the system.

2.6 Flooded evaporator

A flooded evaporator ensures a 100% liquid refrigerant to the evaporator and the exiting refrigerant is 50-80% gas [32]. A forced flow evaporator uses a pump or an ejector to move the flow. For a flooded system to work it needs a receiver to separate the liquid from the gas.

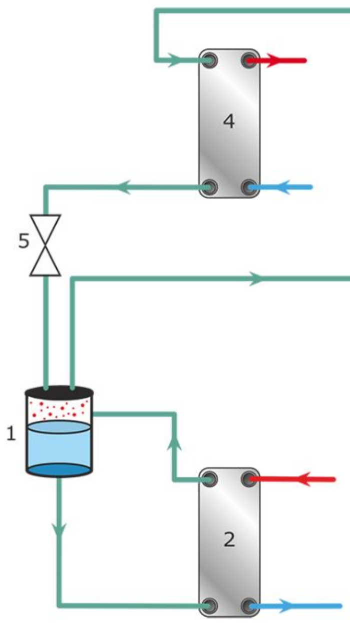


Figure 2.10: Schematic of a heat pump system with a flooded evaporator

In figure 2.10, the separator is marked with the number 1. The bottom part is supplying 100% liquid refrigerant, while the top is 100% gas. An advantage of the flooded system is that because of the separator the problem with poor refrigerant distribution in the evaporator is reduced. Such a system has no need for superheating which means more of the surface area is being used for evaporation.

2.7 Heat transfer

This section will include all the relevant heat transfer theory used to design a heat exchanger.

Heat transfer from a pipe to a surrounding fluid will depend on the convection coefficient of the fluid in the pipe and the conduction coefficient of the pipe material. The general equation for energy potential is shown in equation 2.1 and the energy transferred by convection is given by

$$Q = hA(T_s - T_\infty) \tag{2.3}$$

where T_∞ is the fluid temperature and T_s is the surface temperature, h (W/m^2K) is the convection coefficient and A is the area. For conduction, Fouriers's law is used and defined below where k (W/mK) is the thermal conductivity and the temperature difference is on either side of the wall [33].

$$Q = -kA\frac{dT}{dx} \tag{2.4}$$

In general, heat transfer in a system can be expressed as a function of temperature difference, the overall heat transfer coefficient, U, and the area.

$$Q = UA\Delta T = \frac{\Delta T}{R_{tot}} \tag{2.5}$$

Thermal resistance is a method to evaluate the heat transfer through a material or fluid. It is given in the unit of K/W and can be either put in series or parallel. The equations for conduction and convection resistance in a radial system is shown in equation 2.6 and 2.7 respectively. r_2 is the outer diameter of the pipe, r_1 is the inner diameter and L is the pipe length.

$$R_{th,cond} = \frac{ln(r_2/r_1)}{2\pi Lk} \tag{2.6}$$

$$R_{th,conv} = \frac{1}{h2\pi rL} \tag{2.7}$$

The heat diffusion equation is important to predict the heat transfer in a heat exchanger [13]. The idea is to see how heat evolves in a body. For cylindrical coordinates,

$$\frac{1}{\alpha}\frac{\partial T}{\partial t} = \left(\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial T}{\partial r}\right) + \frac{1}{r^2}\frac{\partial^2 T}{\partial \varphi^2} + \frac{\partial^2 T}{\partial z^2}\right) + \frac{\dot{q}}{k}$$

which can be used to determine the temperature at a given point in a pipe. Assuming incompressible medium, no convection, isotropic material, k independent of T, steady state and no heat source the equation can be simplified to,

$$\frac{1}{r}\frac{\partial}{\partial r}\left(kr\frac{\partial T}{\partial r}\right) = 0$$

solving this for T and imposing boundary conditions $T(r_1) = T_1$ and $T(r_2) = T_2$ gives,

$$T(r) = \frac{T_1 - T_2}{ln(r_1/r_2)} ln\left(\frac{r}{r_2}\right) + T_2. \tag{2.8}$$

From equation 2.8 the temperature at any radius of a pipe can be found. Figure 2.11 shows how the heat will travel in a radial system

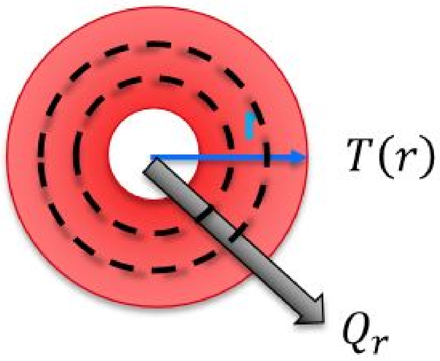


Figure 2.11: Schematic of the heat direction in radial direction [33]

Heat transfer over a radial surface is given by,

$$Q_r = \frac{2\pi Lk}{ln(r_2/r_1)}(T_1 - T_2) = \frac{T_1 - T_2}{R_{th,cond-cyl}} \tag{2.9}$$

Equation 2.9 can be rewritten to find heat transfer per unit length of tube and can be expressed by using the thermal resistance in the material.

Before determining the convection coefficient it is necessary to examine how the nature of the flow will impact the convectional heat transfer. Flow types can be categorized as laminar, turbulent, and transitional, where transitional is when the flow transitions from laminar to turbulent. Laminar flow is characterized by smooth layers or streamlines moving parallel to each other while turbulent is a distorted flow that creates eddies and vortices [41].

When a fluid is flowing past a solid body, the flow is affected by it. This creates a region that has a variable velocity that is located between the flow and the body surface which is called a boundary layer [13]. The thickness of the boundary layer, δ is defined as the distance from the wall where the velocity is within 1% of the freestream velocity and is usually very thin in comparison to the other dimensions of the flow. The boundary layer thickness divided by its characteristic length can be expressed as a function of the Reynolds number. This is the flow boundary layer.

$$\frac{\delta}{x} = f(Re_x)$$

The Reynolds number is a way to predict the flow pattern, i.e. if the flow is laminar or turbulent. Usually, flow in a pipe is laminar when the Reynolds number is below 2100 but this depends on the nature of the flow such as if it is coiled [10]. The critical Reynolds number is the point when the flow transitions and can be defined as

$$Re_{critical} = \frac{\rho D u_{avg,crit}}{\mu}.$$

$u_{avg,crit}$ is the transitional value of the average velocity. The thermal boundary layer is different from the flow boundary layer and is inversely proportional to the Nusselt number $Nu_x = \frac{x}{\delta_t'}$. The Nusselt number is a non-dimensional unit of measurement to show the ratio of convective to conductive heat transfer of a fluid and can be used to find the convection coefficient, h.

$$Nu_x = \frac{hx}{k_f} \tag{2.10}$$

For a turbulent water flow, the convective heat transfer is significantly larger than the conductive heat transfer which means a large Nusselt number. The thermal boundary layer for such a flow will, therefore, be thin. For laminar flow, the convection coefficient will be lower and this is a consequence of the boundary layer that is created because of the smooth fluid flow. The general equation for the Reynolds number is shown in equation 2.11.

$$Re_x = \frac{\rho ux}{\mu} \tag{2.11}$$

Prandtl number is the ratio between momentum diffusivity to thermal diffusivity. ν is the kinematic viscosity and α is the thermal diffusivity. Thermal diffusivity is a measure of the rate of heat from the hot end of a material to a colder end and has the unit m^2/s. The thermal boundary layer divided by the flow boundary layer can be expressed as a function of the Prandtl number and the Nusselt number can in turn be a function of the Prandtl number. This is important for determining the convection coefficient.

$$Pr = \frac{\nu}{\alpha} \tag{2.12}$$

Convection relations straight pipe

For internal cylindrical laminar forced convection relations with constant surface temperature, the Nusselt number is $Nu_D = 3,66$. The notation "D," represents the characteristic length which for a pipe is the inner diameter.

For turbulent flow with a Reynolds between 2300 and 10000 and a Prandtl number between 0,7 and 160 has the following relation. n=0,3 for cooling and n=0,4 for heating [33].

$$Nu_D = 0.023 Re_D^{4/5} Pr^n \tag{2.13}$$

If the flow has a Reynolds number of above 10000 and a Pr of between 0,5 and 3000 the Nusselt number:

$$Nu_D = \frac{(f/8)(Re_D - 1000)Pr}{1 + 12,7(f/8)^{1/2}(Pr^{2/3} - 1)} \tag{2.14}$$

Where f is the friction factor given by:

$$f = (0,790 ln Re_D - 1,64)^{-2}$$

Forced convection relations coil

When the pipe is a coil, the geometry of this coil is important for its heat transfer properties. When the pipe is curved there will be centrifugal forces in the fluid which causes a secondary flow that circulates outward into the core region of the pipe and forms a pair of symmetric vortices. Since there is both a main and secondary flow, the maximum velocity is shifted outwards from the pipe center. The secondary flow creates an additional convective heat transfer between the fluid and the wall which improves the heat transfer. These differences are particularly prevalent in laminar flows [10].

In an experiment comparing the heat transfer of a helical coil to a straight pipe, it was found that the coil has up to 1,43 times larger heat transfer coefficient. The pipes were submerged in a water bath and had water flowing through them. A few different mass flow rates and bath temperatures were tested and compared. In figure 2.12 the difference in heat transfer coefficient is plotted for different mass flow rates in l/min. Both the straight pipe and the coil have a length of 6,38m and the diameter of curvature is 0,203m. The inner diameter for the coil is 15mm while for the straight pipe it is 17mm. Both pipes have higher heat transfer for the 50°C water bath compared to a temperature of 40°C [25].

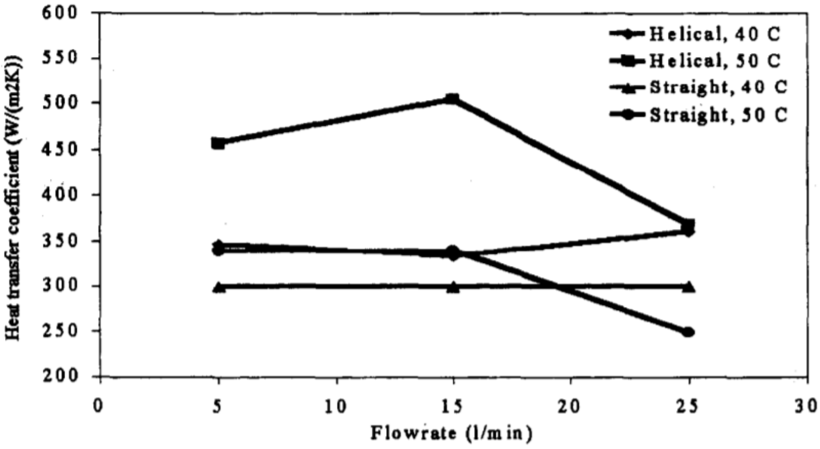


Figure 2.12: Heat transfer coefficient for coiled and straight pipe for different mass flow rates

Average diameter of a spiral with n turns and a pitch of h with a tube length l is given by:

$$D_s = \frac{l}{n\pi} \tag{2.15}$$

The diameter of a winding in the coil:

$$D_C = \sqrt{D_s^2 - \left(\frac{h}{\pi}\right)^2} \tag{2.16}$$

Average diameter of curvature D is given by:

$$D = D_c \left[1 + \left(\frac{h}{\pi D_c}\right)^2\right] \tag{2.17}$$

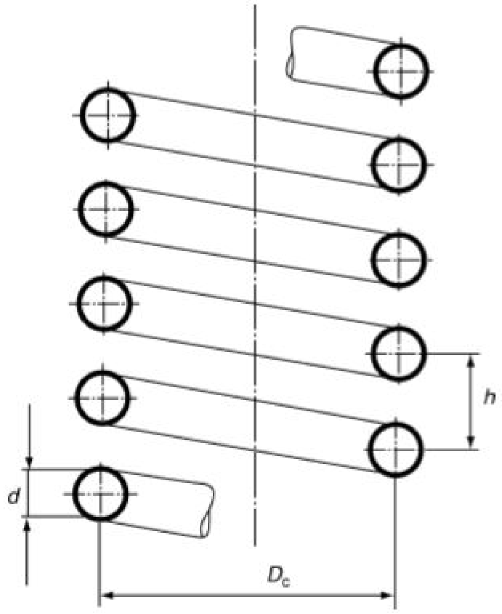


Figure 2.13: Visualization of the coil geometry dimensions [10]

The secondary flow has a stabilizing effect on the laminar flow which means that the transition from laminar to turbulent happens at a higher Reynolds number. For a coil the critical Reynolds is defined as in equation 2.18

$$Re_{crit} = 2300 \left[1 + 8,6 \left(\frac{d}{D}\right)^{0,45}\right] \tag{2.18}$$

For a laminar flow, $Re < Re_{crit}$ in a coil the Nusselt number the Schmidt equation (2.19) can be used. This equation has a deviation of $\pm 15\%$ compared to the measured values of some coils. The Prandtl number is evaluated at mean fluid temperature and the Pr_w is the Prandtl number at tube wall temperature. The factor $\left(\frac{Pr}{Pr_w}\right)^{0,14}$ is there to account for the temperature dependence of the physical properties.

$$\begin{gathered} Nu = 3,66 + 0,08 \left[1 + 0.8 \left(\frac{d}{D}\right)^{0,9}\right] Re^m Pr^{1/3} \left(\frac{Pr}{Pr_w}\right)^{0,14} \\ m = 0,5 + 0,2903(d/D)^{0,194} \end{gathered} \tag{2.19}$$

There is a transitional region for heat transfer when the Reynolds number is above critical but below fully developed flow at $Re = 2,2 \cdot 10^4$. Measurements for a turbulent/fully developed flow were done with air and water and equation 2.20 has a deviation of $\pm 15\%$ when compared. The friction factor, f, has a correctional factor $\left(\frac{\mu_w}{\mu}\right)^{0,27}$ where μ_w is the dynamic viscosity at wall temperature and μ is the dynamic viscosity at mean fluid temperature.

$$
\begin{aligned}
Nu &= \frac{(f/8)RePr}{1+12,7\sqrt{f/8}(Pr^{2/3}-1)}\left(\frac{Pr}{Pr_w}\right)^{0,14} \\
f &= \left[\frac{0,3164}{Re^{0,25}}+0,03\left(\frac{d}{D}\right)^{0,5}\right]\left(\frac{\mu_w}{\mu}\right)^{0,27}
\end{aligned}
\tag{2.20}
$$

When the flow is in the transit region of heat transfer, $Re_{crit} < Re < 2,2 \cdot 10^4$. $Nu_l(Re_{crit}$ is the Nusselt number for laminar flow, equation 2.19, if $Re = Re_{crit}$. $Nu_t(Re = 2,2 \cdot 10^4)$ is Nusselt number at turbulent flow if $Re = 2,2 \cdot 10^4$ in equation 2.21

$$
\begin{aligned}
Nu &= \gamma Nu_l(Re_{crit)} + (1-\gamma)Nu_t(Re = 2,2\cdot 10^4) \\
\gamma &= \frac{2,2\cdot 10^4 - Re}{2,2\cdot 10^4 - Re_{crit}}
\end{aligned}
\tag{2.21}
$$

Boiling

When the temperature on the surface of a body exceeds the saturation temperature at a given pressure, boiling will occur. There are two boiling processes that are possible: Pool boiling is when the liquid is at rest and free convection occurs, while forced boiling is when the liquid is moving [33].

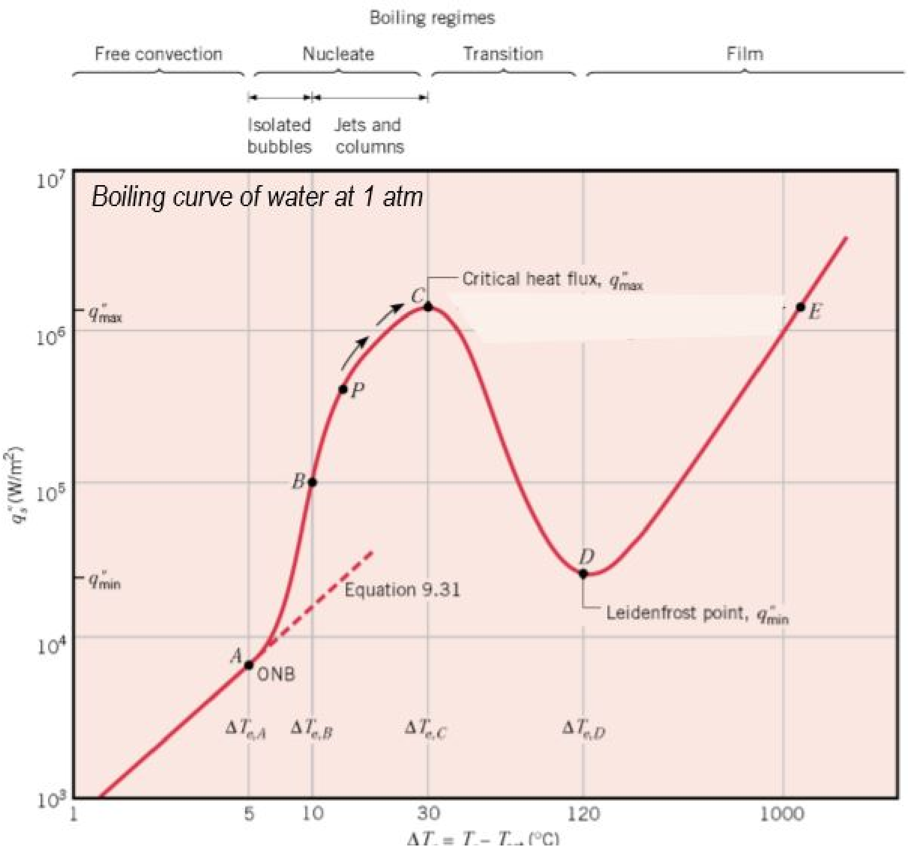


Figure 2.14: Boiling curve for water at atmospheric pressure[33]

figure 2.14 shows water at atmospheric pressure and how it boils. Point A shows the start of the boiling, the region from point A to B is the region of isolated bubbles. Bubbles rise from some isolated nucleation sites and as the heat flux and temperature increase, more sites are activated [13]. Point B to C is the region of slugs and columns. When a certain number of nucleation sites have been activated, the bubbles merge and form a different escape path for the vapor. Jets form that makes overhead bubbles or "slugs" of vapor. From point B to P, the convection coefficient and the excess temperature increase, where P is the point of maximum convection. The temperature continues to rise from P to C but now the convection coefficient decrease. Point C marks the critical heat flux, if heat flux goes beyond this point the temperature increase rapidly and can potentially damage the system. The boiling crisis is from point C to E directly and is caused by a vapor film that replaces the bubbles. Since vapor has lower thermal conductivity, the surface temperature increase rapidly. A heat exchanger must, therefore, be designed to not reach beyond critical heat flux. Point C to D is where the vapor layer forms and the heat flux will decrease until it reaches its minimum and starts to increase again.

