Denisse Adriana Gonzalez Carrillo

Optimizing the heating and cooling system at Teknostallen

Master's thesis in Sustainable Energy Heat Pumping Processes and Systems
Supervisor: Armin Hafner
Co-supervisor: Erik Hoksrød
June 2023

Master's thesis

NTNU
Norwegian University of Science and Technology
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Preface

This thesis forms part of the international master program of Sustainable Energy Heat Pumping processes and systems at the Norwegian University of Science and Technology in the Energy and Process Engineering Department. The work presented in this document is a continuation of the specialization project and it comprises 30 ECTS credits. The specialization project began in fall 2022 and it was done with Sanaz Naemi, colleague from the master program. Due do this, a number of figures, graphics and overall information on the system will be alike with her work and the work delivered in fall 2022 as specialization project. Additonally to this, several sources were consulted and referenced for the making of this thesis.

Abstract

To reduce the CO_2 emissions in the environment, there has been an effort to implement diverse energy sources to meet the energy demand. Since the majority of renewable energy sources can produce electricity, a trend of electrification has been created. This trend, although it encourages the use of renewable energy, it increases electricity prices due to the augmentation of its demand. A viable option to diminish electricity consumption in a building, to diminish its CO_2 emissions, and use different and available heat sources is to utilize a heat pump unit to supply the energy needed for domestic hot water (DHW) and space heating, among other uses. Heat pumps can obtain energy from different sources and when they use natural working fluids, such as propane and CO_2, the systems have a low environmental impact. This project is focused on the building project of Teknobyen, a complex of office buildings located in Trondheim, Norway. An air source heat pump unit of propane and ethylene glycol mixture is proposed as well as a CO_2 heat pump unit for domestic hot water to provide the heating requirements of three of their buildings.

Models were made in Dymola Modelica to simulate the behavior of the propane heat pump and the overall ethylene glycol system, which resembles the systems proposed to be installed. This system aims to provide the heating and cooling requirements of the Profesor Brochs, Teknostallen, and Abels gate 5 buildings. The propane model was made to simulate a winter scenario while the ethylene glycol model was developed to evaluate the possible performance of the overall system under two winter scenarios. A comparison of the models and the existing systems was made as well as an analysis of the relevant parameters to improve their performance, a discussion on the validity of the models, and the possible future work for them. Additionally, a tank sizing estimation was made for the defrosting system of the air source heat pump unit with a comparison to the actual tank to be implemented.

Sammendrag

For å redusere CO_2 utslippet til atmosfæren har det vært forsøkt å implementere ulike energikilder for å møte energibehovet. Siden flertallet av fornybare energikilder kan produsere elektrisitet, har det blitt skapt en trend med elektrifisering. Denne trenden, selv om den oppmuntrer til bruk av fornybar energi, øker strømprisene på grunn av økningen i etterspørselen. Et levedyktig alternativ for å redusere elektrisitetsforbruket i en bygning, redusere CO_2-utslippene og bruke forskjellige og tilgjengelige varmekilder, er å bruke en varmepumpeenhet for å levere energien som trengs Til blant annet varmtvann og husholdningsbruk (DHW) og romoppvarming. Varmepumper kan hente energi fra ulike kilder og når de bruker naturlige arbeidsvæsker, som propan og CO_2, har systemene lav miljøpåvirkning. I dette prosjektet er det fokusert på byggeprosjektet til Teknobyen, et kompleks av kontorbygg lokalisert i Trondheim, Norge. En luftkildevarmepumpeenhet av propan og etylenglykolblanding er foreslått samt en CO_2-varmepumpeenhet for varmtvann til husholdningsbruk for å dekke varmebehovet til tre av deres bygninger.

Modeller ble laget i Dymola Modelica for å simulere oppførselen til propanvarmepumpen og det totale etylenglykolsystemet, som ligner systemene som er foreslått installert. Dette systemet tar sikte på å dekke kravene til oppvarming og kjøling til bygningene Profesor Brochs, Teknostallen og Abels gate 5. Propanmodellen ble laget for å simulere et vinterscenario mens etylenglykolmodellen ble utviklet for å evaluere den mulige ytelsen til det totale systemet under to vinterscenarier. Det ble foretatt en sammenligning av modellene og de eksisterende systemene, samt en analyse av de relevante parameterne for å forbedre ytelsen, en diskusjon om modellenes gyldighet og mulig fremtidig arbeid for dem. I tillegg ble det gjort en estimering av tankdimensjonering for avrimingssystemet til luftvarmepumpeenheten med en sammenligning med den faktiske tanken som skal implementeres.

Acknowledgements

First of all, I want to thank my supervisor Armin Hafner and co-supervisor Erik Hoksrød for their help and guidance on this project. Special thanks to Lukas Köster for his assistance and support with Dymola Modelica through this project. Moreover, thanks to Sanaz Naemi for the collaboration during the specialization project.