The heat flux given in W/m^2 from the surface to the fluid is,

$$q_s^{''} = h(T_s - T_{sat}) = h\Delta T_e \tag{2.22}$$

where T_s is the surface temperature and T_{sat} is the saturation temperature of the fluid. For nucleate boiling the heat flux can also be estimated using the Rohsenow correlation.

$$q_s^{''} = \mu_l h_{fg} \left[\frac{g(\rho_l - \rho_v)}{\sigma}\right]^{1/2} \left(\frac{C_{p,l}\Delta T_e}{C_{s,f} h_{fg} Pr_l^n}\right)^3 \tag{2.23}$$

h_{fg} is the latent heat of vaporization in kJ/kg, μ_l is the dynamic viscosity of the liquid in Ns/m^2, ρ_l and ρ_v is the density of liquid and vapor respectively. σ is the surface stress tensor in N/m, $C_{p,l}$ is the specific heat of liquid, $C_{s,f}$ is the Rohsenow coefficient is constant and an empirical correction for surface-fluid condition. The exponent n is also an found empirically for common surface-fluid combinations.

The peak heat flux can be found with the following equation,

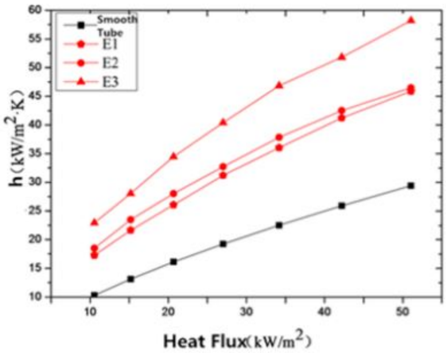
$$q_{s,max}^{''} = C h_{fg} \rho_v \left[\frac{h\sigma(\rho_l - \rho_v)}{\rho_v^2}\right]^{1/4}$$

The constant C is $\frac{\pi}{24}$ for large horizontal cylinders, spheres and finite heated surfaces.

Experimental research has been conducted and showed that experimental results for CO_2 pool boiling outside a single tube compared to theoretical results have a maximum error of 8,73%. The boiling pressure was in the range of $2 - 4MPa$ and the heat flux studied ranged from $10 - 50kW/m^2$. The temperature difference from the pipe wall to saturated CO_2 was $1,27K$.

Number of tubes	E0	E1	E 2	E3
Inner diameter d_i	19	19	19	22.86
Outer diameter d_0	22	22	22	25.24
Screwed pitch		0.8	1	
Slot width of screwed pitch		0.8	1	
Height of screwed pitch		1	1	
Root diameter of fin				24.3
Minimum wall size under fin				0.635
Full fin diameter				25.5

(a) Pipe configurations



(b) Heat flux plotted against convection coefficient

Figure 2.15: Experimental CO_2 pool boiling results outside a single tube

E1-3 indicates screwed tubes with different configurations. The results show that the screwed tubes gave a better convection coefficient compared to the smooth tube. E3 gave the best results because of the fins. Finns increase the heat transfer area and the screw configuration enhances the heat transfer mechanism since it disturbs the boundary layer and promotes the breakaway of bubbles while maintaining the nucleate points. All the results in the graph are given for an evaporating pressure of 3,2 MPa. To calculate the convection coefficient, the heat flux was divided by the temperature difference of the wall and saturated CO2, $h = \frac{q"}{\Delta T}$.

2.8 Pressure drop

A fluid that is flowing through a pipe will experience a pressure drop due to the resistance to the flow [24]. There are several contributions to the pressure drop along the pipe:

- Friction between the wall and the fluid
- Friction between fluid layers
- Friction due to the flow passing through bends or components
- Pressure loss due to elevation change
- Pressure gain added by a pump

Flows used for heating or cooling in pipes are often driven by a fan or a pump. The pressure drop can be used to determine the power needed from the pump or fan [41].

The general equation for the pressure drop is:

$$\Delta P = \zeta a \frac{\rho u^2}{2}$$

where a is the factor that depends on the nature of the flow and ζ is the Darcy friction factor which will be called drag coefficient in this paper. In a pipe flow, a, will depend on the length and diameter of the pipe. The density can change along the pipe and if this is the case, the calculations should be done over several steps. The pressure drop in a pipe with circular cross section is given by:

$$\Delta P = \zeta \frac{l}{d_i} \frac{\rho u_i^2}{2} \tag{2.24}$$

The drag coefficient depends on the Reynolds number and for a straight pipe this coefficient will be different depending on laminar of turbulent flow. For pipes with little roughness, $k = 0,07$ follows the Hagen-Poiseuille law which can be seen in equation 2.25.

$$\Delta P = \frac{32\mu ul}{d_i^2} \tag{2.25}$$

The general equation for the drag coefficient is

$$\zeta = \frac{8\tau}{\rho u^2} \tag{2.26}$$

where τ is the viscous stress tensor and for laminar flow it can be expressed in relation to the Reynolds number

$$\zeta = \frac{64}{Re}$$

For turbulent flow the drag coefficient can be expressed as

$$\zeta = \frac{0,3164}{\sqrt[4]{Re}},$$

which is for Reynolds numbers of 3000 to 100000. With an increased Reynolds of $2 \cdot 10^4$ to $2 \cdot 10^6$ the Hermann equation can be used.

$$\zeta = 0,00540 + \frac{0,3964}{Re^{0,3}} \tag{2.27}$$

For Reynolds numbers above this, the Prandtl and von Kármán equation must be used.

$$\frac{1}{\sqrt{\zeta}} = -0,8 + 2\log(Re\sqrt{\zeta} \tag{2.28}$$

Pressure drop in coiled pipes

If the pipe is coiled the pressure drop will be greater due to the secondary current. and the drag coefficient will depend on the $\frac{d}{D}$ ratio. The drag coefficient for a coil, ζ_c, for laminar flow or when $1 < (Re\sqrt{d/D}) < Re_{crit}\sqrt{d/D}$, can be expressed as

$$\zeta_c = \frac{64}{Re}\left[1 + 0,033(log_{10}\{Re\sqrt{d/D}\})^4\right] \tag{2.29}$$

For turbulent flow or when the Reynolds range is $Re_{crit} < Re < 10^5$ can be expressed as

$$\zeta_c = \frac{0,3164}{Re^{0,25}}[1 + 0,095(d/D)^{1/2}]Re^{1/4} \tag{2.30}$$

The pressure drop is propotrional to the pressure drop in the pipe and given by

$$W_{pump} = \dot{V}\Delta P. \tag{2.31}$$

$\dot{V}$ is the volumetric flow rate in m^3/s and the pressure drop, ΔP, is in kPa. There is $1kW$ for every $kPa \cdot m^3/s$ and the volumetric flow rate can be found by multiplying the inner pipe cross section with the fluid velocity $\dot{V} = uA_{c,i}$ [41]

2.9 Heat pumps

The technology is based on a reversed Carnot cycle, using an energy input to produce heat [27]. Heat pumps can be used both for heating and cooling. Several different heat sources are common for heat pumps:

- Outdoor air or gases from different processes
- Water from rivers or ground
- Soil or ground source

The choice of heat pump depends on which source heat is taken from and where it is delivered. There are three types of heat pumps:

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

A heat pump consists of four main components. An evaporator, compressor, condenser, and expansion valve [1]. The evaporator and condenser are heat exchangers and all the components are connected to form a closed circuit. Figure 2.16 show a simple schematic of the cycle. In the evaporator, heat is taken from the surroundings to vaporize the working fluid. The second stage is to enter the compressor to increase the pressure and temperature of the vapor. When it enters the condenser the heat is expelled to the surroundings and the vapor condenses before it is expanded in the expansion valve.

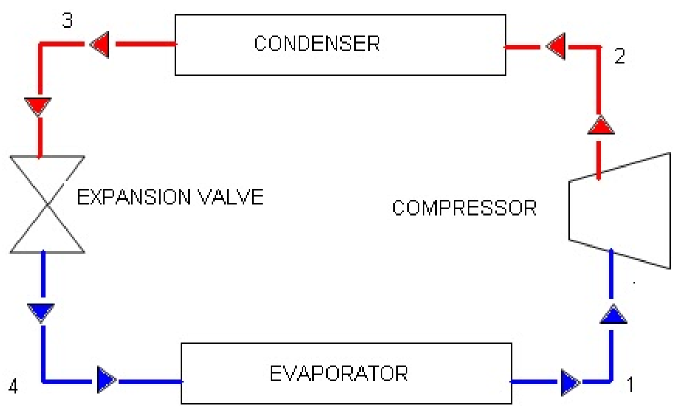


Figure 2.16: Simple schema of a heat pump cycle [28]

CO_2 heat pump

CO_2 is a non-toxic and non-flammable working fluid that does not contribute to the depletion of ozone if leakage were to occur. With a global warming potential (GWP) of 1, it has a net impact of zero when used in heat pumps. This working fluid is a good alternative to the conventional (H)CFC's since they are now being phased out due to their impact on the climate. It has good heat transfer characteristics and efficient compression. An advantage of CO_2 compared to other working fluids is that it can be used for the production of hot tap water with a temperature of up to 90°C without operational difficulties [18].

Transcritical process

The transcritical process is characterized by its ability to operate around the critical temperature. Heat rejection happens at constant pressure and a falling temperature and when transcritical, the heat exchanger is called a gas cooler [31]. The difference between the conventional heat pump cycle and the transcritical is the temperature drop. In a conventional cycle, the refrigerant is condensing and the temperature remains constant. Figure 2.17, which illustrates the cycle, shows that vapor will be compressed to a high temperature and pressure from point 1-2. This compression is isentropic and point 2' therefore shows the real compression when compressor efficiency is accounted for [35]. The gas exits the compressor and flows into the condenser and rejects heat. This process is isobaric, meaning constant pressure, which is in the supercritical region. Proceeding further into the expansion valve which decreases the temperature and pressure. Point 3-4' shows an isenthalpic throttling process which is traditionally used in a CO_2 system. 3-4" is an isentropic expansion process and 3-4 is a process using an expander instead of an expansion valve. The fluid is then two-phase and enters the evaporator and absorbs heat.

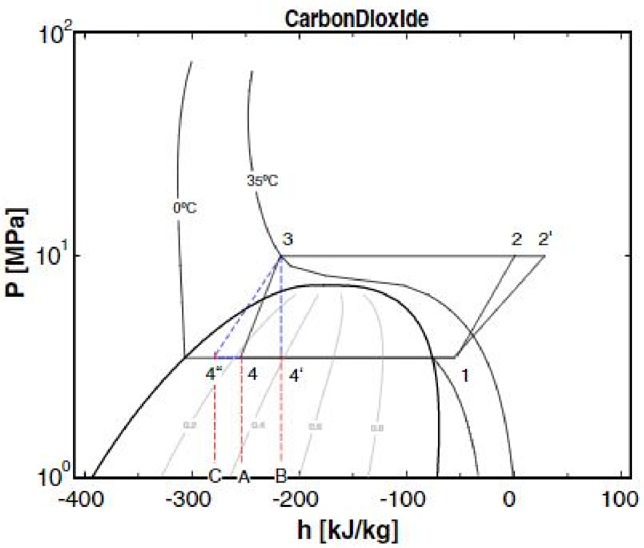


Figure 2.17: Pressure-enthalpy diagram for a CO_2 trans-critical cycle [35]

Figure 2.18 shows the cooling curve of CO_2, 2-3, and the heating curve for water, a-b. The water in the figure has constant c_p and is therefore linear. Having counter-current flow in the heat exchanger will minimize the temperature approach of water and CO_2.

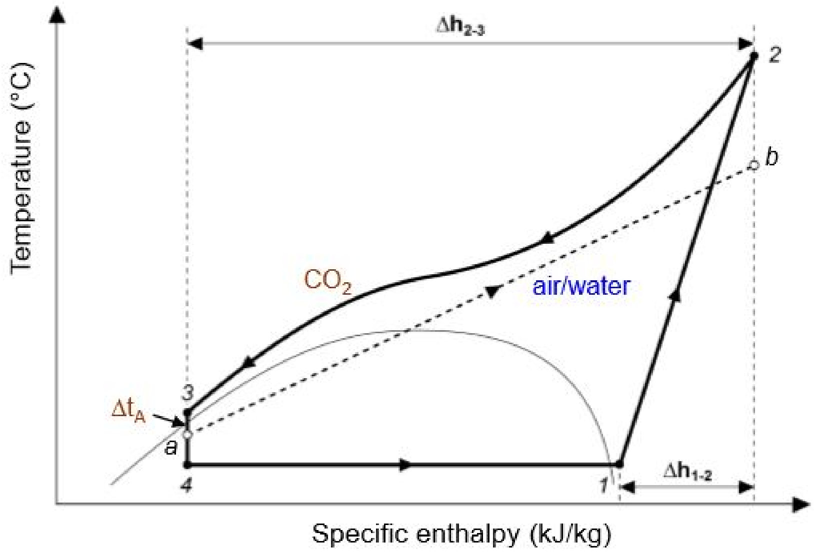


Figure 2.18: The temperature-enthalpy diagram showing the cooling of CO_2 and heating of air/water [31]

The following equations are used for calculations in a CO_2 trans-critical process.

$$W_r = (h_3 - h_4)\dot{m}$$

W_r is the recovered power and h_3 and h_4 is the enthalpy in point 3 and 4 respectively

$$Q_k = (h_1 - h_4)\dot{m}$$

Q_k is the cooling capacity.

$$W_{in} = (h_2 - h_1)\dot{m}$$

W_{in} is the compressor input power.

$$COP = \frac{Q_k}{W_{in}} = \frac{h_1 - h_4}{h_2 - h_1}$$

$$COP_h = \frac{Q_0}{W_{in}} = \frac{h_2 - h_3}{h_2 - h_1} \tag{2.32}$$

Q_0 is heating capacity and COP_h is the COP for heating

Superheat

Superheating is the temperature difference between the saturation temperature of the evaporator outlet and the actual evaporator temperature [9]. It works as safety for evaporating all the refrigerant to avoid damage to the compressor, although it is better for the efficiency to minimize the superheat. To achieve superheat, an internal heat exchanger (IHX) can be used [36]. The IHX reduces the vapor fraction entering the evaporator which increases the efficiency, but at the same time, it increases the suction temperature for the compressor which will reduce the mass flow rate and increase specific compressor work.

Chapter 3

Method

3.1 System description

Figure 3.1: Map of the different sections that will be built at Leangen

Leangen will be built in different building stages and with several GW collection tanks. Below will be an explanation of the different abbriviations.

- B: Living areas
- KB: Office buildings
- o_BOP: Assisted living facility

Table 3.1: Overview over the area

Field	Living quarters [m^2]	Service [m^2]	Office [m^2]	Kindergarden [m^2]
B1	10 600			
B2	13 200			
B3	11 000			
B4	12 700	100		
B5	16 000	100		
B6	29 400	200		
B7	13 800			
B8	9000			
B9	22 700			
BKB1	6600	1900		2000
BKB2	9000	2100		
KB			9200	

The focus for this thesis will be field B1 since it is solely living quarters. In figure 3.2, a proposed system is shown where a heat pump with a natural working fluid is supplying heat to the buildings. This heat pump will have two evaporators where one will be used during winter and one during summer. In the figure, dotted lines represent where the water will go in the summertime. When the heating evaporator is operating, it will be supplied by the LT-supply with a temperature of about $28°C$ and exiting with $23°C$. A heat pump will be used to heat a water loop supplying hot water to the building complex which will return to the condenser for reheating. In the warmer months when there is less need for space heating the cooling evaporator will be operating. There will be a closed loop of water being heated in the building that will be used as a source for this evaporator. The heat pump is still operating, but the heat will be used to heat the GW in the tank. A CO_2 heat pump is using the GW as a heat source to heat city water to a temperature of approximately 75-85°C to be used for domestic hot water. Dual evaporators will be used for where one is a flooded evaporator using high-velocity GW as its source. An alternative is to have a second pipe in the flooded evaporator. The other evaporator or pipe will use the outlet of the heating evaporator for the space heating heat pump and will function as a backup if the GW is not sufficient. The hot water will go to DHW storage tanks for later use. A heating device will be placed in the last DHW tank to heat the water to the requirements when needed. A reheater is also there to reheat circulating water if the temperature is too low.

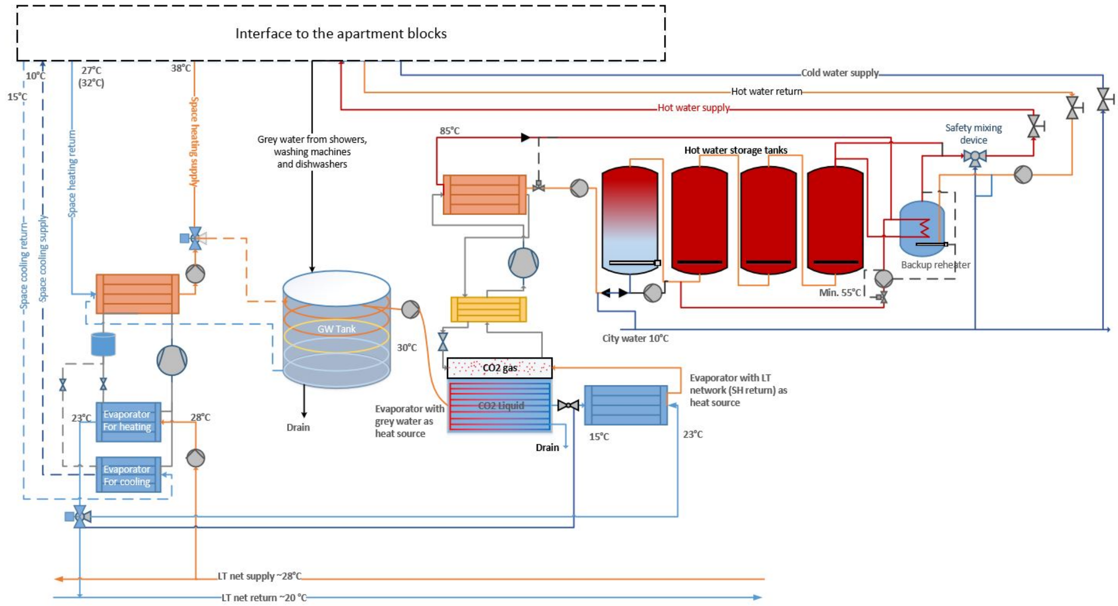


Figure 3.2: Schematic of the energy system supplying heat for the different fields

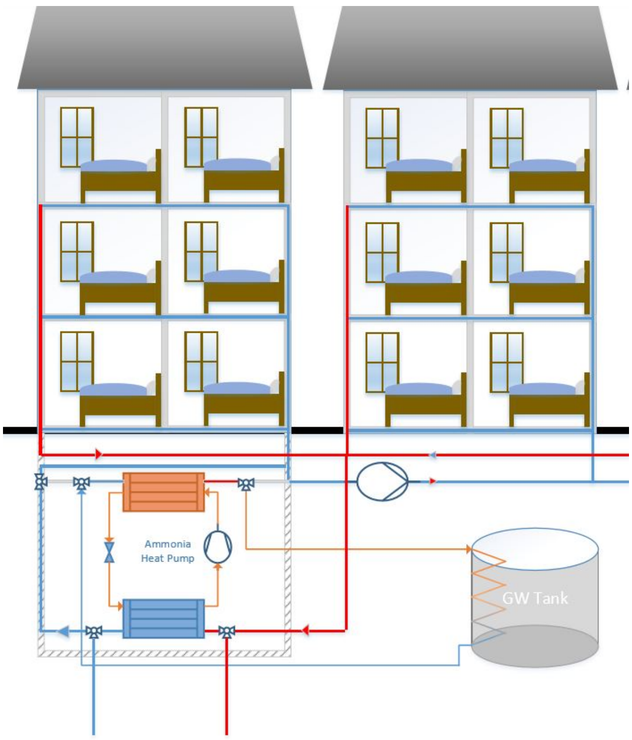


Figure 3.3: Schematic of the field with summer mode activated

In figure 3.3 the distribution system inside the buildings is shown. This schematic shows how it looks during the summer when the heat is used in the GW. The cooled water from the evaporator will now be used in the building piping. During winter the valve to the GW tank is closed and the water from the condenser will circulate in the building. The cooled water from the evaporator will now be sent in the return line to be reheated at the ice skating rink. The figure only shows one of the evaporators for simplicity.