I would also like to thank my colleagues from my master's program, Marina, Victoria, Marthine, Lukas, Even, Mohammad, and Marco for their support and friendship throughout these two years. Additionally, I would like to express my gratitude to my friends back home, Atenea, Marlhene, Carolina, Alejandra, Marel, Sofía, Gabriela, Ana Sofía, Gilberto, Raúl, and Mauro for their friendship and encouraging words during this period.

Finally, I would like to thank my family. Mamá, Karen, and papá, thank you so much for your help, support, and encouraging words. Thank you for supporting me in the best way you can and encouraging me to keep on studying and following my dreams. Thanks to my extended family, tía Roselia, tía Teresa, Marla, Verónica, David, Fátima y Violeta for their support from afar, I am very grateful to have you in my life.

Dedication

The work presented in this document is dedicated to my parents, Olivia and Sergio, to my sister Karen, to my grandmas Elva and Adelina and to Darwin and Emmy. Thank you for being there for me and believing in me, I love you very much.

Contents

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Nomenclature

Abbreviations

$AG5$ Abels gate 5

$AG9$ Abelshus

$ASHP$ Air Source Heat Pump

$CFCs$ Chlorofluorocarbons

CFD Computational Fluid Dynamics

CO_2 Carbon Dioxide

COP Coefficient of Performance

DHW Domestic Hot water

$ECTS$ European Credit Transfer and Accumulation System

EER Energy Efficiency Ratio

$GSHP$ Ground Source Heat Pump

GWP Global Warming Potential

HE Heat Exchanger

$HSPF$ Heating season Performance Factor

KLP Kommunal Landspensjonskasse,

$NTNU$ Norges teknisk-naturvitenskapelige universitet

ODP Ozone Depletion Potential

PID Piping and Instrumentation Diagram

$PB2$ Professor Brochs gate 2

PER Primary Energy Ratio

PI Proportional Integral

TST Teknostall

Greek symbols

ρ Density (kg/m^3)

Latin symbols

$\dot{Q}$ Heat power (kW)

$\dot{V}$ Air flow rate (m^3/h)

$\dot{W}$ Work (kW)

A Area (m^2)

T Time (s)

CHAPTER 1

Introduction

1.1 Project description and objective

Teknobyen is a multipurpose complex composed of modern neighboring buildings located between NTNU Gløshaugen and St. Olavs Hospital in Trondheim, Norway. This area has experienced several changes since its initial construction in 1923, the latest being the construction of the Teknostall building. Teknobyen consists mainly of four buildings; Teknostall, Professor Brochs gate 2, Abels gate 5, and Abelshus, located in the previous Trondheim railway building. These buildings are owned and operated by KLP Eiendom Trondheim and are designed to host conferences, and cultural events and provide flexible office spaces and common areas for small and large companies.

Additionally, a new energy center will be designed by KELVIN AS to meet the heating and cooling requirements of the Teknostall, Professor Brochs gate 2, and Abels gate 5 buildings. This energy central will consist of several heat exchangers, dry coolers, and a propane and CO2 heat pumping units to provide the necessary hot water required by the buildings and their users.

The system will work mainly as a cooling device in the summer and as a heat pump during the winter. For the rest of the year, it will work as both a cooling and heating device. The unit will heat tap water and cool data servers throughout the entire year as well as cool the ventilation system in the summer. In winter the unit will mainly provide heat for the ventilation system and floor heating. The main objective of this thesis is to develop an energy flow tool in Dymola Modelica to estimate the performance of the system in different conditions or scenarios and analyze and discuss these results.

1.2 Energy requirements and sources

An analysis was made on the energy consumption from the Professor Brochs gate 2, Abelshus, and Abels gate 5 buildings to know the energy requirements of the existing buildings and to estimate the dimensions of the heat pump systems. From this, it could be observed that the energy demand for Abelshus is considerably higher than Abels gate 5 and Professor Brochs 2. On that account, it was decided that the new energy center should include only Professor Brochs 2, Abels gate 5, and the new Teknostall building. The energy requirements of the Teknostall building are estimated to be the DHW from the Abelshus building and the heating load requirements from the Statkraft calculator. Additionally, since most of the energy for these buildings is provided by a district heating supplier, it is essential that they are as energy efficient as possible without compromising the comfort of the users. To achieve this, the implementation of a propane heat pump and a CO_2 heat pump will be proposed.

1.3 Outline

The following text is organized as follows:

Chapter 2 Heat pump systems and refrigerants: Definition and description of heat pumping systems for cooling, heating of buildings, components of these systems, categories of heat pumps, defrosting systems, working fluids, and propane and CO_2 as refrigerants.

Chapter 3 System description: Description of the overall system to be installed for the cooling and heating of the buildings.

Chapter 4 Defrosting and tank sizing: Explanation of the defrosting methods, their relevance to this project, frost conditions, frost growth rate, and tank sizing for the defrosting unit.

Chapter 5 System simulation: Description of the software used for the simulations and data visualization, description of the different models with their input data information.

Chapter 6 Results: Presentation of the results obtained from the simulations.

Chapter 7 Discussion: Interpretation and discussion of the results.

Chapter 8 Conclusion and further work: Conclusions from the results obtained and possible further work for the project.

Appendix A Weather data analysis

Appendix B Simulation Parameters

CHAPTER 2

Heat pump systems and refrigerants

This chapter will cover the overall principle of a heat pump unit, its components, the fundamentals behind an air source heat pump, heat pumps for cooling and heating of buildings, and the working fluids available. This aims to provide the fundamentals to understand the heat pump system proposed for the buildings in Teknobyen, which will be further described in Chapter 3. This chapter presents a compilation of the literature review introduced in the specialization project work.

2.1 Heat pump systems and components

Heat pumps are devices that transfer heat from a low-temperature area to a higher-temperature area. Since the heat flows naturally from a higher to a lower temperature, heat pumps need high-quality energy, such as electricity, to transfer the heat in the opposite direction [1]. There are four basic types of heat pumps: water-to-water, water-to-air, ground-to-air and air-to-air, the first being the source of heat and the latter the medium treated by the refrigerant [2]. Air source heat pumps are relevant to this project, therefore, further explanation of this type of heat pump will be given in this chapter.

The main components of a heat pumping system are the compressor, condenser, throttling or expansion device, such as an expansion valve, and the evaporator. The selection of these components will be influenced by the type of system, the system's efficiency and maintainability, the refrigerant used, and whether the system requires a defrosting system, among others[3].

The compressor has the main function to increase the pressure and temperature of the refrigerant, in a gaseous state, until it reaches the condenser pressure. The compressor requires work applied to it to complete its function, this is usually supplied by an electric motor. The compressor is a very relevant part of the process since it determines the capacity of the system.

The condenser is a heat exchanger where the refrigerant releases the energy gained from the compressor and evaporator and transfers it to another fluid. This is done by the phase change of the refrigerant from gas to liquid [3]. The relevant factors to consider when choosing the appropriate condenser are the condensing temperature and pressure, the flow rate of the refrigerant, their operation period, weather conditions, and availability of water and electricity. Yet condensers are commonly seen as large plate heat exchangers [1].

The expansion device is typically an expansion valve that lowers the refrigerant's condensing pressure (high pressure) to the evaporating pressure (low pressure) and regulates the refrigerant flow to the evaporator depending on the load [1].

The evaporator is generally a plate heat exchanger that warms up the refrigerant and produces a phase change from liquid to gas. This heat absorbed will be released in the condenser on the next cycle [3]. Figure 2.1 shows a heat pumping system with its main components and in Figure 2.2 the pressure enthalpy diagram of a heat pumping cycle can be observed.