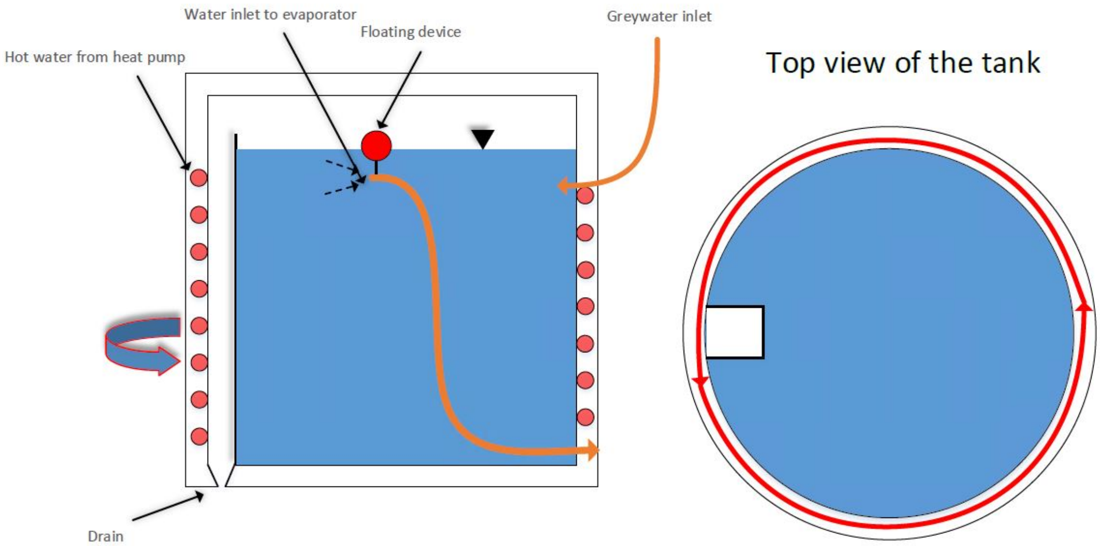


Figure 3.4: Schematic of the GW tank from the side and above

Figure 3.4 shows the cross section and top view of the proposed GW tank. Hot water from the space heating heat pump will circulate in the tank wall to increase the GW temperature during summer. A hose with a floating device connected to it will be placed in the tank to account for the fluctuation in water level. If the water level reaches a maximum level, the excess water will be drained.

3.2 Energy demand and potential

A simulation of the energy demand was done by Hanne Kauko for all the different areas. This section shows the results from this simulation which can further be used in the analysis of the energy system. The focus is on the living quarters and the office and commercial buildings are not included in the figures. Figure 3.5 shows the heating demand of the B1 field which is the main focus of this thesis. DHW demand is consistent throughout the year, while space varies significantly and even reaches no demand during the summer.

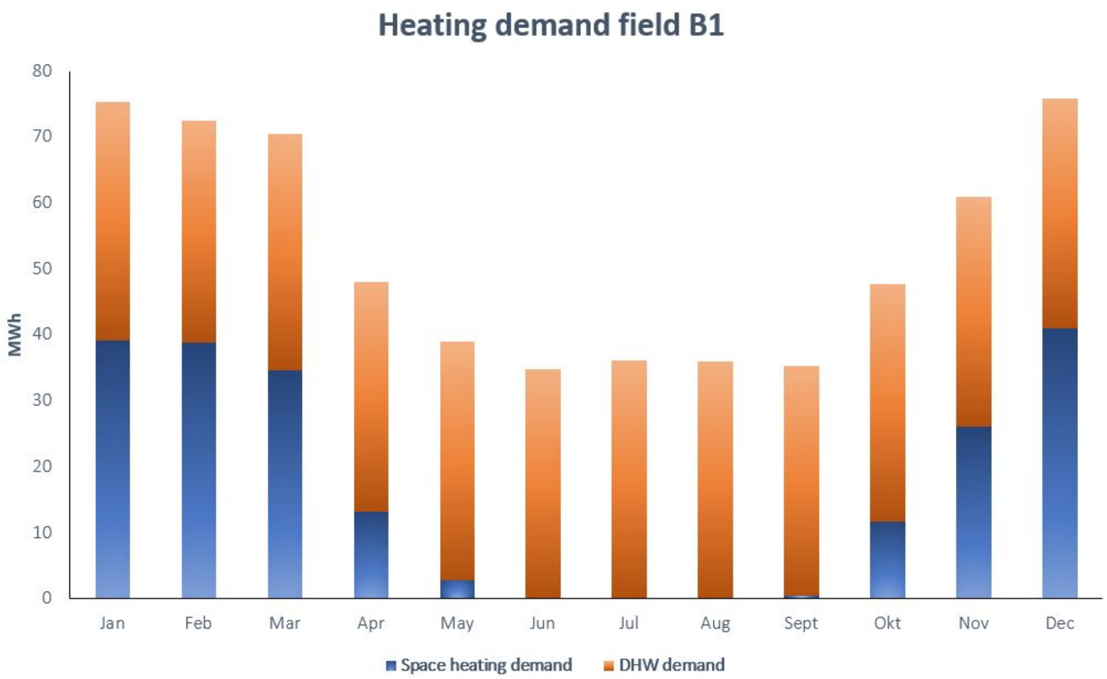


Figure 3.5: The heating demand in MWh for field B1

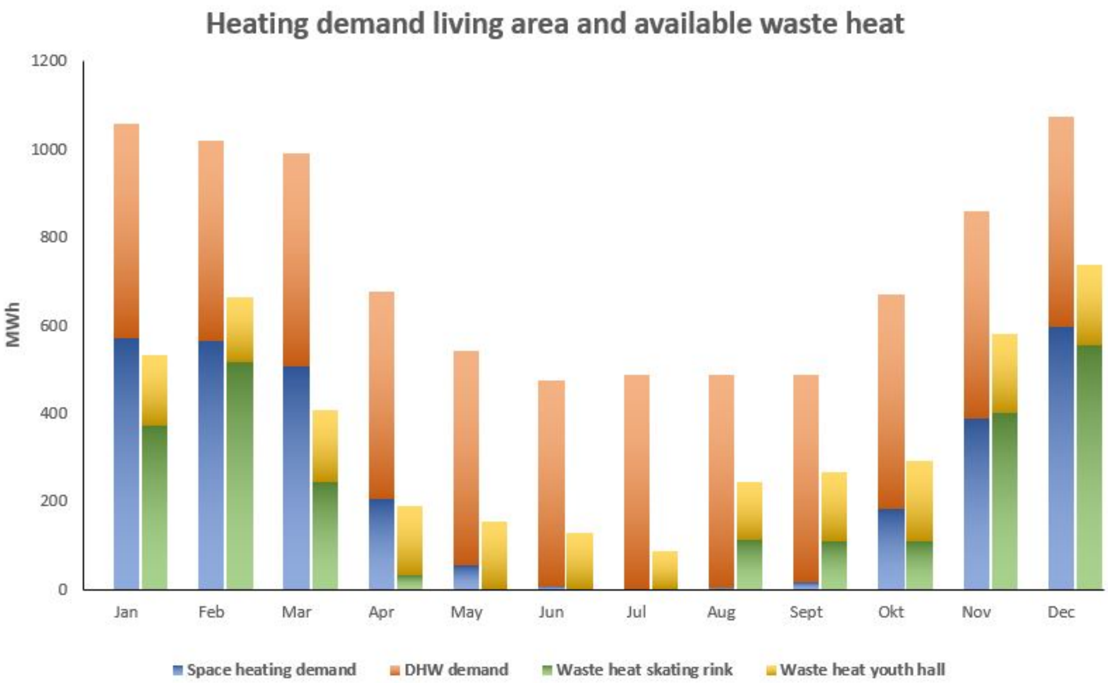
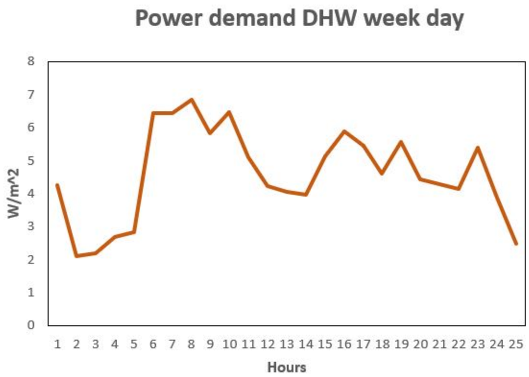
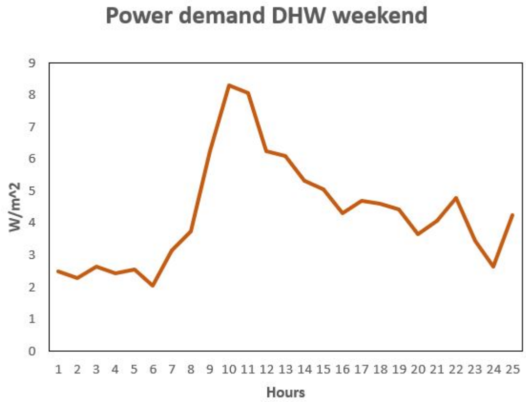


Figure 3.6: Heating demand of the entire living area compared with the available waste heat

The waste heat available is not enough to provide energy for all the apartments, figure 3.6. A majority of the heat available is provided by the ice skating rink, while some can be provided by the youth hall. Heat pumps can, therefore, be helpful to increase the quality of the heat which can reduce the discrepancy.



(a) DHW demand for a week day



(b) DHW power demand for a weekend

Figure 3.7: Comparison of the hourly DHW power demand during a weekday and a weekend

In figure 3.7 the hourly value for the power demand for DHW is shown. During weekends there is a higher peak demand that may be due to people being home during that time. For a weekday, the demand is more and has a high demand during the morning hours and in the evening. The demand is relatively constant throughout the year and does not vary much from winter to summer. In figure 3.8 the average daily power demand for DHW is shown. The maximum value is 1165 kWh for some moths. If the heat pump is operating 20 hours per day there should be a supply of 58,25 kW for all the 20 hours.

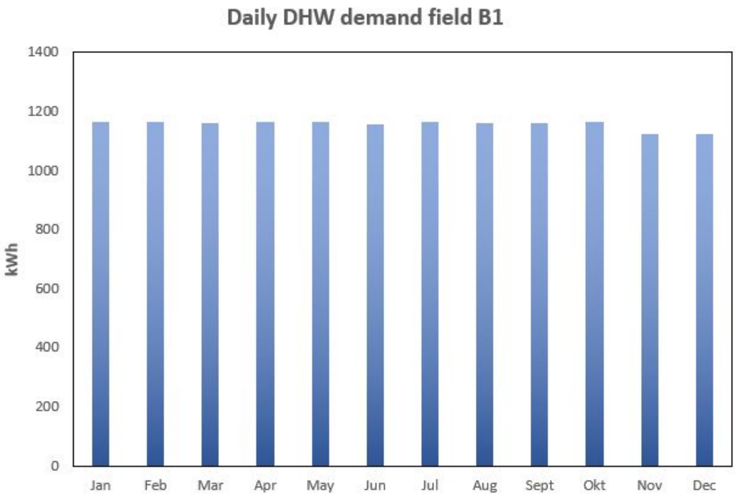
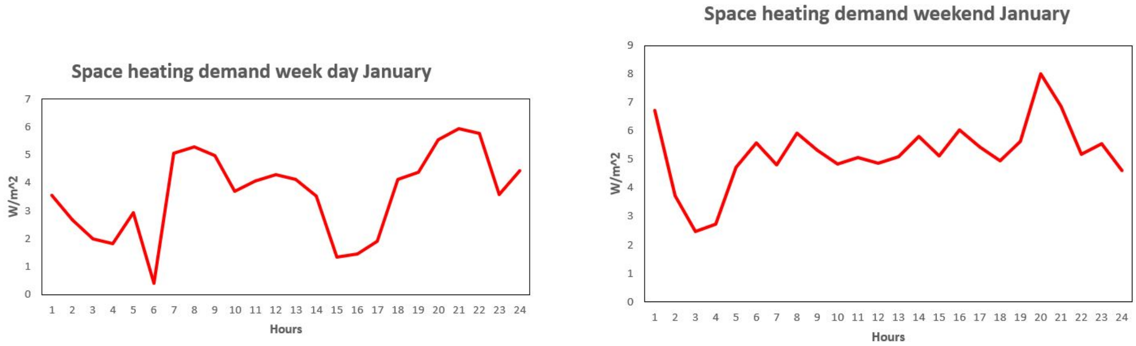


Figure 3.8: Average daily DHW demand for field B1



(a) Space heating demand for a weekday (b) Space heating power demand for a weekend

Figure 3.9: Comparison of the hourly space heating power demand during a weekday and a weekend

The power demand for space heating varies a lot from winter to summer as can be seen in figure 3.6. Therefore, figure 3.9 shows the hourly demand during a weekend and weekday in January. The power demand is on average higher during the weekends but stays more constant, while during the weekdays it varies more.

Greywater potential

From the project thesis last year, the energy potential of the GW was calculated and the results will be summarized here. The amount of GW produced per apartment is $162,75$ liters per day with an average temperature of $34,89°C$. Every apartment is set to be $70m^2$ which means that the GW production is $2,325l/m^2$ per day. For the field B1 which is $10600m^2$ the total production per day is $24645kg/day$, assuming that 1 liter of water is 1 kg.

3.3 Flooded evaporator

To find the heat transfer of a coil in the flooded evaporator with high velocity, a model was developed in Python. The pipe dimensions had to be chosen. To avoid a too high mass flow rate a pipe diameter of 15mm was chosen and according to Lister Tube the standard wall thickness for such a pipe is 1,5mm [14]. The set properties are listed in table 3.2

An assumption is that the temperature of the inner pipe wall has about the same temperature as the water flow. This is because of a negligible boundary layer due to the high Reynolds number due to the high velocity.

Table 3.2: Flooded evaporator dimensions and assumptions

Pipe dimensions	
Inner diameter [d]	0,0012 m
Outer Diameter [d_o]	0,0015m
Distance between coils [h]	0.03 m
Number of turns [n]	25
Various properties	
Cp	4,19 kJ/kgK
Temperature at inlet [T_{in}]	30°C
Density [ρ]	997 kg/m^3
Conduction coefficient stainless steel (AISI 304) [k]	14,9 W/mK
Temperature of CO_2 [T_{CO_2}]	273 K
Inlet water temperature [T_i]	303,15K
Minimum water outlet temperature [T_o]	273.15
Water velocity [V]	3-6 m/s

The velocity for GW has to be high to avoid fouling in the heat exchanger pipe and a range from $3m/s$ to $6m/s$ with an interval of $0,2$ was tested. The required length of the pipe is not known and therefore the calculations are done for different sections with a temperature drop of $1K$ per section until it reaches equilibrium with the external CO_2. The mass flow rate was calculated using the velocity with the following formula:

$$\dot{m} = V\pi r^2 \rho \tag{3.1}$$

The power for all mass flow rates with a temperature drop of 1K was then calculated for using equation 2.1 from the literature review. Viscosity, Prandtl number, and conductivity of the fluid vary for different temperatures and the values were found in a table of thermophysical properties [13]. Reynolds number was then calculated with the following formula:

$$Re_d = \frac{4\dot{m}}{\pi d \mu}$$

Which is a modified version of equation 2.11. Before using relations for heat transfer in a coil and implementing boiling relations, straight pipe relations were used.

To find the heat transfer in the straight pipe it was necessary to calculate the Nusselt number. The relations for a straight pipe can be seen in equations 2.13 and 2.14 before using the general equation 2.10 and solve for the convection coefficient. The power was then calculated by dividing the energy potential in the section with a $1K$ temperature drop by the radial heat transfer per unit length of pipe derived from equation 2.9.

When the heat transfer was found in the straight pipe the critical Reynolds number for the longest pipe was calculated using equations 2.15, 2.16, 2.17 and 2.18. To avoid a large evaporator, the diameter winding of the coil was chosen to be 0,3 m. Coil heat transfer relations were then used to find the convection coefficient for the coil, equations 2.19, 2.20 and 2.21. For the coil, the convection coefficient for external CO_2 was accounted for. First, the temperature of the outer pipe surface was calculated using equation 2.9, solving for the T_2. Choosing the Rohsenow coefficient was based on a paper by Tao Ding et al.

[7]. For CO_2 a $C_{s,f}$ of $0,0049$ and $n = 1,7$ was used. This was for n-pentane-Lapped cooper. Copper is not a material used for this heat exchanger and the results were not realistic with this value. The value for n-pentane-polished copper was used instead which also matches the coefficient used for water and stainless steel. $C_{s,f} = 0.0154$ and $n = 1,7$ was used for this heat exchanger [2] [11].

The properties of liquid CO_2 at $35bar$ are shown in table 3.3

Table 3.3: Thermal properties of CO_2 at 35 bar, 0°C,[8], [23], [37], [40], [34]

h_{fg}	$234,5kJ/kg$
μ_l	$105,4 \cdot 10^{-6}$ pas
ρ_l	$926,78kg/m^3$
ρ_v	$98,145kg/m^3$
σ	$5,57 \cdot 10^{-3}N/m$
$C_{p,l}$	$2,47kJ/kgK$
$C_{s,f}$	$0,0154$
n	$1,7$
Pr	$2,38$

The properties for CO_2 was evaluated based on several sources listed in the table description. Heat flux for boiling was then calculated using the Rohsenow correlation, equation 2.23, where the excess temperature is the temperature difference from the surface of the pipe to the CO_2. The convection coefficient of the boiling CO_2 was found using equation 2.22 and used to find the thermal resistance. Since all the thermal resistances was found, but coil length was still unknown, an iterative approach was used to find the new length. The length for a straight pipe was used as an initial guess for the length and used to calculate the convection coefficients over the pipe. The new length was subtracted by the old length and then run again until the difference was below 0,1.

The pressure drop over the pipe was estimated using the total coil length and caluclated drag coefficient. Equations 2.24, 2.29 and 2.30 was used.

The pipe was then divided into several pipes to see which solution was better. Velocity is still a criterion, and therefore the mass flow rate through all the pipes combined was chosen to be the same as for a single pipe and the diameter was reduced. The method used was the same as explained above, but the pipes were smaller.