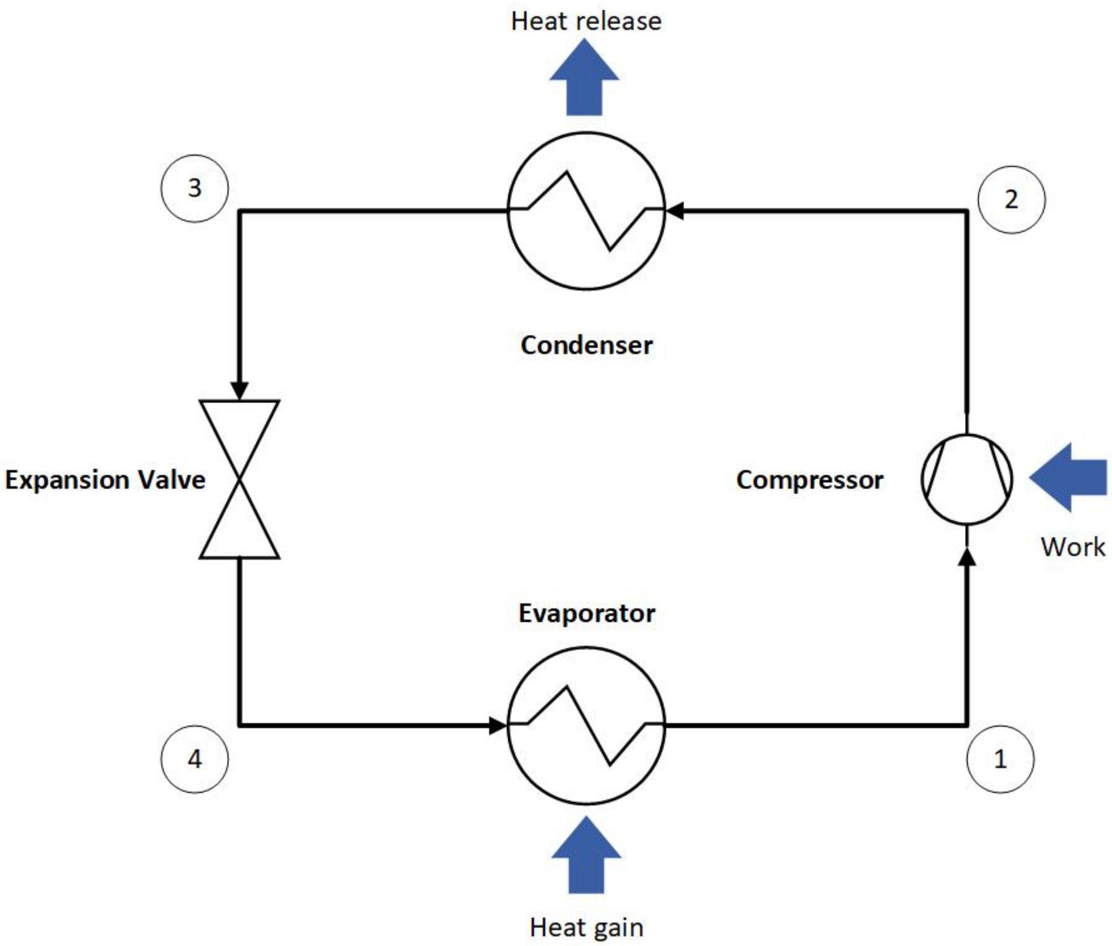


Figure 2.1: *Heat pumping cycle with main components.*

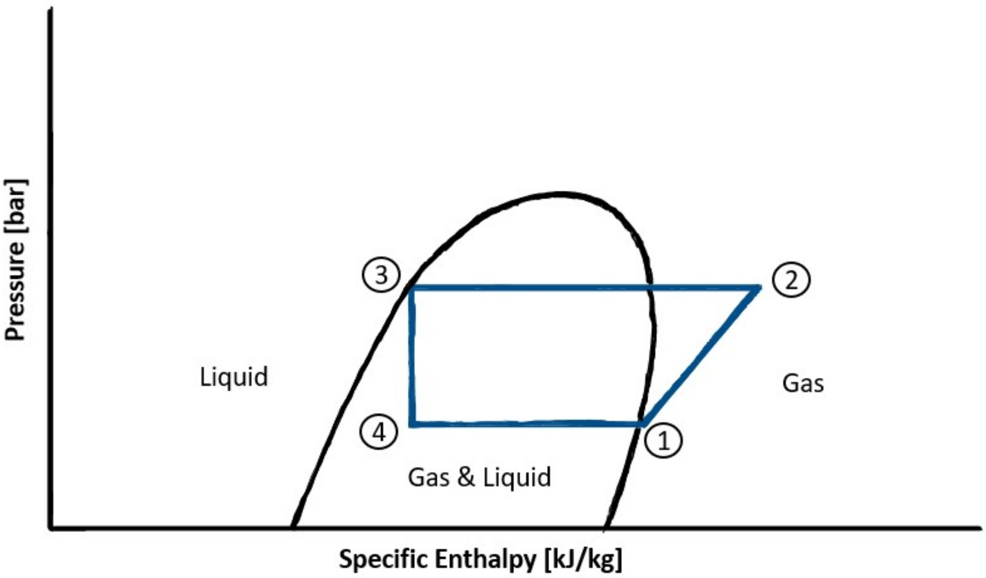


Figure 2.2: *Pressure-Enthalpy diagram of a heat pumping cycle.*

2.2 Air source heat pumps

Heat pumps can gain heat from the cold air since heat is still present in mediums down to a temperature of -273.15°C or absolute zero [2]. Air source heat pumps are one of the most used options due to their economical installment and maintenance costs [4]. Nevertheless, two major downsides to the air source heat pumps are that the temperature level of the heat source is lower compared to other types of heat pumps and that the temperature varies greatly depending on the season [5]. Therefore, this type of heat pump requires a defrost system to prevent frost formation on the equipment since this ice adds an extra insulation layer and diminishes the heat transfer. This affects the efficiency of the system, hence, the frost must be removed. The conditions for frost growth and the different defrosting methods will be discussed further on Chapter 4.

2.3 Heat pumps efficiency evaluation

Heat pump efficiency is determined by the amount of energy consumed by the heat pump and the amount of energy provided by it. There are four principal parameters to evaluate the efficiency of a heat pump: Coefficient of Performance (COP), Primary Energy Ratio (PER), Energy Efficiency Ratio (EER), and the Heating Season Performance Factor (HSPF) [1]. The most common one is the Coefficient of Performance, hence, this parameter will be used to evaluate the system of interest for this project. The COP is calculated as the output heat of the heat pump over the electrical energy input as stated below [1]:

$$COP = \frac{\text{Heat output}}{\text{electrical energy input}} \tag{2.1}$$

The heat output will be the refrigeration capacity or heating capacity, depending if the system is in cooling or heating mode and the electrical energy input is the work done on the compressor and the auxiliaries such as fans, pumps, etc. This gives as an outcome the following equations for the coefficient of performance for cooling and heating.

$$COP_{\text{cooling}} = \frac{\dot{\text{Q}}_{\text{cooling}}}{\dot{\text{W}}_{\text{compressor}} + \dot{\text{W}}_{\text{auxiliaries}}} \tag{2.2}$$

$$COP_{\text{heating}} = \frac{\dot{\text{Q}}_{\text{heating}}}{\dot{\text{W}}_{\text{compressor}} + \dot{\text{W}}_{\text{auxiliaries}}} \tag{2.3}$$

The higher the COP the better the system is because it provides more energy than the one it is consuming. Ground source heat pumps (GSHPs) and water source heat pumps usually have a COP between 3 and 5 while air source heat pumps have a COP between 2 and 4 [1].

2.4 Working Fluids

The selection of the working fluid is influenced by the application of the heat pump, the thermodynamic properties, physical and chemical properties, environmental impact, price, and availability of the working fluid [6]. Refrigerants can be classified as halocarbons, hydrocarbons, inorganic compounds, azeotropic mixtures, and non azeotropic mixtures [1]. The following sections describe the two working fluids that are of interest for this project.

2.5 Propane

Propane belongs to the hydrocarbons category of refrigerants, which is characterized by its low Global Warming Potential (GWP), no Ozone Depletion Potential (ODP), and high energy efficiency. It is non-corrosive and it has good oil compatibility, yet it is flammable [7]. More properties on propane can be found in Table 2.1. One of the principal characteristics of propane is that it is an attractive substitute for Chlorofluorocarbons (CFCs) in domestic refrigeration systems. This is because it has a similar performance with a lower charge and with minor to no changes in the design [8]. It is known as a common refrigerant in heat pumps, air conditioners, and commercial refrigeration systems and it has been in use since the 1930's [9].

Table 2.1: *Propane properties*

Properties	Propane R290
Molecular Weight [kg/kmol]	44.097
Critical Temperature [°C]	97
Critical Pressure [bar]	43
Ozone Depleition Potential (ODP)	0
Global Warming Potential (GWP)	3
Flammability	Highly flammable

A relevant disadvantage of propane is that, due to its flammability, special attention has to be paid during the installation and maintenance of the propane systems. Additionally, the flammability of this refrigerant limits the amount of propane you can have in the system depending on its location, type of room, and the occupancy of the area where it is installed [7].

2.6 Carbon dioxide

Carbon dioxide belongs to the inorganic refrigerants group and it is a colorless, odorless, non-toxic, non-flammable, and nonexplosive refrigerant [1]. It has a low cost and is reasonably available, as well as having zero ODP, a GWP of 1, and being thermally stable. It has good compatibility with normal lubricants and common construction materials for the devices. Furthermore, liquid carbon dioxide's properties are favorable for convective heat transfer in heat exchangers. It has a fairly good thermal conductivity, a low viscosity, and a high specific heat capacity. The benefit of having low viscosity is that this causes minimum friction pressure loss [7]. More properties of carbon dioxide can be found in Table 2.2.

Table 2.2: *CO_2 properties*

Properties	**CO_2 R744**
Molecular Weight [kg/kmol]	44
Critical Temperature [°C]	31.1
Critical Pressure [bar]	73.8
Ozone Depletion Potential (ODP)	0
Global Warming Potential (GWP)	1
Flammability	Not flammable

A downside of this working fluid is that it requires high pressures, therefore, in case of a leak, the loss of refrigerant can be high. Furthermore, its triple point pressure is higher than ambient and consequently, it requires the application of certain safety rules that are not necessary for other working fluids[7].

Since the critical temperature of carbon dioxide is relatively low, these heat pumps are usually operated in transcritical mode because the space heating applications require a higher temperature level than the one obtained in a subcritical CO_2 cycle. For this reason, the CO_2 heat pump equipment should be particularly designed for this purpose and will have some differences compared to the usual cycle with other refrigerants. One of the major changes is that the condenser will be replaced with a gas cooler since the carbon dioxide will not reach condensation conditions and will release its heat through sensible heat instead of a phase change. Figure 2.3 shows both the subcritical and transcritical cycles. One of the best and more common applications for CO_2 heat pumps is the production of DHW since it can achieve water temperatures of 85 to 95°C and has a higher COP compared to regular hot water heat pumps [10].

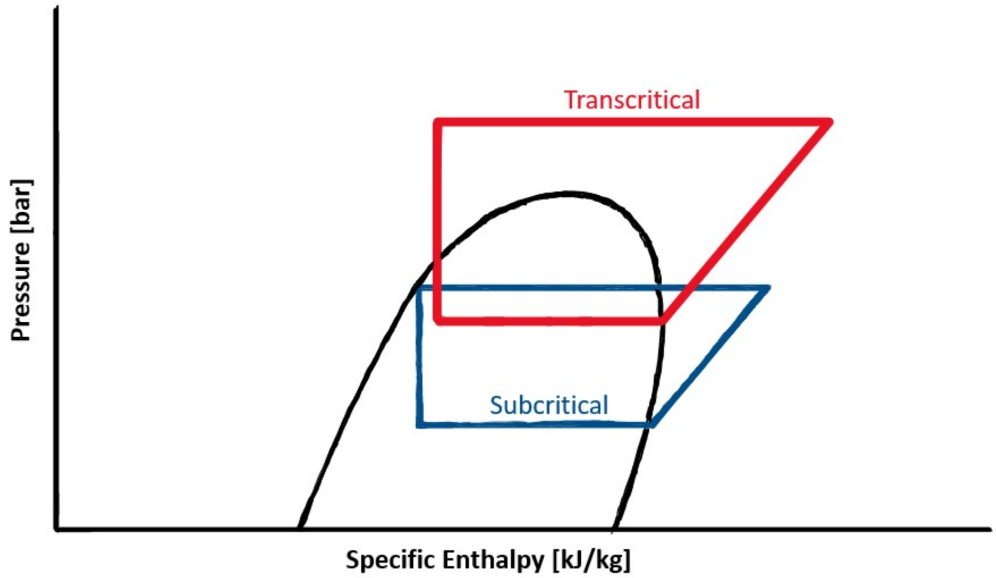


Figure 2.3: *Subcritical and transcritical CO_2 cycles within the pressure-enthalpy diagram.*