A backup pipe can also be placed in the same evaporator to supply the additional heat needed to supply the heat pump if needed. This pipe will have the same dimensions as the pipe with GW for simplicity. The same calculation method was used, but since the velocity is not a criterion it was calculated with velocities from 0,2-3,2m/s. It is unknown what the mass flow rate will be here. The inlet temperature will be 23$°C$ as can be seen in figure 3.2.

Overall heat transfer coefficient

The overall heat transfer coefficient was found using equation 2.5 and then solve for U. To assure realistic values the coefficient was calculated for all the sections cumulative. Every section was added together and the average temperature difference from the water to the CO_2 was used as the pipe got longer. For example, for the whole pipe, the ΔT value is $15°C$ because the inlet is $30°C$ and the outlet is $0°C$ and the CO_2 is kept at $0°C$.

3.4 Heat pump system in dymola

Evaporator

To estimate the size of the evaporator, the system was simulated in Coolpack with a gas cooler size of 60 kW based on how water amount needed. The input and output from Coolpack is listed in table 3.4.

Table 3.4: Input and results from initial Coolpack calculations

Input	
Evaporation temperature	$0°C$
Superheat	0,1 K
Gas cooler capacity	60 kW
High pressure side	90 bar
Compressor isentropic efficiency	0,75
Discharge temperature compressor	100 $°C$
Gas cooler outlet temperature	20 $°C$
Output	
Evaporator capacity	41,821 kW
Mass flow rate	0,2243 kg/s
Compressor work	11,381 kW

This is an initial calculation to begin the modeling in Dymola and the results should be considered as such. When sizing the compressor, Bitzer software was used [3]. A semi-hermetic compressor is considered and the cooling capacity is set to 41 kW. The software is considering cooling machines and the results are only an initial guess to be used in Dymola. Displacement size with the correct input is 12 m^3/h for 50 hz. The input in Dymola is in m^3 and so dividing the by the speed and seconds in an hour gives a displacement of $\frac{12m^3/h}{3600s/h \cdot 50hz} = 6,66 \cdot 10^{-5} m^3$. The valve has an effective flow area (A_{eff}) which represents the smallest constriction in the valve in Dymola. The mass flow rate of refrigerant can be calculated using the following equation

$$\dot{m} = A_{eff}\sqrt{(P_{high} - P_{low}) * 2\rho_{high}}$$

ρ_{high} is the density of the gas at the valve inlet, or at high pressure which in this case is 90 bar. Using the tabulated value for density at high pressure and in the equation gives a value of $2,33 \cdot 10^{-6} m^2$ [37].

The evaporator dimensions will be taken from the calculations that will be presented later in the thesis and the gascooler will be designed to achieve a temperature lift from $5°C$ to $75°C$

To control the system, two controllers will be placed in the system. One will control the valve effective area and does this to ensure 90 bar at the high-pressure side, while the other controls the compressor speed and reacts to changes in suction pressure and will keep it at 35 bar or $0°C$ evaporation temperature.

The following configuration was found for the evaporator:

Table 3.5: Evaporator configuration

-	Size	Unit
Geometry outside		
Bottom clearance	0,1	[m]
Pipe length	25	[m]
Inner diameter	12	[mm]
Wall thickness	1,5	[mm]
Heat transfer		
HT coefficient single phase CO_2	0	[W/m^{2}K]
HT coefficient evaporation CO_2	14573	[W/m^{2}K]
HT coefficient water	18263	[W/m^{2}K]
Material	Stainless steel	[-]
Mass flow rate		
Water	0,451	[kg/s]

Gascooler

The gascooler capacity is 60 kW and is designed to achieve a temperature lift of 5-75°C. The heat transfer coefficients are chosen to be slighly lower than for the evaporator since the CO_2 is in vapor form and calculations has not been done for these values. In table 3.6, the configuration of the gascooler is shown.

Table 3.6: Gas cooler configuration

-	Size	Unit
Geometry outside		
Number plates	40	[-]
Length of plate	0,6	[m]
Width of plate	0,3	[m]
Flow type	Cross flow	[-]
Geometry inside		
Wall thickness	0,75e-3	[m]
Pattern amplitude	2e-3	[m]
Pattern wave length	12,6e-3	[m]
Heat transfer		
HT coefficient CO_2	10000	[W/m^{2}K]
HT coefficient water	14000	[W/m^{2}K]
Material	Copper	[-]
Mass flow rate		
Water	0,20501	[kg/s]

3.5 Cost analysis

The preliminary cost analysis will focus on the saving potential using the CO_2 heat pump instead of district heating. The electricity price will be based on an average spot price from Nord Pool Group for Trondheim [20]. This price will be used in combination with an electricity deal by Norges Energi where the price for electricity is 2,2 $øre/kWh$ plus the spot price. A monthly fee of 29 kr will also be included. The price for district heating is taken from Statkraft Varme [30]. The company has to pay for the power amount used which is 45 $kr/kWh/month$ in the summer and 60 $kr/kWh/month$ during winter and a fee of 26,08 $øre/kWh$. The cost of installing the heat pumps is hard to estimate and therefore the payback time will not be considered.

3.6 Software

Dymola with TIL-library
Dynamic Modeling Laboratory, or Dymola is a tool for simulating and modeling different systems including thermal systems [6]. It is based on the language Modelica which is an equation-based language specifically made to model physical systems [17]. It can add TIL-library made by TLK-Thermo GmbH. The library has heat pump components such as compressors, evaporators, condensers and expansion valves.

DaVe
DaVe is a visualization software that can be used to analyze the heat pump system with p-h and T-s diagrams and can use Dymola result values directly. It is also used to make graphs showing different values over time in the simulation.

To simulate the refrigeration cycle, Coolpack will be used. It can be used to calculate COP at different evaporation temperatures and with different working fluids. Drawings will be made primarily in Visio which has pre-made process components to easily make system illustrations.

Chapter 4

Results

4.1 Flooded evaporator

The results for the flooded evaporator will be divided into results from a single coil, results where it is divided into several pipes and results for the second coil. With a total production of 24645 kg/day assuming the heat pump will run for 20 h/day the mass flow rate can be 1232,25 kg/h or 0,34229 kg/s. The inner diameter is 12 mm, and using equation 3.1 and solving for the velocity show that for the GW supply to be sufficient the water can have a velocity of up to 3,035 m/s. Before examining the results from the flooded evaporator it is necessary to look at the boiling convection coefficient and how it relates to the heat flux from the pipe shown in figure 4.1. The convection coefficient increases with the heat flux increase however, it is lower than the heat flux by a factor of 10.

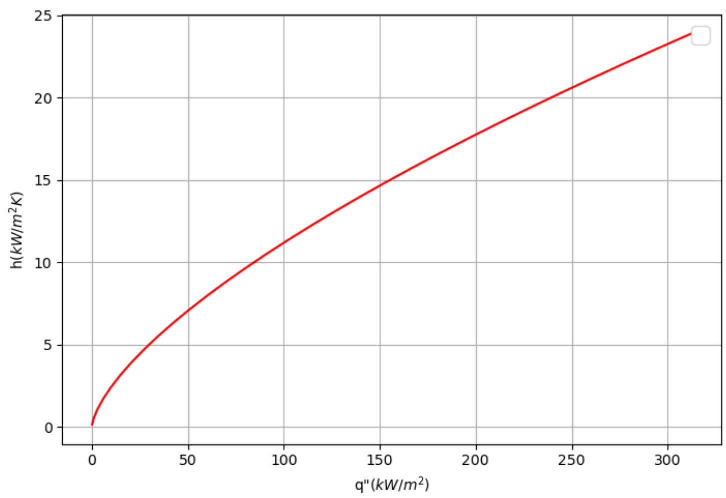
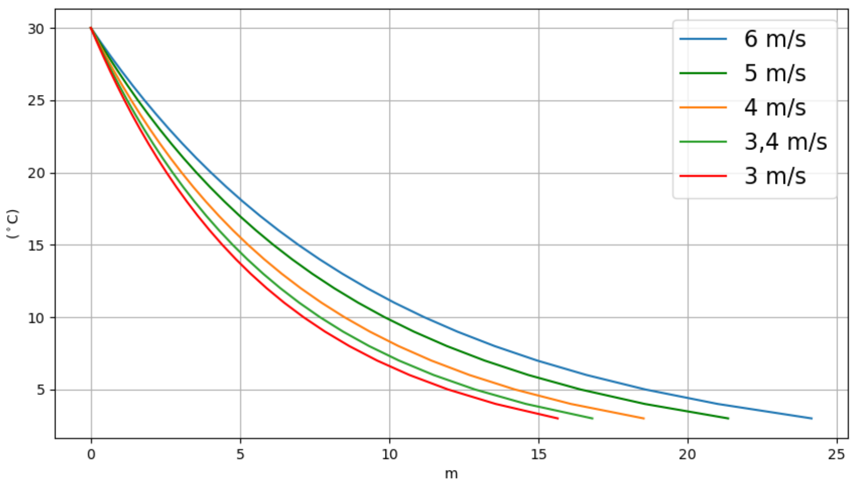
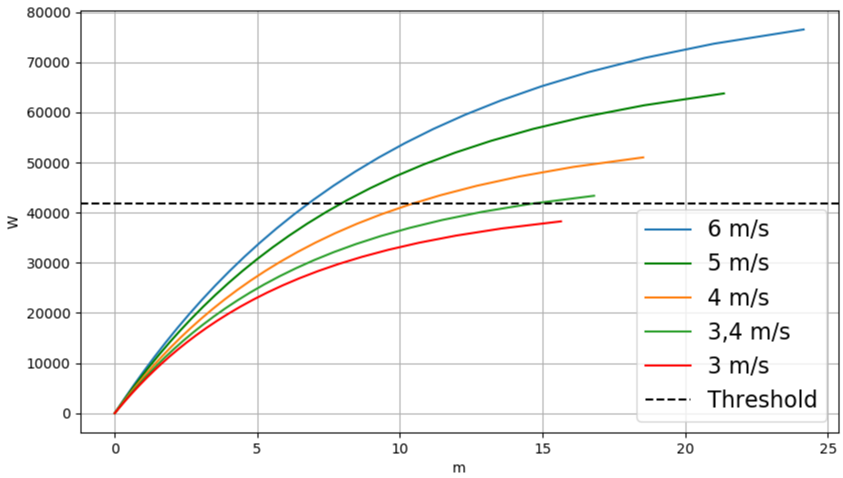


Figure 4.1: Convection coefficient for boiling plotted against the heat flux for the flooded evaporator

The minimum temperature that is possible to reach in the coiled pipe is 3°C before the sectional length became excessive, see figure 4.2a. For all velocities, the temperature drops more rapidly in the first section of the pipe due to the higher temperature difference between water and CO_2. The last $15°C$ needs more than twice the length of the first temperature drop form $30°C$ to $15°C$. From figure 4.2b, it is apparent that the velocity should be above $3{,}4m/s$ to have a safety margin for reaching the threshold of 41,821 kW which leads to a mass flow rate of above 0,38 kg/s.



(a) Temperature drop over pipe length



(b) Delivered power from the single coil

Figure 4.2: Comparison of the water temperature drop in the pipe and the equivalent power of the evaporator

How the overall heat transfer coefficient will evolve in the pipe is shown in figure 4.3. The mass flow rate or velocity has an impact on the coefficient as well as the overall area. Both the convection coefficient of the water and the boiling decrease with the length of the pipe due to the temperature drop of water and the reduced temperature difference of the outer wall of the pipe. The thermal resistance of the wall is also increased due to the increased length. The results start at the outlet of the first pipe section and end when the minimum temperature of $3°C$ is achieved.

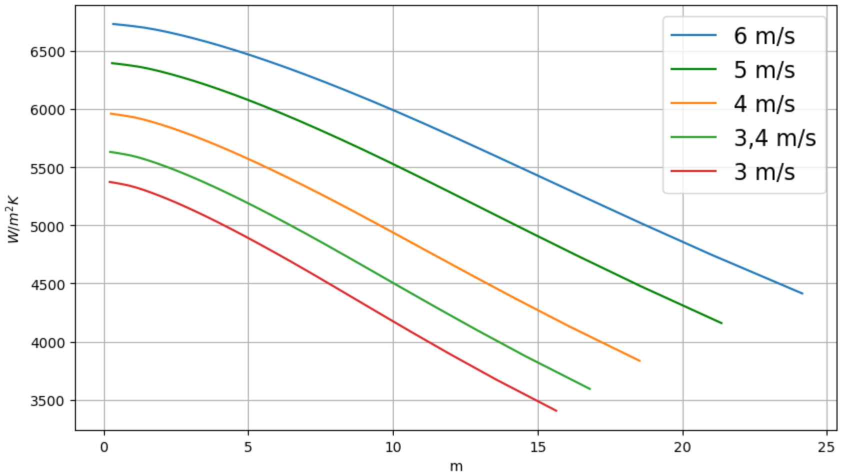
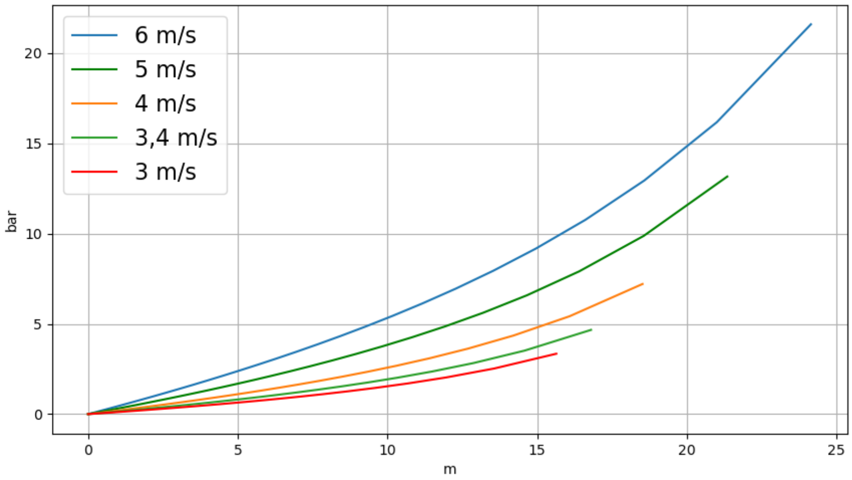
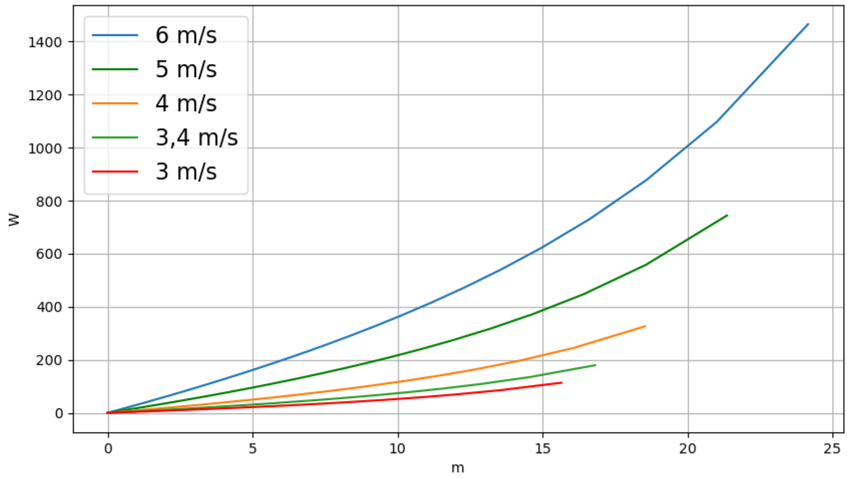


Figure 4.3: Overall heat transfer coefficient plotted over pipe length

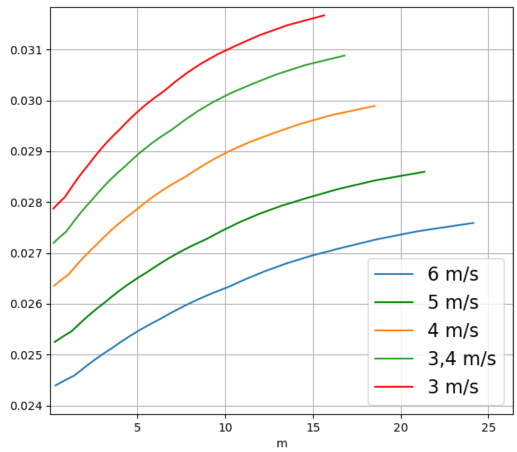
The pressure drop and required pump power in the coiled pipe is shown in figure 4.4a and 4.4b respectively. Velocity is a large contributor to the pressure drop in the pipe, and for the same length, higher velocity increases the pressure drop. The length of the pipe is also a contributor to the difference shown in figure 4.4. Power required by the pump is proportional to the pressure drop and the last sections of the pipe increase the requirement significantly. The friction factor is dependent on the Reynolds number and is therefore higher at lower velocities. It also has a slight increase throughout the pipe but this is not a significant increase and can be seen in figure 4.4c. The heat power output from the pipe is significantly higher than the pumping power required, indicating that the system is energy efficient for all the velocities.



(a) Pressure drop per unit length single coiled pipe



(b) Pump power plotted against the length of single coiled pipe

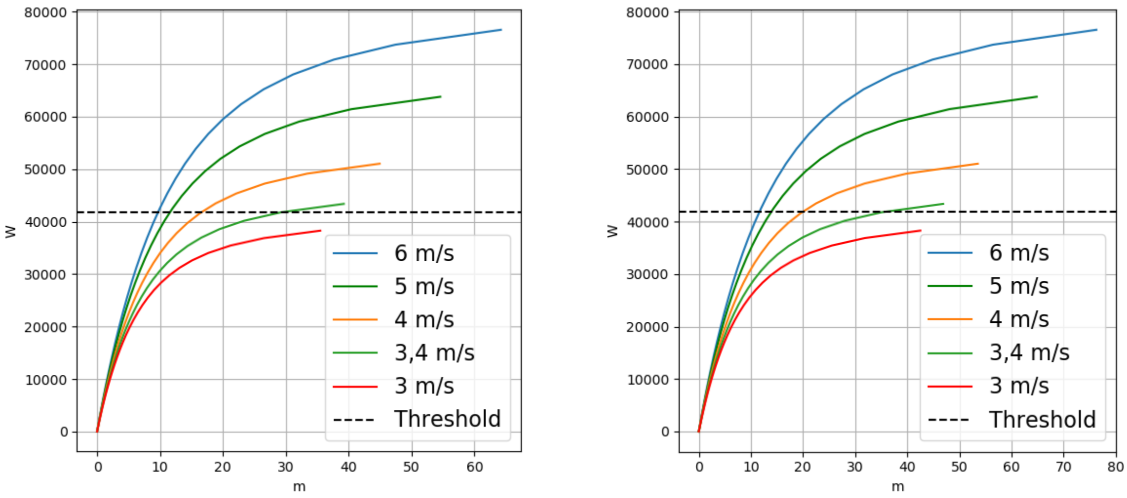


(c) Friction factor plotted against the length of the pipe

Figure 4.4: A comparison of the pressure drop and the equivalent required pump power and the friction factor

Flooded evaporator several pipes

When dividing the pipe into either 2 or 3 pipes the pipe length increases to deliver the same amount of heat as a single coil. The velocity remains the same while the mass flow rate decreases to avoid fouling inside the pipe. The diameters are now 0,0069 m for 3 pipes and 0,0085 m for 2 pipes. The convection coefficient for water and CO_2 increases, but because of the reduction in the cross-sectional area, the length increases to compensate.

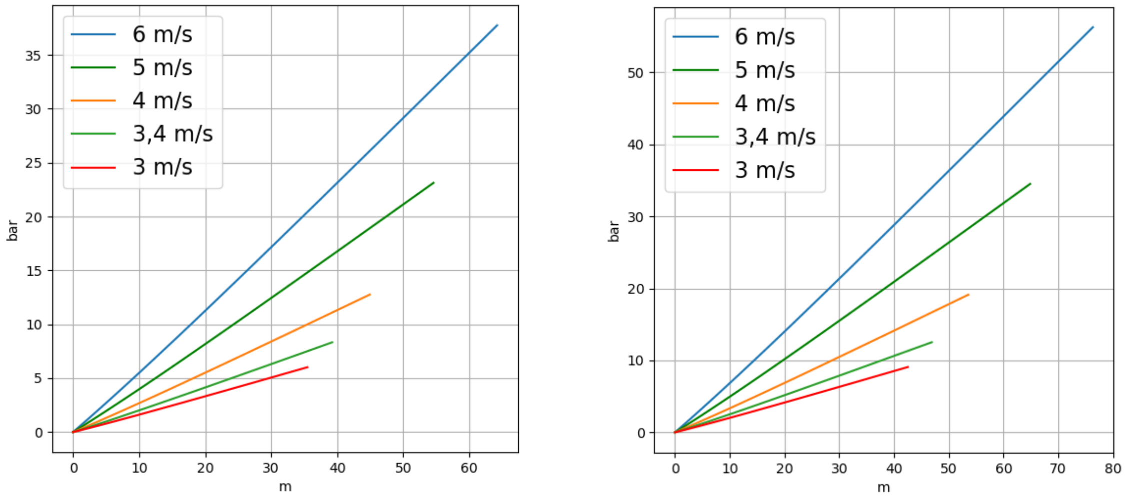


(a) Power plotted over length for a 2 coil configuration

(b) Power plotted over length for a 3 coil configuration

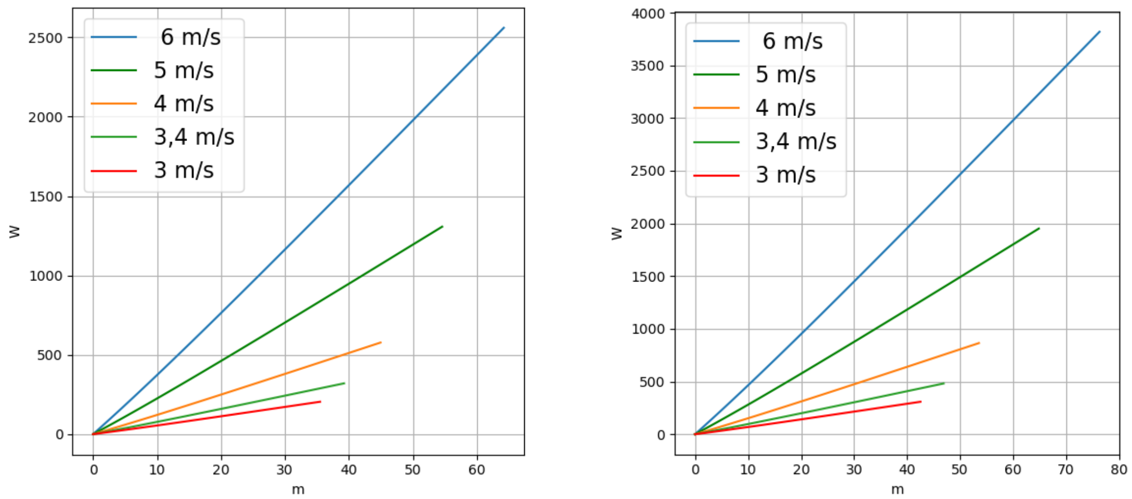
Figure 4.5: A comparison of the necessary pipe length for several pipe configuration

This also affects the pressure drop, see figure 4.7. The reduction in diameter will increase the pressure drop due to the increased friction in the pipe and the increased length to transfer heat. The power requirement also increases because of this. When divided into 3 pipes the power consumption of the pump more than doubles and for 2 pipes it needs an additional 1+ kW.



(a) Pressure drop for 1/2 pipe configuration (b) Pressure drop for 1/3 pipe configuration

Figure 4.6: Comparison of pressure drop for 2 and 3 pipe configuration

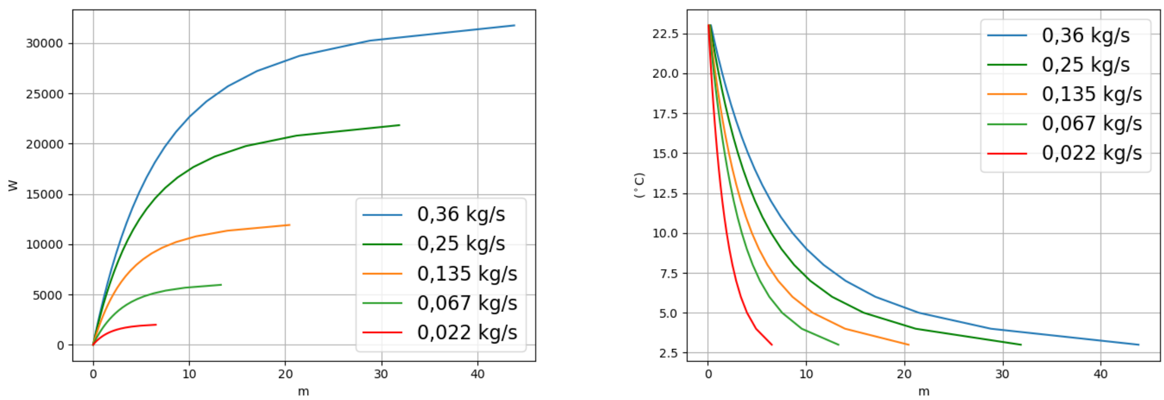


(a) Required pump power for a 2 pipe configuration for all the pipes (b) Required pump power for a 3 pipe configuration for all the pipes

Figure 4.7: Comparison of the required power when dividing into 2 and 3 pipes

Secondary pipe

As mentioned in the method section, the second pipe will have the outlet from the space heating evaporator as a source. Using the same configuration on the pipe, it is possible to reach a power output of slightly more than 30 kW at the highest chosen mass flow rate. The temperature drop has, as in the previous coil, a minimum temperature of 3°C before excessive length must be added to the coil. It is possible to get more power from this pipe with an increased mass flow rate. When the mass flow rate is at 0,135 kg/s or below, the Reynolds number is below critical for either the entire or most of the pipe which reduces the heat transfer. It should therefore at least be above these values. This can be observed in figure 4.9 where the value is small for the lowest values of mass flow rate. The low mass flow rate alone does not account for the low U-value and it is likely also because of the laminar flow.



(a) Power supply at different mass flow rates for second coil

(b) Temperature drop inside pipe for second coil

Figure 4.8: The power and temperature drop over second coil length

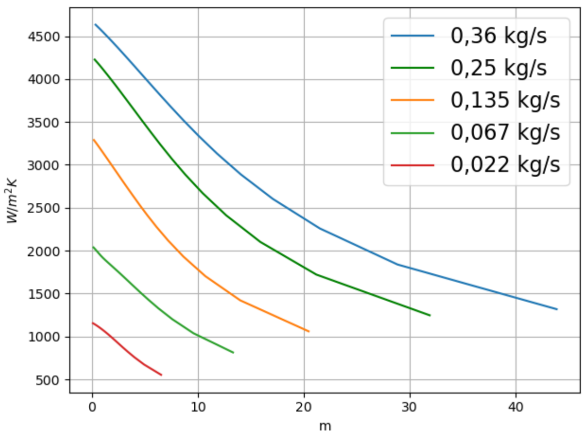


Figure 4.9: The overall heat transfer coefficient for the second pipe

4.2 CO_2 heat pump

In the following heat pump model, the assumed velocity of GW is 4 m/s to achieve the desired power output. The convection coefficient for water ranges from 21344-15182 W/m^2K and for CO_2 from 27437-1709W/m^2K over the pipe length. An average value of this is calculated to input to Dymola. The pipe dimensions in table 3.2 was used in the geometry of the evaporator. The water mass flow rate in the gas cooler has to supply the required hot water for the buildings. Using equation 2.1 and assuming a ΔT of $70K$ the mass flow rate should be 0,205 kg/s.

The initial model, seen in figure 4.10, has no controllers for the compressor or the valve. With all the constant values for the displacement and effective flow area, the water temperature does not reach 75°C. From the Ph-diagram it can also be seen that this heat pump achieves more than the desired high pressure. Figure 4.11 a and b, shows the Ph and Th diagrams show the temperature reached after the compressor is 84 °C and the low pressure side is kept just above 35 bar. The model does not achieve any super heat, and the remaining liquid is removed by the separator before it enters the compressor

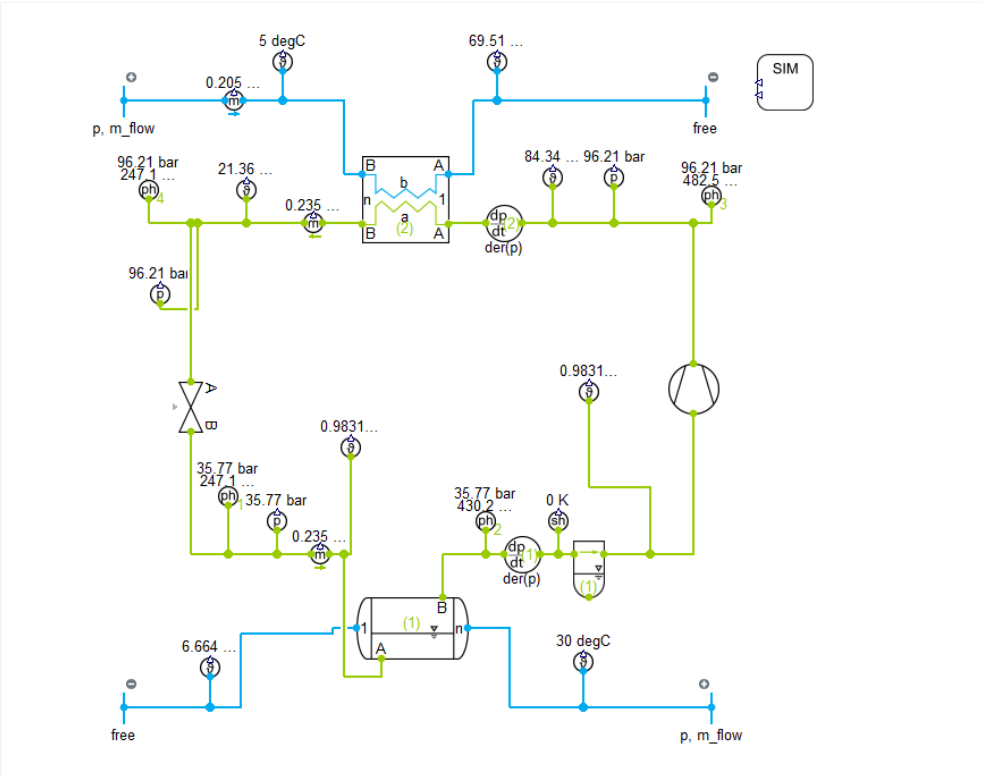
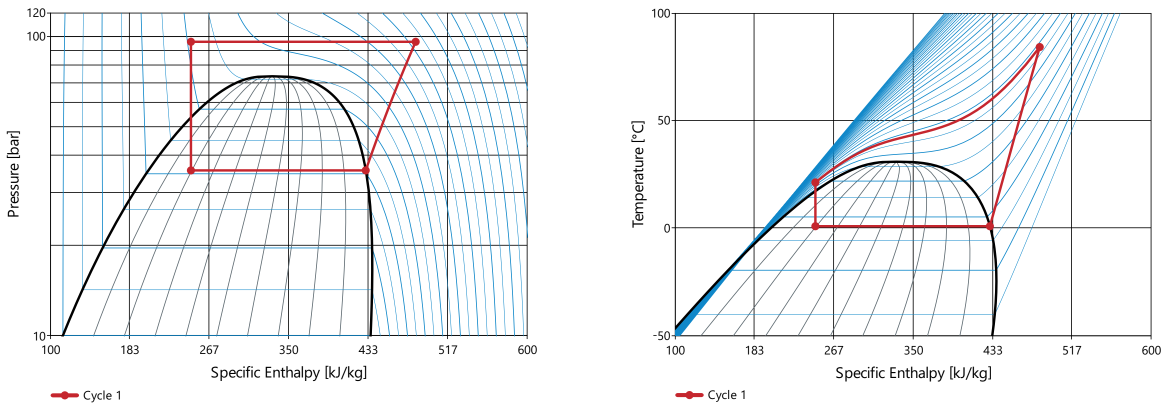


Figure 4.10: Configuration of the initial heat pump model in Dymola

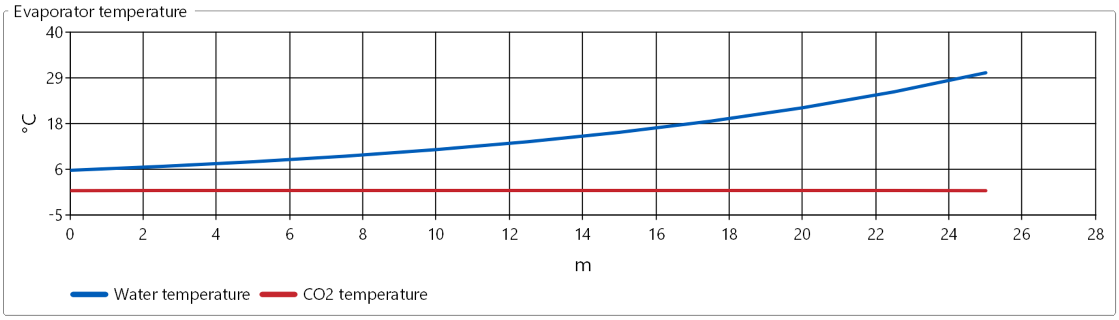


(a) The Ph-diagram of the initial heat pump model

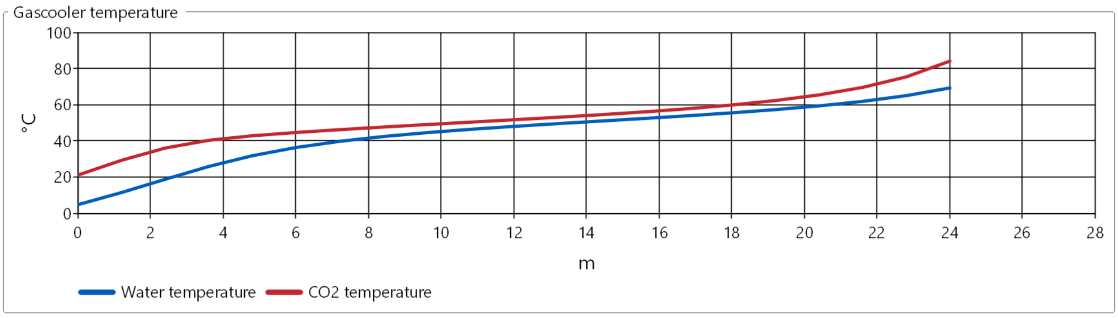
(b) Th-diagram of the initial heat pump model

Figure 4.11: Ph and Th-diagrams of the initial heat pump model

In figure 4.12a and b the temperature over the length of the flooded evaporator and gascooler are shown respectively. The blue color represents water while the red represents the refrigerant. Since the refrigerant does not achieve any superheat, all the heat goes toward evaporation and the CO_2 is kept at about $1°C$. The water reaches an outlet temperature of 6,6$°C$. When comparing to the initial calculations in table 3.4, the compressor discharge temperature is not the same, but the refrigerant mass flow rate matches quite well. The initial model needs to be regulated which will be done in the next model.



(a) How the temperature in the flooded evaporator evolves over time



(b) How the temperature in the plate heat exchanger evolves over time

Figure 4.12: Temperatures at the inlet and outlet of both heat exchangers over time

PI regulated model

In figure 4.13, a PI-regulated heat pump with an internal heat exchanger is shown. In this model, pumps and an internal heat exchanger has been implemented in addition to the controllers. The internal heat exchanger superheats the vapor to 9,9 K. The filling level in the evaporator is set to 0,8 instead of 0,5 to keep the pipe fully submerged. A low K_p and a high T_i value was chosen to evoke a slow response and to avoid too much oscillations. The opening is initially set as the same value as in the initial model with a certain margin. High pressure is kept at 90 bar through the whole simulation, indicating that this PI-regulator is functioning as it should. A second regulator controls the compressor and responds to changes in pressure on the evaporator side and keeps the suction pressure at 35 bar.

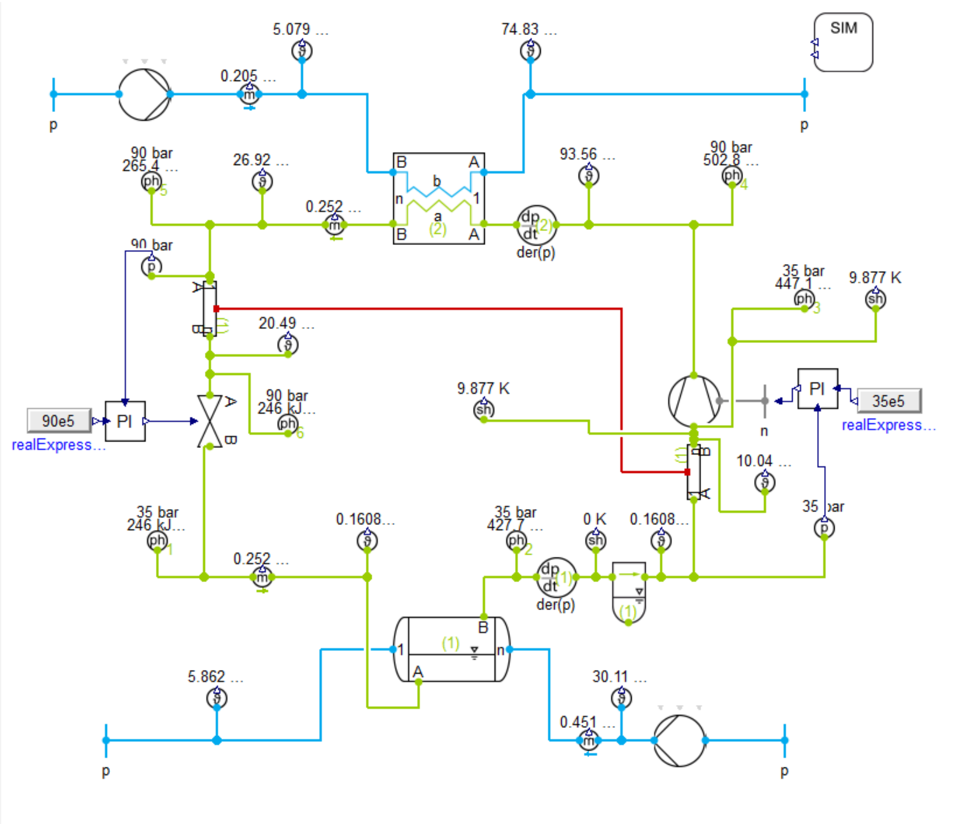
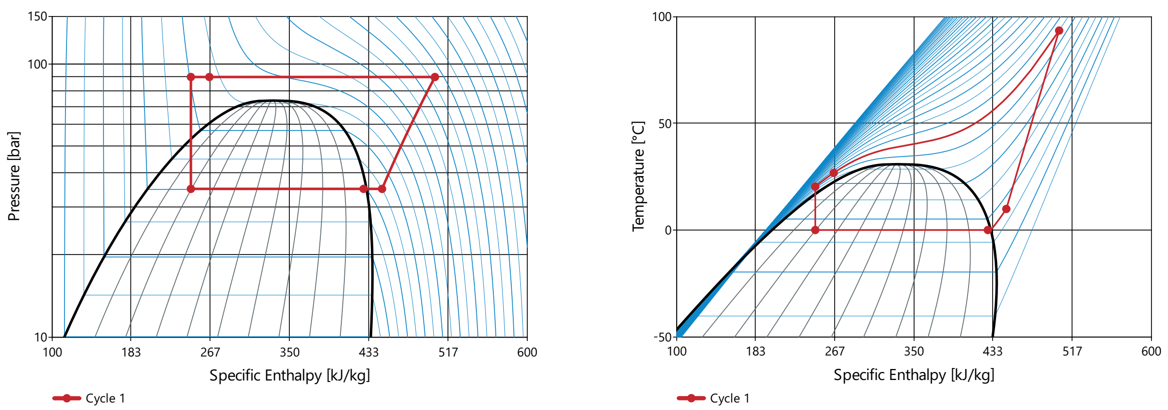


Figure 4.13: Configuration of the PI-regulated heat pump model in Dymola

The heat pump does reach the trans-critical area of operation and has a high compressor outlet temperature. Looking at figure 4.14a and b it is possible to see the amount of superheat produced by the IHX. With a water outlet temperature of 74,83°C, the capacity of the gascooler is 60 kW which is what it is designed for.