2.7 Heat pumps for cooling and heating of buildings

In traditional refrigeration systems, the heat rejected in the condenser is wasted, nevertheless, this energy can be used to heat water for domestic purposes with a simultaneous heating and cooling system [11]. Heat pumps for cooling and heating of a building operate in three principal modes:

- Heating mode; to produce hot water from the heat in ambient air
- Cooling mode; to produce cold water by rejecting heat to the ambient air
- Simultaneous mode; to produce hot water from the heat removed from the cold water, making the latter even colder.

It is highly recommended for the heat pump to run in a simultaneous mode as often as possible to provide both cooling and heating [12]. Heating and cooling heat pumps for buildings can be divided mainly into four categories [13]:

- Heating-only heat pumps for space heating and/or water heating
- Heating and cooling heat pumps for space heating and cooling
- Integrated heat pumps that supply space heating, cooling, water heating, and exhaust air heat recovery
- Heat pump water heaters which only provide heating for water

The heat pump system proposed for this project is an integrated air source heat pump that supplies space heating, cooling, and water heating from the ambient air heat to heat a 37% mix of ethylene glycol and water that is connected to a propane heat pump and a CO_2 heat pump. Additional description of the system will be provided in Chapter 3.

CHAPTER 3

System description

The following chapter will describe the heat pumping systems proposed to deliver the heating requirements for the Teknostall, Abels gate 5, and Professor Brochs gate 2 buildings. In order to provide this heat, an air-source heat pump with propane as working fluid will be used for space heating and ventilation of the buildings, and a CO_2 heat pump is proposed for the domestic hot water demand.

3.1 Air source heat pump

The heating and cooling demands of the buildings will be covered mainly by an air-source heat pump (ASHP). This ASHP unit extracts heat from air at a lower temperature through four dry coolers that will be installed on the roof of AG5. Additionally, the ethylene glycol stream recovers the heat from the buildings and the data center cooling system. Then, the ethylene glycol mixture will warm up the propane and the CO_2 through the heat pump systems for space heating and DHW systems, respectively.

The propane cycle warms the water cycles that heat the floor and ventilation heating for the Teknostall building, the space heating and DHW for PB2, the DHW and space heating for AG5, and the defrosting of the dry coolers. An overall panorama of this process can be seen in the Piping and Instrumentation Diagram (P&ID) in Figure 3.1.

The system will operate in two modes: cooling and heating, depending on the weather conditions and the requirements of the buildings. The temperature of the ethylene glycol mixture in the system will not be the same in these two modes. Since in winter, the air temperature is lower, the glycol temperature has to be even lower to be able to obtain heat from the air flowing through the dry coolers. Meanwhile, in cooling mode, the dry coolers will obtain ethylene glycol only through the stream that is connected to the heat exchanger number 11 in Figure 3.1. During the heating mode, the ethylene glycol flow to the dry coolers is done through the other stream, and the warm ethylene glycol from the heat exchanger 11 is only used as a defrosting system when necessary.