(a) Ph-diagram

(b) Th-diagraml

Figure 4.14: Ph and Th-diagrams of the PI-regulated heat pump model

In figure 4.15, the valve opening is monitored over time. The area occilates as the compressor but has significantly lower variations. It stabilizes at just above the initially calculated value as the compressore reaches a speed of 61,5 Hz, see figure 4.16. The suction pressure responds to the change in compressor speed and stabilizes at 35 bar, see figure 4.17.

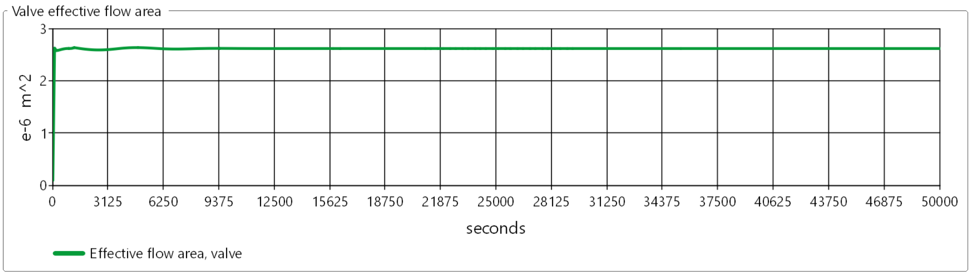


Figure 4.15: The change in effective flow area over time

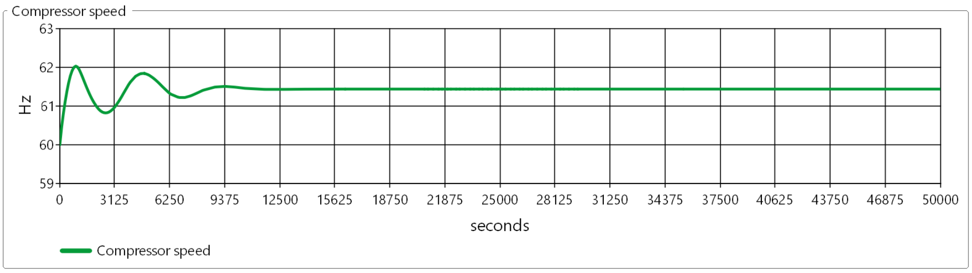


Figure 4.16: The change in compressor speed over time

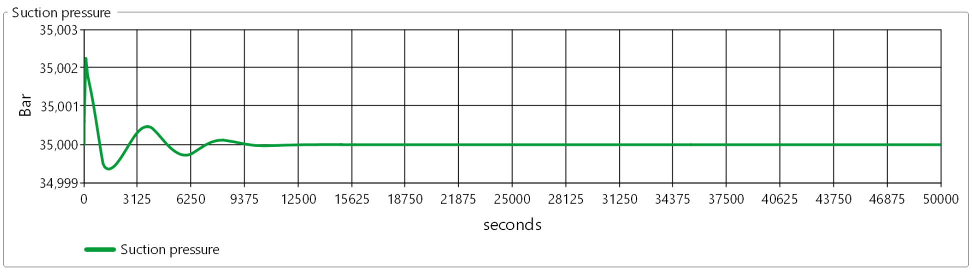


Figure 4.17: The change in suction pressure over time

Evaporator

The results that are the most interesting in this thesis are the heat exchangers. The temperature of the water and the refrigerant can be seen in figure 4.18. Compared to the flooded evaporator results, the water is cooled to reach above the threshold in figure 4.2 which means it is supplying The evaporator has a power transfer of 45,8 kW of power. There is a discrepancy in the area needed since the flooded evaporator is supposed to cool the water further with an 18 m pipe length and this evaporator pipe is longer at 25 m. The filling level shown in figure 4.19, decreases slightly, but this is negligible.

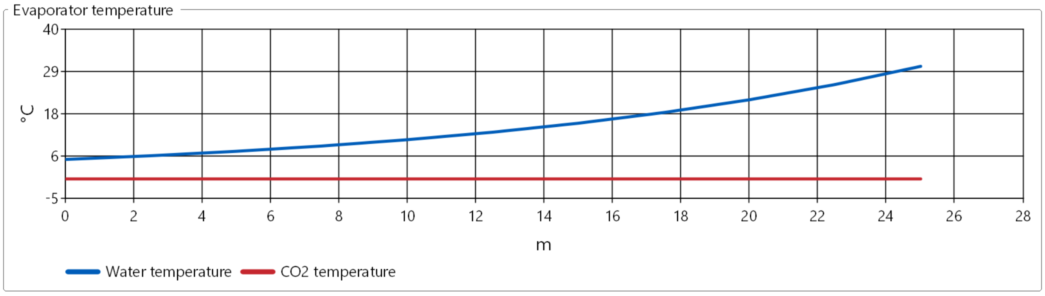


Figure 4.18: The temperature of water and refrigerant in the evaporator

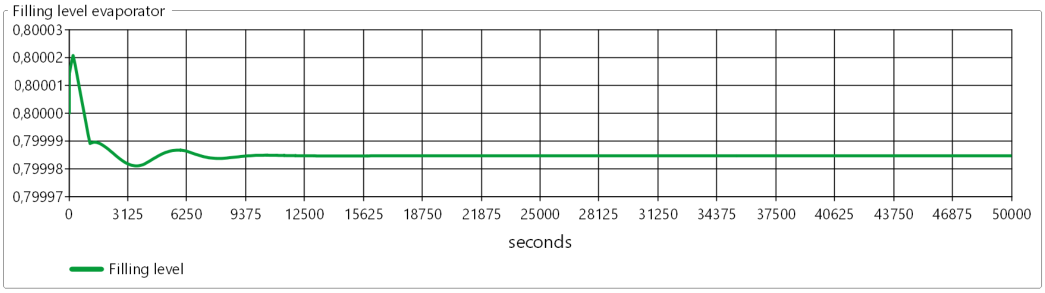


Figure 4.19: The filling level of the evaporator

Gas-cooler

In figure 4.20, the temperature of the water and the refrigerant is plotted over the pipe length. In the mid-section the water and CO_2 temperatures are almost the same, however, at the inlet and outlet the temperature difference increases. The temperatures curves are constant through the simulation since the mass flow rate is not very affected by the oscillations in the compressor speed, see figure 4.21. It does oscillate slightly, but again it is negligible.

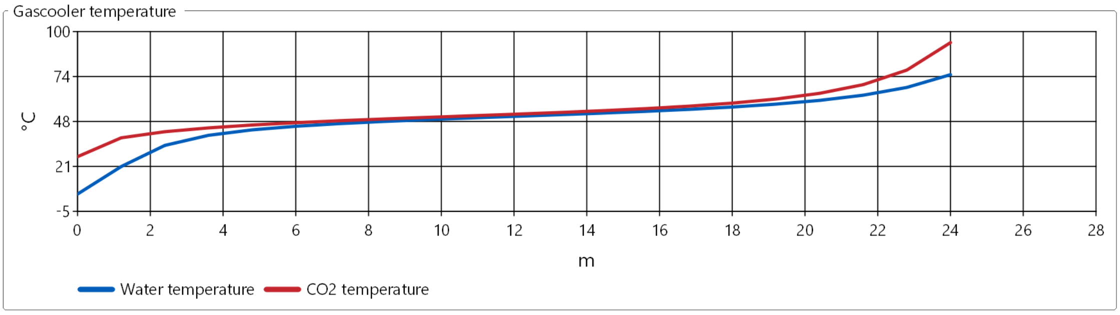


Figure 4.20: The temperature of water and refrigerant in the gas-cooler

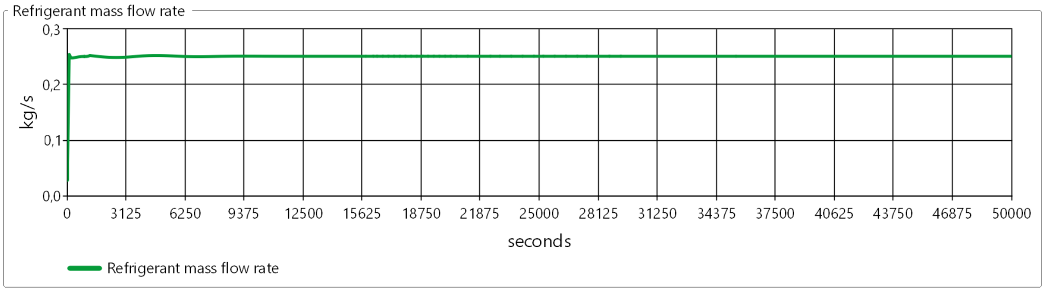


Figure 4.21: The mass flow rate through the gas-cooler

When the heat pump is in steady-state the COP is found by using equation 2.32. It is 3.

4.3 Energy and cost analysis field B1

With the chosen mass flow rate, the GW will be sufficient to supply the heat pump for 15,179 hours. Figure 4.22 shows the remaining demand that needs to be supplied when the GW runs out.

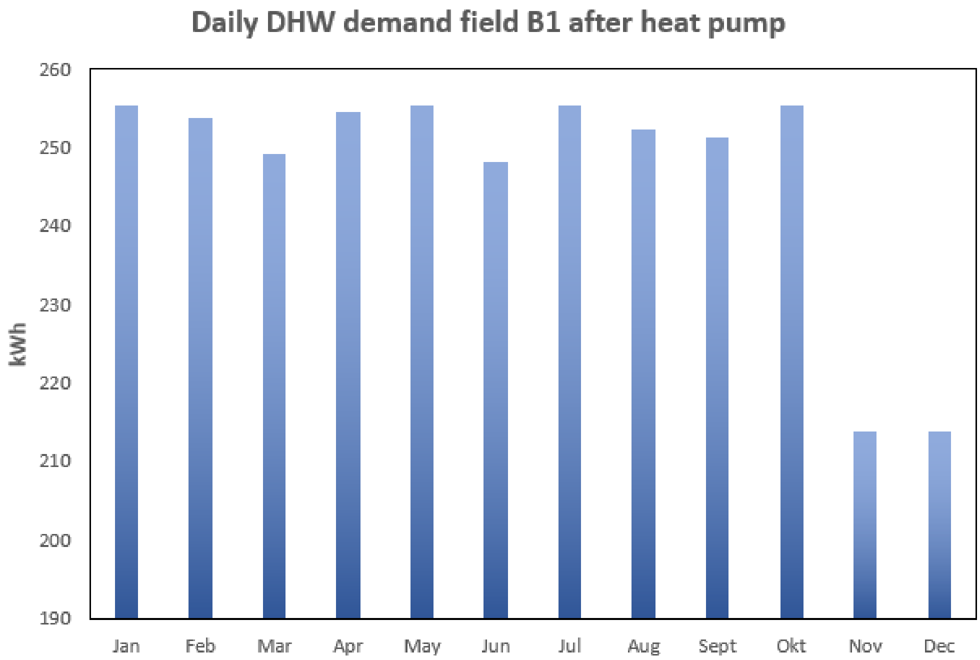


Figure 4.22: DHW demand after the GW supply is empty

The gascooler output is 1200 kWh per day and the pumps and compressor uses approximately 297 kWh when running at steady state, see figure 4.23. Looking at figure 4.24 the price for electricity and the price for this amount of district heating is shown. The spot price for electricity is set to 109 kr/MWh. The saving potential using a heat pump is 8420 $kr/month$.

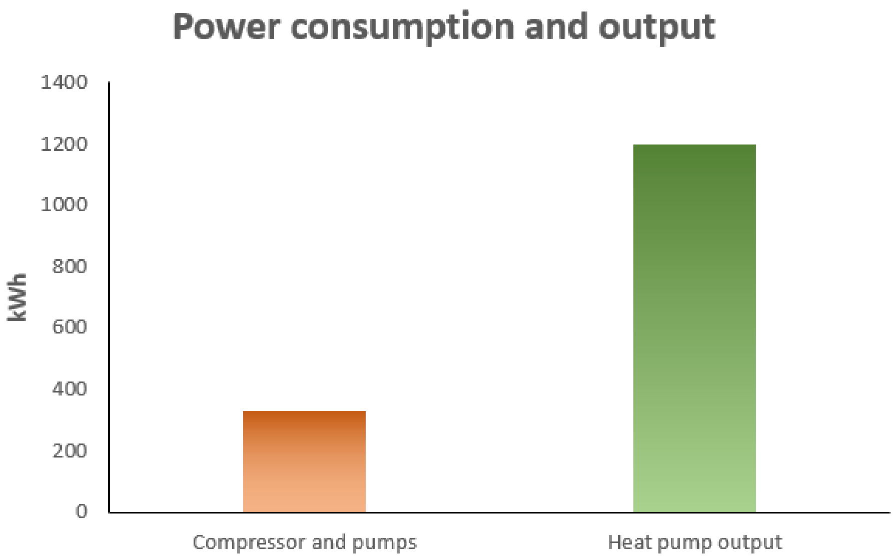


Figure 4.23: Power from pumps and compressor compared to the output from the gas cooler during 20 hours of operation

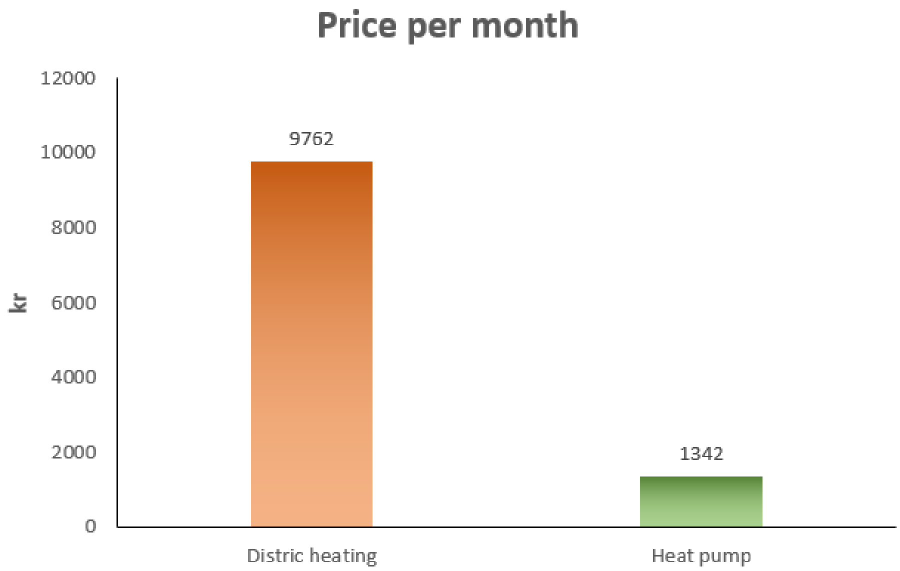


Figure 4.24: Price per month using district heating compared to heat pump electricity demand

Chapter 5

Discussion

5.1 Flooded evaporator

The heat flux calculated in the model is dependent on the Rohsenow coefficient which is assumed. A material fluid combination of n-pentane-copper was chosen because a coefficient for CO_2 and stainless steel was not found since it has to be found empirically through experiments. This can impact the calculations for the CO_2 convection coefficient and should be considered when reviewing the results. Both the heat flux and the convection coefficient significantly drops in the last section of the pipe due to the low-temperature difference and heat flux, and this is the reason that the water can not realistically reach a lower temperature than $3°C$. When comparing to the results found in the experimental study, the convection coefficient over the heat flux is significantly lower in the calculations done in this thesis which is due to the temperature difference being higher, see figure 4.1 and 2.15. The Roshesnow correlation has an error of up 20% [7].

When dividing the coil into several smaller pipes, the length needs to be increased. This is probably mostly due to a decrease in heat transfer area. This is a probable cause since the pipes have a wall thickness of 0,5mm less than the single coil and the convection coefficients are higher. The Nusselt number on the other hand is decreased because the factor $\frac{d}{D}$ is less. Another cause can be that the flow is significantly less turbulent.

The pressure drop is also different for the single-coil and several coils. For more than one coil, the drop is linear while for a single coil it increases more as the pipe gets longer. The reason for this is not understood and might be a calculation error. For smaller pipes, the pressure drop is higher because of increased friction in the pipe and longer pipes.

The secondary pipe can be used either when the GW has run out, or parallel to the main pipe. This can reduce the mass flow rate of GW needed which might decrease pump power. As this pipe also needs to pump the water it is not certain that this is more efficient and should be examined. If the secondary pipe is compensating when the GW tank is empty it will need a higher mass flow rate to reach the desired evaporator power.

In the equation for a coiled pipe, there is a $\pm 15\%$ deviation in the results compared to measured values. Some of this is due to the correctional factor that considers the difference in the pipe wall temperature and the fluid temperature. In the calculations in this thesis, this factor is 1 because it is assumed an equal temperature on the wall surface and the fluid.

5.2 CO_2 heat pump

The compressor speed is oscillating at the start of the simulation before it eventually stabilizes. This is probably due to poor choice of starting values and the values for gain and integral time. It is still a relatively slow response and is most likely safe for the compressor.

The flooded evaporator is functioning as it should since there is 0 K superheat in the vapor at the outlet. When comparing the evaporator model to how the Dymola model functions, it is apparent that in Dymola, more length is needed for the same amount of heat transfer. A reason might be the average heat transfer coefficient chosen. In the Python model, the pipe transfers more heat in the beginning compared to at the end, the heat transfer coefficients does not decrease linearly. For the Dymola model, one value is chosen for the entire pipe which might cause the pipe length to be different. When choosing the "correct" length, there is not enough vapor produced to lift the water temperature to the desired level. Another reason can be the thermal conductivity of the stainless steel. In the Python model it is set for a certain type of stainless steel while in Dymola it is not known which value is used. It is uncertain how much of an impact this has.

The evaporator needs more power produce enough vapor for the gascooler. The efficiency of the compressor is set to 0,7 and can be a reason for this.

5.3 Energy and cost analysis

The cost analysis does not consider how much the energy system will cost since it is unknown at this time. This should be considered to see the actual profitability of using the proposed energy system. It also only considers the CO_2 heat pump. Space heating needs electricity too and the savings potential is likely higher than when only looking at the DHW heat pump. The amount of heat in the GW is not calculated for the entire area due to the uncertainty in the amount it will produce. It might be misleading to include it and the estimate from B1 also has some uncertainty since it is based on assumptions on the user patterns.

Chapter 6

Conclusion

A CO_2 heat pump will be used to supply DHW to a building complex consisting of 10600 m^2. Water will be heated from 5°C to 75-85°C. The main source for the evaporator will be 24645 kg of greywater produced every day from the apartments. A flooded evaporator is used to increase heat transfer from the greywater to the refrigerant and the pipe will be coiled to increase it further. To avoid fouling in the evaporator pipe due to impurities in the greywater, high velocity is a criterion. To avoid a high mass flow rate, the inner diameter is set to be 12mm with a wall thickness of 1,5mm. The bending diameter of the coil is set to be 0,3m. The results from calculations done on the flooded evaporator were then put into a heat pump model in Dymola.

The demand for domestic hot water in the field B1 is just under 1200 kWh every day on average and the heat pump must be able to supply 60 kW since it is operating 20 hours a day. The coiled pipe is capable of transferring 45,8 kW of power in the Dymola model when the water velocity is $4m/s$ and an outlet temperature of 75°C. The gascooler capacity is then 60 kW. Greywater can supply the heat pump for 15,179 of the 20 hours and a backup must supply the remaining power that ranges from 210-260 kWh. The heat pump has a COP of 3.

The backup coil can supply the necessary amounts of heat when using the space heating evaporator outlet. With a mass flow rate of 0,36 kg/s it can supply an additional 30 kW but needs to be longer than the GW coil. This mass flow can be increased if this coil will run when the GW is empty and needs to reach the threshold of 41,821kW. When dividing the pipe into several smaller pipes, the heat transfer per m^2 is reduced and pressure drop increases. It is therefore better to have a single coil with 12mm inner diameter than several smaller.

Using a heat pump instead of district heating can save 8420 kr every month when comparing the price for electricity and district heating only.

Chapter 7

Further work

This thesis takes a closer look at the B1 field and its DHW heat pump. All the different fields will have different sized heat pumps and a more comprehensive analysis for the entire building area should be done. This can include evaluating how well larger heat pumps can operate and the savings that can be made from using the waste heat instead of district heating.

Work should be done on the amount of heat that will be transferred to the GW during summer to see how this will affect the DHW heat pump. It might be possible that this increase in temperature eliminates the need for the backup coil during these months. To consider this accurately it is should be examined in detail since it still has to satisfy the high-velocity criteria. It would also be interesting to see how the heat pump reacts to increase in power demand. It might be possible to control the mass flow rate of GW into the evaporator to produce more power. The compressor can run at higher speed and simulations could be run to see how it affects the COP and if the controllers can handle change in capacity. Power demand for the water that is going in to storage could also be calculated. This does affect the overall efficiency of the system and could lead to a more realistic cost analysis.

The cost analysis should be improved and take in to consideration the investment cost of the heat pump and GW tank. It should be compared to the investment of connecting to the district heating grid to see the actual profitability and not just monthly savings.

Space heating is neglected in this thesis, and it can be useful to consider the energy coverage from the ice skating rink and the total heating demand. Space cooling should also be evaluated.

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Appendix

A

This is the python code used to calculate the heat transfer in the flooded evaporator. The results are summarized in section 4.1. The code can be seen on the next page.

```
#Geometry and heat transfer calculations for the flooded evaporator
import numpy as np
import matplotlib.pyplot as plt
import scipy as scp
#####################################  The properties of the pipe flow and geometry   ################################
d = 0.012                                   # [m] Inner diameter of pipe
d_o = 0.015                                 # Outer diameter
n = 90                                      # [-] Number of turns
height = 0.03                               # [m] distance between coils
V = np.arange(3,6.2,0.2)                    # m/s This needs 16 steps
print('Velocities:',V,'m/s')
rho = 997                                   # [kg/m**3] density
Cp=4190                                     # [J/kgK] Specific heat
k = 14.9                                    #Conduction of stainless steel of type AISI 304
T_co2 = 273.15
T_range = np.arange(303.15,273.15,-1)       #Temperature range from 30-0 degrees
T_range1 = np.arange(303.15,275.15,-1)      #Temperature range from 30 to 3 degrees

#       Boiling constants       #
Csf = 0.0154
n_boil = 1.7
dynvisc = 105.4*10**(-6)
pr_boil = 2.38
hfg = 230.41*10**3
surf_stresstens = 4.57*10**(-3)
dens_l = 932.85
dens_v = 98.145
g = 9.81
Cp_l = 2.5185 *10**3

#The mass flow rate and the power at different speeds
m_dot = V * np.pi * (d / 2) ** 2  * rho
q = m_dot * Cp
print('Mass flowrate',m_dot,'kg/s')
print('Power over a temperature drop of 1K',q,'W')


LMTD = []
for i in range(len(T_range)):
    if i+1<len(T_range):
        lmtd = ((T_co2-T_range[i])-(T_co2-T_range[i+1]))/(np.log((T_co2-T_range[i])/(T_co2-T_range[i+1])))
        LMTD.append(lmtd*(-1))
    else:
        lmtd = ((T_co2 - T_range[i]) - (T_co2 - 273.16)) / (np.log((T_co2 - T_range[i]) / (T_co2 - 273.16)))
        LMTD.append(lmtd * (-1))
print(LMTD)

########################################################################################################################
############################### Interpolerte verdier ###################################################################
########################################################################################################################
print()
print('Interpolated values')
print()
T_range_table = [305,300,295,290,285,280,275,273.15]
visc_range_table = [796*10**(-6),855*10**(-6), 959*10**(-6),1080*10**(-6),1225*10**(-6),1422*10**(-6),1652*10**(-6),1750
*10**(-6)]
visc= []
for i in range(len(T_range)):
    vask = np.interp(T_range[i], T_range_table[::-1], visc_range_table[::-1])
    visc.append(vask)
print('Viscosity array for temperature range',visc,'N*s/m**2')

#            Reynolds number     #
Re_d = ([],[],[],[],[],[],[],[],[],[],[],[],[],[],[],[])
for i in range(len(m_dot)):
    for j in range(len(visc)):
        Re_di = (4*m_dot[i])/(np.pi*d*visc[j])
        Re_d[i].append(Re_di)
print(len(Re_d[0]),'Reynolds number for temperature range',Re_d)

#            Prnumbers           #
pr = []
pr_tab = [5.2,5.83,6.62,7.56,8.81,10.26,12.22,12.99]
for i in range(len(T_range)):
    pra = np.interp(T_range[i], T_range_table[::-1], pr_tab[::-1])
    pr.append(pra)
print('Prandtl number',pr)

#            Conduction of water         #
k_f_w = []
k_f_w_tab = [620*10**(-3), 613*10**(-3), 606*10**(-3),598*10**(-3),590*10**(-3),582*10**(-3),574*10**(-3),569*10**(-3)]
for i in range(len(T_range)):
    kfw = np.interp(T_range[i], T_range_table[::-1], k_f_w_tab[::-1])
    k_f_w.append(kfw)
print('Conduction Water', k_f_w,'W/mK')

########################################################################################################################
######################################## Straight pipe calculations  ###################################################
########################################################################################################################
print()
```

```
print('Straight pipe calculations')
print()
#           Nusslet number loop          #
nusslet = ([],[],[],[],[],[],[],[],[],[],[],[],[],[],[],[])
for i in range(len(Re_d)):
    for j in range(len(Re_d[i])):
        if Re_d[i][j] < 2300:
            nus = 3.66
            nusslet[i].append(nus)
        elif 2300<=Re_d[i][j]<=10000:
            nus = 0.023*(Re_d[i][j])**(4/5)*pr[j]**0.3                  #0.3 is since the water is cooling
            nusslet[i].append(nus)
        else:
            f = (0.790 * np.log(Re_d[i][j]) - 1.64)**(-2)
            nus = ((f / 8) * (Re_d[i][j] - 1000) * pr[j]) / (1 + (12.7 * (f / 8) ** 0.5) * (pr[j] ** (2 / 3) - 1))
            nusslet[i].append(nus)
print('Nusslet numbers Straight pipe',nusslet)

#           Convection coefficient straight pipe    #
h = ([],[],[],[],[],[],[],[],[],[],[],[],[],[],[],[])
for i in range(len(nusslet)):
    for j in range(len(k_f_w)):
        hh=nusslet[i][j]*k_f_w[j]/d
        h[i].append(hh)
print('convection coefficient',h,'W/m**2K')

#           Length of straight pipe          #
Ltot  = ([],[],[],[],[],[],[],[],[],[],[],[],[],[],[],[])
L  = ([],[],[],[],[],[],[],[],[],[],[],[],[],[],[],[])
L_section = ([],[],[],[],[],[],[],[],[],[],[],[],[],[],[],[])
for i in range(len(h)):
    Le=0
    summ=0
    for j in range(len(h[i])):
        Lens=q[i]/(2*np.pi*LMTD[j])*((1/(h[i][j]*d/2))+np.log(d_o/d)/(k))
        L_section[i].append(Lens)
        Le=Le+Lens
        L[i].append(Le)
        summ=summ+Lens
    Ltot[i].append(summ)
print('Cumulative pipe length (straight)',L)
print('Total pipe length (Straight)',Ltot)
print('sectional length pipe (straight)',L_section)

q_15 = ([],[],[],[],[],[],[],[],[],[],[],[],[],[],[],[])
for i in range(len(q_15)):
    cumulat = 0
    q_15[i].append(cumulat)
    for j in range(len(L[i])-1):
        cumulat = cumulat+q[i]
        q_15[i].append(cumulat)
print('Power cumulative (straight)',q_15)


#                       Figure straight pipe             #
plt.figure('Temperature area correlation straight pipe')
plt.grid()
plt.plot(L[15], q_15[15], '-', label = '6 m/s')
plt.plot(L[10], q_15[10], '-',color='g', label = '5 m/s')
plt.plot(L[7], q_15[7], '-', label = '4,4 m/s')
plt.plot(L[4], q_15[4], '-', label = '3,8 m/s')
plt.plot(L[0], q_15[0], '-',color='r', label = '3 m/s')
plt.axhline(y = 41821, linestyle = '--', color = 'black', label = 'Threshold')
plt.ylabel(r'W')
plt.xlabel('m')
plt.legend(fontsize = 16)
plt.tight_layout()
#plt.show()
plt.close()


#######################################################################################################################
##################################   Coil calculations   ##############################################################
#######################################################################################################################
print()
print('Coil calculations')
print()

#           Geometry of coil        #

D=0.3
Recrit=2300*(1+8.6*(d/D)**0.45)
print('Critical Reynolds',Recrit)


#           Nusslet number coil      #

```

```python
nusslet2 = ([],[],[],[],[],[],[],[],[],[],[],[],[],[],[],[])
for i in range(len(Re_d)):
    for j in range(len(Re_d[i])):
        if Re_d[i][j] < Recrit:
            nus2 = 3.66+0.08*(1+0.8*(d/D)**0.9)*Re_d[i][j]**(0.5+0.2903*(d/D)**0.194)*pr[j]**(1/3)*(pr[j]/pr[j])**0.14
            nusslet2[i].append(nus2)
        elif Recrit<=Re_d[i][j]<=22000:
            gimmi = (22000-Re_d[i][j])/(22000-Recrit)
            Nu1 = 3.66+0.08*(1+0.8*(d/D)**0.9)*Recrit**(0.5+0.2903*(d/D)**0.194)*pr[j]**(1/3)*(pr[j]/pr[j])**0.14
            fcrit = ((0.3164 / 22000**0.25) + 0.03 * (d / D) ** 0.5) * (visc[j]/ visc[j]) ** 0.27
            Nut = (((fcrit / 8) * 22000 * pr[j]) / (1 + (12.7 * np.sqrt(fcrit / 8)) * (pr[j] ** (2 / 3) - 1))) * (pr[j] / pr[j]) ** 0.14
            nus2 = gimmi*Nu1+(1-gimmi)*Nut                    #0.3 is since the water is cooling
            nusslet2[i].append(nus2)
        else:
            f2 = ((0.3164/Re_d[i][j]**0.25)+0.03*(d/D)**0.5)*(visc[j]/visc[j])**0.27
            nus2 = (((f2 / 8) * Re_d[i][j] * pr[j]) / (1 + (12.7 * np.sqrt(f2 / 8) ) * (pr[j] ** (2 / 3) - 1)))*(pr[j]/pr[j])**0.14
            nusslet2[i].append(nus2)
print('nusslet nr 2', nusslet2)

#           Convection coefficient coil     #
h= ([],[],[],[],[],[],[],[],[],[],[],[],[],[],[],[])
for i in range(len(nusslet2)):
    for j in range(len(k_f_w)):
        hh=nusslet2[i][j]*k_f_w[j]/d
        h[i].append(hh)

eps=10
while eps>0.1:
    T_o = ([], [], [], [], [], [], [], [], [], [], [], [], [], [], [], [])
    for i in range(len(L_section)):
        for j in range(len(L_section[i])):
            Rcond = np.log((d_o / 2) / (d / 2)) / (2 * np.pi * k * L_section[i][j])
            To = T_range[j] - q[i] * Rcond
            T_o[i].append(To)


    #           Boiling heat flux       #
    qm2_boil = ([], [], [], [], [], [], [], [], [], [], [], [], [], [], [], [])
    qm2_boilkW = ([], [], [], [], [], [], [], [], [], [], [], [], [], [], [], [])
    for i in range(len(qm2_boil)):
        for j in range(len(T_o[i])):
            qm2 = dynvisc * hfg * ((g * (dens_l - dens_v)) / surf_stresstens) ** (0.5) * (
                        (Cp_l * (T_o[i][j] - T_co2)) / (Csf * hfg * pr_boil ** n_boil)) ** 3
            qm2_boil[i].append(qm2)
            qm2_boilkW[i].append(qm2/1000)


    #           Boiling convection coefficient      #
    h_boil = ([], [], [], [], [], [], [], [], [], [], [], [], [], [], [], [])
    h_boilkW =([], [], [], [], [], [], [], [], [], [], [], [], [], [], [], [])
    for i in range(len(h_boil)):
        for j in range(len(qm2_boil[i])):
            hboil = qm2_boil[i][j] / (T_o[i][j] - T_co2)
            h_boilkW[i].append(hboil/1000)
            h_boil[i].append(hboil)


    Rcond = np.log((d_o / 2) / (d / 2)) / (2 * np.pi * k)
    Ltotcoilnew = ([], [], [], [], [], [], [], [], [], [], [], [], [], [], [], [])
    L_sectionnew = ([], [], [], [], [], [], [], [], [], [], [], [], [], [], [], [])
    Lnew = ([], [], [], [], [], [], [], [], [], [], [], [], [], [], [], [])
    for i in range(len(h_boil)):
        Le = 0
        summ = 0
        Lnew[i].append(Le)
        for j in range(len(h_boil[i])):
            Lens = q[i] / (2 * np.pi * LMTD[j]) * (
                        (1 / (h[i][j] * d / 2)) + 1 / (h_boil[i][j] * d_o / 2) + np.log(d_o / d) / (k))
            L_sectionnew[i].append(Lens)
            Le = Le + Lens
            Lnew[i].append(Le)
            summ = summ + Lens
        Ltotcoilnew[i].append(summ)



    eps=np.abs(Ltotcoilnew[15][0]-Ltot[15][0])
    print(eps)
    L_section=L_sectionnew
    Ltot=Ltotcoilnew
    Lnew=L

print('Outer wall temperature: (K)', T_o)
print('heat flux boiling W/m^2', qm2_boil)
print('Convection coefficient boiling', h_boil)
print('convection coefficient', h, 'W/m**2K')
print(len(Lnew[0]),'Length cumulative (coil)', Lnew)
print('total pipe length (coil)', Ltotcoilnew)
```

```
print('Length of each section(coil)', L_sectionnew)



print(len(Lnew[0]),'L',Lnew)
print(len(q_15[0]),'q_15',q_15)
plt.figure('Convection coefficient over heat flux')
plt.grid()

plt.plot(qm2_boilkW[15], h_boilkW[15], '-',color='r')


plt.ylabel(r'h($kW/m^2K$)')
plt.xlabel('q"($kW/m^2$)')
plt.legend(fontsize = 16)
plt.tight_layout()
#plt.show()
#plt.close()

##          Pressure drop          ##
#          Friction coefficient    #
Zeta=([],[],[],[],[],[],[],[],[],[],[],[],[],[],[],[])
for i in range(len(Re_d)):
    zeta=0
    for j in range(len(Re_d[i])):
        if 1<Re_d[i][j]*np.sqrt(d/D)<Recrit*np.sqrt(d/D):
            zeta = (64/Re_d[i][j])*(1+0.033*np.log10(Re_d[i][j]*np.sqrt(d/D)))**4
            Zeta[i].append(zeta)
        if Recrit<Re_d[i][j]<10**5:
            zeta = (0.3164/(Re_d[i][j]**(0.25)))*(1+0.095*(d/D)**(0.5)*Re_d[i][j]**(0.25))
            Zeta[i].append(zeta)
        if 10**5<Re_d[i][j]<2*10**6: #This is a straight pipe relation so it might not be correct
            zeta=0.00540+0.3964/(Re_d[i][j]**0.3)
            Zeta[i].append(zeta)

#      Pressure drop      #
print(len(V),len(L_section[0]), len(Re_d[0]), len(Zeta[0]))
deltTa =([],[],[],[],[],[],[],[],[],[],[],[],[],[],[],[])
P_drop_kPa= ([],[],[],[],[],[],[],[],[],[],[],[],[],[],[],[])
for i in range(len(Zeta)):
    deltT=0
    deltTa[i].append(deltT)
    P_drop_kPa[i].append(deltT)
    for j in range (len(Zeta[i])):
        dP = Zeta[i][j]*(L_sectionnew[i][j]/d)*((rho*V[i]**2)/2)
        deltT = deltT+dP
        deltTa[i].append(deltT / 100000)
        P_drop_kPa[i].append(deltT/1000)

print('friction',Zeta)
print('Lengthhhhhhh',L_sectionnew)
print(len(deltTa[0]),'Pressure drop in bar',deltTa)
print('Pressure drop kPa',P_drop_kPa)
print('Pressure drop bar 4m/s',deltTa[5])



#As we can see the pressure drop is very dependent on the inner diameter compared to the length.
for i in range(len(L)):
    Lnew[i].insert(0,0)
    Zeta[i].insert(0,0)

q_15 = ([],[],[],[],[],[],[],[],[],[],[],[],[],[],[],[])
for i in range(len(q_15)):
    cumulat = 0
    q_15[i].append(cumulat)
    for j in range(len(L[i])-1): #this needs - 1
        cumulat = cumulat+q[i]
        q_15[i].append(cumulat)
print('Power cumulative (coil)',q_15)
#      Average area of the coil sections      #
Am2 = ([],[],[],[],[],[],[],[],[],[],[],[],[],[],[],[])
for i in range(len(L)):
    for j in range(len(L[i])):
        am2 = (d - d_o) / (np.log(d / d_o)) * np.pi * Lnew[i][j]
        Am2[i].append(am2)
print('mean area sections cumulative', Am2)
print(len(L[0]),'Section length', L)

#      Remove the last ones to make better graphs
for i in range(len(Am2)):
    Am2[i].pop(30)
    Am2[i].pop(29)
    Am2[i].pop(28)

for i in range(len(Lnew)):
    Lnew[i].pop(30)
    Lnew[i].pop(29)
    Lnew[i].pop(28)