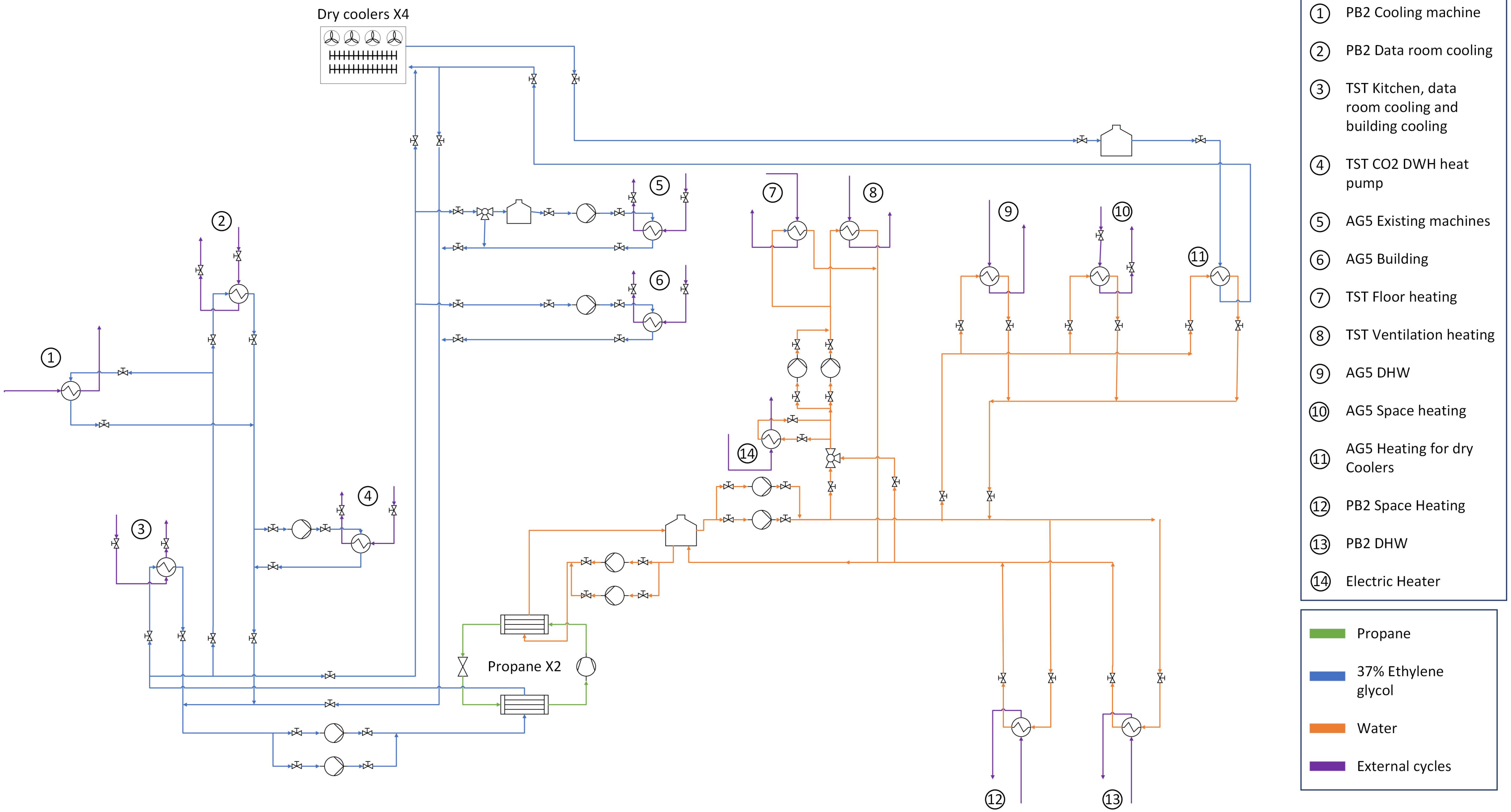


Figure 3.1: *Proposed process diagram for Teknobyen*

3.2 Propane heat pump

The propane system is composed of two heat pumps with four circuits each. Every circuit includes an evaporator, compressor, condenser, and expansion valve. As mentioned before, this heat pump warms the water that runs through the different buildings for heating purposes. The water temperature before and after the condenser varies according to the cooling or heating mode in which the system is set. Additionally, the water circuit has storage tanks in certain parts of the system for better control of its temperature and it has the possibility of being further heated up by an electric heater and district heating if necessary.

3.3 CO_2 heat pump

The tap water in the PB2 and AG5 buildings is preheated via the building heating circuit in each building, while the tap water in the Teknostall building is heated by the CO_2 heat pump in the power plant. The Teknostall building accounts for the largest share of hot water consumption, therefore, there will be a lot of savings when the heat pump is used for this purpose. The technical data of this heat pump can be found in Table 3.1.

Table 3.1: *Technical data of CO_2 DHW heat pump*

Parameter	**Value**
Heat Power [kW]	50.4
Evaporator [kW]	38.7
Electric Power [kW]	11.7
DHW [Liter/hour]	787.8

The P&ID of system is shown in Figure 3.2. This system includes a compressor, separator, expansion valve, and 3 plate heat exchangers that are working as the evaporator, internal heat exchanger, and gas cooler.