```

```
for i in range(len(q_15)):
    q_15[i].pop(30)
    q_15[i].pop(29)
    q_15[i].pop(28)

for i in range(len(deltTa)):
    deltTa[i].pop(30)
    Zeta[i].pop(30)
    deltTa[i].pop(29)
    Zeta[i].pop(29)
    deltTa[i].pop(28)
    Zeta[i].pop(28)


#########################################################################################################################
##########################################   Figures   ##################################################################
#########################################################################################################################
print(len(Lnew[0]),'L',Lnew)
print(len(q_15[0]),'q_15',q_15)
plt.figure('Power per unit length pipe (coil)')
plt.grid()

plt.plot(Lnew[15], q_15[15], '-', label = '6 m/s')
plt.plot(Lnew[10], q_15[10], '-',color='g', label = '5 m/s')
plt.plot(Lnew[5], q_15[5], '-', label = '4 m/s')
plt.plot(Lnew[2], q_15[2], '-', label = '3,4 m/s')
plt.plot(Lnew[0], q_15[0], '-',color='r', label = '3 m/s')
plt.axhline(y = 41821, linestyle = '--', color = 'black', label = 'Threshold')
plt.ylabel(r'W')
plt.xlabel('m')
plt.legend(fontsize = 16)
plt.tight_layout()
#plt.show()
#plt.close()

###
plt.figure('Pressure drop per unit length pipe (coil)')
plt.grid()

plt.plot(Lnew[15], deltTa[15], '-', label = '6 m/s')
plt.plot(Lnew[10], deltTa[10], '-',color='g', label = '5 m/s')
plt.plot(Lnew[5], deltTa[5], '-', label = '4 m/s')
plt.plot(Lnew[2], deltTa[2], '-', label = '3,4 m/s')
plt.plot(Lnew[0], deltTa[0], '-',color='r', label = '3 m/s')

plt.ylabel(r'bar')
plt.xlabel('m')
plt.legend(fontsize = 16)
plt.tight_layout()
#plt.show()
#plt.close()



###

###
plt.figure('Temperature drop per unit length pipe (coil)')
plt.grid()

plt.plot(L[15], T_range1-273.15, '-', label = '6 m/s')
plt.plot(L[10], T_range1-273.15, '-',color='g', label = '5 m/s')
plt.plot(L[5], T_range1-273.15, '-', label = '4 m/s')
plt.plot(L[2], T_range1-273.15, '-', label = '3,4 m/s')
plt.plot(L[0], T_range1-273.15, '-',color='r', label = '3 m/s')

plt.ylabel(r'($^\circ$C)')
plt.xlabel('m')
plt.legend(fontsize = 16)
plt.tight_layout()
#plt.show()
#plt.close()

###
q_15new=q_15
for i in range(len(q_15)):
    q_15new[i].pop(0)
print('new q15',q_15new)

#Should not get rid of the first one here. but whats up....
for i in range(len(Am2)):
    Am2[i].pop(0)
print('Average Area (coil)',Am2,)
avg=0
summ=0
Tempppp=[]
# Find the average temperature in the entire pipe at the time
```

```python
for i in range(len(T_range)):
    avg =(summ+T_range[i])/(i+1)
    summ=summ+T_range[i]
    Tempppp.append(avg)
print(Tempppp)
#When i now use the average temp i think it might be correct.

#Rtot=1/h*np.pi*d*L
U2 = ([],[],[],[],[],[],[],[],[],[],[],[],[],[],[],[],[],[])
for i in range(len(q_15new)):
    u2=0
    U2[i].append(u2)
    for j in range(len(q_15new[i])):
        u2=q_15new[i][j]/(Am2[i][j]*(Tempppp[j]-273.15))
        U2[i].append(u2)
print('Overall heat transfer coefficient cumulative W/m^2K',U2)

for i in range(len(L)):
    L[i].pop(0)
    U2[i].pop(0)
plt.figure('Overall heat transfer coefficient')
plt.grid()

plt.plot(L[15],U2[15], '-', label = '6 m/s')
plt.plot(L[10],U2[10], '-',color='g', label = '5 m/s')
plt.plot(L[5],U2[5], '-', label = '4 m/s')
plt.plot(L[2],U2[2], '-', label = '3,4 m/s')
plt.plot(L[0],U2[0], '-', label = '3 m/s')


plt.ylabel(r'$W/m^2K$')
plt.xlabel('m')
plt.legend(fontsize = 16)
plt.tight_layout()
#plt.show()
#plt.close()

#Power from pumps
W_pump = ([],[],[],[],[],[],[],[],[],[],[],[],[],[],[],[],[],[])
Ac=np.pi*(d**2/4)
for i in range(len(P_drop_kPa)):
    for j in range(len(P_drop_kPa[i])):
        w_pump = V[i]*Ac*P_drop_kPa[i][j]
        W_pump[i].append(w_pump*1000)

print('Pump power W', W_pump)
for i in range(len(W_pump)):
    W_pump[i].pop(30)
    W_pump[i].pop(29)
    W_pump[i].pop(28)
for i in range(len(L)):
    L[i].insert(0,0)
    q_15[i].insert(0,0)

plt.figure('Required pump power per unit length pipe (coil)')
plt.grid()

plt.plot(L[15], W_pump[15], '-', label = '6 m/s')
plt.plot(L[10], W_pump[10], '-',color='g', label = '5 m/s')
plt.plot(L[5], W_pump[5], '-', label = '4 m/s')
plt.plot(L[2], W_pump[2], '-', label = '3,4 m/s')
plt.plot(L[0], W_pump[0], '-',color='r', label = '3 m/s')

plt.ylabel(r'W')
plt.xlabel('m')
plt.legend(fontsize = 16)
plt.tight_layout()
plt.show()
plt.close()

##########################################################################################################################
################################### Dividing into several pipes #####################################################
##########################################################################################################################
print()
print('Smaller pipes calculations')
print()
#Now i should start looking at if we divide it into two or 3 pipes.
#Finding new diameter of pipe fist
N = 3            #Number of pipes wanted
m_dot_per2 = []
for i in range(len(m_dot)):
    m_dot_per1 = m_dot[i]/N
    m_dot_per2.append(m_dot_per1)
print('Mass flowrate per pipe',m_dot_per2)

d_i_per= []
for i in range(len(m_dot)):
```

```
    d_i = np.sqrt(m_dot_per2[i]/(V[i]*np.pi*rho)*4)
    d_i_per.append(d_i)
print('new inner diameter', d_i_per)
d_o_per = []
for i in range(len(m_dot)):
    d_op = d_i_per[i]+0.002
    d_o_per.append(d_op)



q3=[]
for i in range(len(m_dot_per2)):
    qq3=m_dot_per2[i]*Cp
    q3.append(qq3)
print('Power per 1K temp drop',q3)



Re_d = ([],[],[],[],[],[],[],[],[],[],[],[],[],[],[],[])
for i in range(len(m_dot)):
    for j in range(len(visc)):
        Re_di = (4*m_dot_per2[i])/(np.pi*d_i_per[i]*visc[j])
        Re_d[i].append(Re_di)
print('Reynolds number for temperature range',Re_d)



D=0.3
print('Inner diameter:',d,'bend diameter',D)

Recrit3=2300*(1+8.6*(d_i_per[i]/D)**0.45)
print('critical Reynolds 3',Recrit3)

#Convection of water inside tube
nusslet3 = ([],[],[],[],[],[],[],[],[],[],[],[],[],[],[],[])
for i in range(len(Re_d)):
    for j in range(len(Re_d[i])):
        if Re_d[i][j] < Recrit3:
            nus2 = 3.66+0.08*(1+0.8*(d_i_per[i]/D)**0.9)*Re_d[i][j]**(0.5+0.2903*(d_i_per[i]/D)**0.194)*pr[j]**(1/3)*(pr[j]/pr[j])**0.14
            nusslet3[i].append(nus2)
        elif Recrit3<=Re_d[i][j]<=22000:
            gimmi = (22000-Re_d[i][j])/(22000-Recrit3)
            Nu1 = 3.66+0.08*(1+0.8*(d_i_per[i]/D)**0.9)*Recrit3**(0.5+0.2903*(d_i_per[i]/D)**0.194)*pr[j]**(1/3)*(pr[j]/pr[j])**0.14
            fcrit = ((0.3164 / 22000**0.25) + 0.03 * (d_i_per[i]/ D) ** 0.5) * (visc[j]/ visc[j]) ** 0.27
            Nut = (((fcrit / 8) * 22000 * pr[j]) / (1 + (12.7 * np.sqrt(fcrit / 8)) * (pr[j] ** (2 / 3) - 1))) * (pr[j] / pr[j]) ** 0.14
            nus2 = gimmi*Nu1+(1-gimmi)*Nut                  #0.3 is since the water is cooling
            nusslet3[i].append(nus2)
        else:
            f2 = ((0.3164/Re_d[i][j]**0.25)+0.03*(d_i_per[i]/D)**0.5)*(visc[j]/visc[j])**0.27
            nus2 = (((f2 / 8) * Re_d[i][j] * pr[j]) / (1 + (12.7 * np.sqrt(f2 / 8) ) * (pr[j] ** (2 / 3) - 1)))*(pr[j]/pr[j])**0.14
            nusslet3[i].append(nus2)
print('nusslet nr 3', nusslet3)

h3= ([],[],[],[],[],[],[],[],[],[],[],[],[],[],[],[])
for i in range(len(nusslet3)):
    for j in range(len(k_f_w)):
        hh3=nusslet3[i][j]*k_f_w[j]/d_i_per[i]
        h3[i].append(hh3)
print('Convection coefficient 3',h3)

#BOILING
Ltotcoil = ([10],[10],[10],[10],[10],[10],[10],[10],[10],[10],[10],[10],[10],[10],[10],[10])
eps=10
while eps>0.1:
    T_o = ([], [], [], [], [], [], [], [], [], [], [], [], [], [], [], [])
    for i in range(len(L_section)):
        for j in range(len(L_section[i])):
            Rcond = np.log((d_o_per[i] / 2) / (d_i_per[i] / 2)) / (2 * np.pi * k * L_section[i][j])
            To = T_range[j] - q[i] * Rcond
            T_o[i].append(To)

    qm3_boil = ([], [], [], [], [], [], [], [], [], [], [], [], [], [], [], [])
    for i in range(len(qm3_boil)):
        for j in range(len(T_o[i])):
            qm3 = dynvisc * hfg * ((g * (dens_l - dens_v)) / surf_stresstens) ** (0.5) * (
                        (Cp_l * (T_o[i][j] - T_co2)) / (Csf * hfg * pr_boil ** n_boil)) ** 3
            qm3_boil[i].append(qm3)


    h_boil = ([],[],[],[],[],[],[],[],[],[],[],[],[],[],[],[])
    for i in range(len(h_boil)):
        for j in range(len(qm3_boil[i])):
            hboil=qm3_boil[i][j]/(T_o[i][j]-T_co2)
            h_boil[i].append(hboil)

    k = 14.9
```

```
    Rcond = np.log((d_o_per[i]/2)/(d_i_per[i]/2))/(2*np.pi*k)
    Ltotcoilnew =  ([],[],[],[],[],[],[],[],[],[],[],[],[],[],[],[])
    L_sectionnew = ([],[],[],[],[],[],[],[],[],[],[],[],[],[],[],[])
    Lnew =  ([],[],[],[],[],[],[],[],[],[],[],[],[],[],[],[])

    for i in range(len(h_boil)):
        Le=0
        summ=0
        Lnew[i].append(Le)
        for j in range(len(h_boil[i])):
            Lens=q[i]/(2*np.pi*LMTD[j])*((1/(h3[i][j]*d_i_per[i]/2))+1/(h_boil[i][j]*d_o_per[i]/2)+np.log(d_o_per[i]/
d_i_per[i])/(k))
            L_sectionnew[i].append(Lens)
            Le = Le + Lens
            Lnew[i].append(Le)
            summ = summ + Lens
        Ltotcoilnew[i].append(summ)

    eps = np.abs( Ltotcoilnew[15][0]-Ltotcoil[15][0])
    print(eps)
    L_section = L_sectionnew

    Ltotcoil = Ltotcoilnew


print('Outer temperature', T_o)
print('Convection coefficient boiling', h_boil)
print('convection coefficient',h3,'W/m**2K')
print('heat flux boiling W/m^2', qm3_boil)
print(Lnew)
print('total pipe length',Ltotcoil)
print('Length of each section',L_sectionnew)

q_15s  = ([],[],[],[],[],[],[],[],[],[],[],[],[],[],[],[])
for i in range(len(q_15)):
    cumulat = 0
    q_15s[i].append(cumulat)
    for j in range(len(Lnew[i])-1):
        cumulat = cumulat+q3[i]
        q_15s[i].append(cumulat*N)
print('power',q_15s)

for i in range(len(Lnew)):
    Lnew[i].pop(30)
    Lnew[i].pop(29)
    Lnew[i].pop(28)
   # L[i].pop(27)

for i in range(len(q_15s)):
    q_15s[i].pop(30)
    q_15s[i].pop(29)
    q_15s[i].pop(28)
  #  q_15[i].pop(27)

plt.figure('Power per unit length 2 pipes (coil)')
plt.grid()

plt.plot(Lnew[15], q_15s[15], '-', label = '6 m/s')
plt.plot(Lnew[10], q_15s[10], '-',color='g', label = '5 m/s')
plt.plot(Lnew[5], q_15s[5], '-', label = '4 m/s')
plt.plot(Lnew[2], q_15s[2], '-', label = '3,4 m/s')
plt.plot(Lnew[0], q_15s[0], '-',color='r', label = '3 m/s')
plt.axhline(y = 41821, linestyle = '--', color = 'black', label = 'Threshold')
plt.ylabel(r'W')
plt.xlabel('m')
plt.legend(fontsize = 16)
plt.tight_layout()
plt.show()
plt.close()

#            Pressure drop            #
#            Friction coefficient     #
Zeta=([],[],[],[],[],[],[],[],[],[],[],[],[],[],[],[])
for i in range(len(Re_d)):
    zeta=0
    Zeta[i].append(zeta)
    for j in range(len(Re_d[i])):
        if 1<Re_d[i][j]*np.sqrt(d_i_per[i]/D)<Recrit*np.sqrt(d_i_per[i]/D):
            zeta = (64/Re_d[i][j])*(1+0.033*np.log10(Re_d[i][j]*np.sqrt(d_i_per[i]/D))**4)
            Zeta[i].append(zeta)
        if Recrit<Re_d[i][j]<10**5:
            zeta = (0.3164/(Re_d[i][j]**(0.25)))*(1+0.095*(d_i_per[i]/D)**(0.5)*Re_d[i][j]**(0.25))
            Zeta[i].append(zeta)
        if 10**5<Re_d[i][j]<2*10**6: #This is a straight pipe relation so it might not be correct
            zeta=0.00540+0.3964/(Re_d[i][j]**0.3)
            Zeta[i].append(zeta)
print('Friction factor', Zeta)
for i in range(len(L_section)):
    L_section[i].insert(0,0)
#        Pressure drop        #
```

```python
deltTas =([],[],[],[],[],[],[],[],[],[],[],[],[],[],[],[])
P_drop_kPa= ([],[],[],[],[],[],[],[],[],[],[],[],[],[],[],[])
for i in range(len(Zeta)):
    deltT=0

    for j in range (len(Zeta[i])):
        dP = Zeta[i][j]*(L_sectionnew[i][j]/d_i_per[i])*((rho*V[i]**2)/2)
        deltT = deltT+dP
        deltTas[i].append(deltT / 100000)
        P_drop_kPa[i].append(deltT/1000)

print(len(deltTas[0]),'Pressure drop in bar (small coil per pipe)',deltTas)
print('Pressure drop kPa (small coilper pipe)',P_drop_kPa)

for i in range(len(deltTas)):
    deltTas[i].pop(30)
    deltTas[i].pop(29)
    deltTas[i].pop(28)
   # deltTa[i].pop(27)
#Power from pumps
W_pump = ([],[],[],[],[],[],[],[],[],[],[],[],[],[],[],[])


for i in range(len(P_drop_kPa)):
    for j in range(len(P_drop_kPa[i])):
        Ac = np.pi * (d_i_per[i] ** 2 / 4)
        w_pump = V[i]*Ac*P_drop_kPa[i][j]
        W_pump[i].append(w_pump*1000*N)

print('Pump power W small coils', W_pump)

for i in range(len(W_pump)):
    W_pump[i].pop(30)
    W_pump[i].pop(29)
    W_pump[i].pop(28)
  #  W_pump[i].pop(27)
plt.figure('Pressure drop per unit length for 1 of 2 pipes (coil)')
plt.grid()

plt.plot(Lnew[15], deltTas[15], '-', label = '6 m/s')
plt.plot(Lnew[10], deltTas[10], '-',color='g', label = '5 m/s')
plt.plot(Lnew[5], deltTas[5], '-', label = '4 m/s')
plt.plot(Lnew[2], deltTas[2], '-', label = '3,4 m/s')
plt.plot(Lnew[0], deltTas[0], '-',color='r', label = '3 m/s')

plt.ylabel(r'bar')
plt.xlabel('m')
plt.legend(fontsize = 16)
plt.tight_layout()
#plt.show()
#plt.close()


plt.figure('Required pump power for 2 pipes (coil)')
plt.grid()

plt.plot(Lnew[15], W_pump[15], '-', label = ' 6 m/s')
plt.plot(Lnew[10], W_pump[10], '-',color='g', label = '5 m/s')
plt.plot(Lnew[5], W_pump[5], '-', label = '4 m/s')
plt.plot(Lnew[2], W_pump[2], '-', label = '3,4 m/s')
plt.plot(Lnew[0], W_pump[0], '-',color='r', label = '3 m/s')

plt.ylabel(r'W')
plt.xlabel('m')
plt.legend(fontsize = 16)
plt.tight_layout()
plt.show()
#plt.close()
```

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