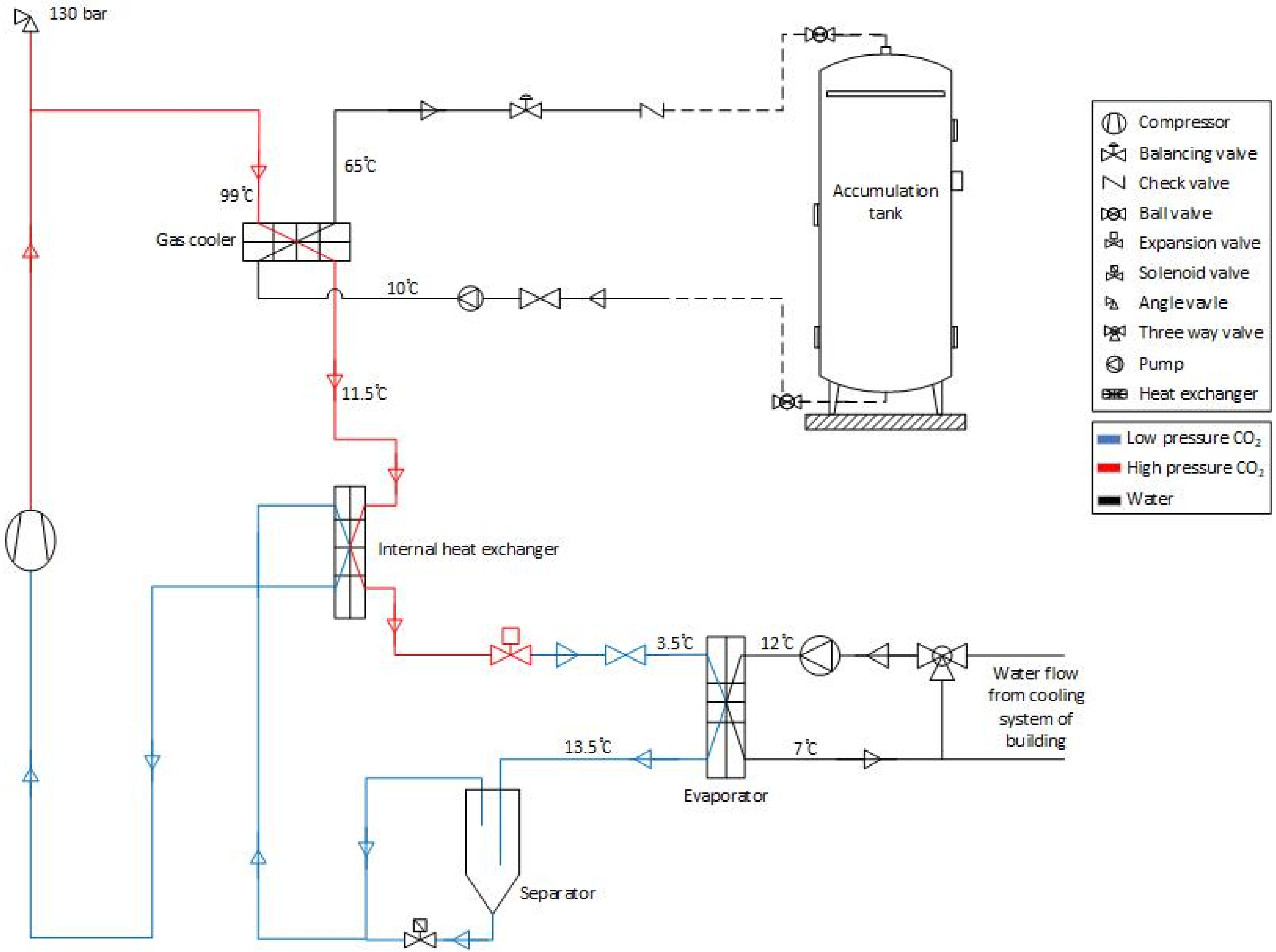


Figure 3.2: *Proposed process diagram for DHW*

CHAPTER 4

Defrosting and tank sizing

The following chapter will explain the relevance of the defrosting system for an ASHP, what are the frosting conditions, the defrosting methods, a weather data analysis on Trondheim, the frost growth rate, the estimated frost melting rate, and proposed tank sizing. This work began in fall 2022 with the specialization project and was continued with more weather data conditions.

4.1 Frosting condition

Frost can be formed and accumulated on the outdoor coils of the dry coolers when the outdoor weather conditions where the ASHP operates are low temperatures and high humidity. This limits the flowing air through the coils and decreases the heat exchange, reducing the COP and the heating capacity of the system [14]. The frost layer behaves as an additional insulating resistance that diminishes the heat transfer [15]. Therefore, defrosting systems must be implemented to avoid this problem and guarantee an efficient operation of the ASHP. The frost growth requires the two following conditions to occur [14]:

- The heat exchanger must have a surface temperature lower than 0°C.
- The heat exchanger's surface temperature must be lower than the ambient air's dew-point temperature.

The surface temperature varies according to the temperature difference between the refrigerant's evaporating temperature, the 37% ethylene glycol mixture in this case, and the ambient air temperature. The dew-point temperature of the air is dependent on the relative humidity, therefore, temperature and relative humidity of air are the main factors to determine whether the surface presents frost or not.

According to research, the conditions where the frost growth is more prevalent are when the air temperature is between -5°C and 5°C and the relative humidity is higher than 70%. When the air temperature is lower than -5°C, the moisture content in the air will be very low, approximately 2 to 3 grams of water per kilogram of air, and regardless of the humidity level, this will not result in a strong frosting level [14].

Meanwhile, when the air temperature is higher than 5°C, the frost can not be formed. In addition, if the air temperature is lower than 5°C with a relative humidity lower than 67%, there will be no frosting because the surface temperature of the heat exchanger is higher than the ambient air's dew-point temperature [14].

4.2 Weather data analysis

For the purpose of determining the relevance of an adequate defrosting system in an ASHP located in Trondheim Norway, a weather data analysis was made with data from the Norwegian Meteorological Institute and the Norwegian Broadcasting Corporation [16]. As mentioned before, frost growth can be present at temperatures between 5°C to -5°C and relative humidity higher than 70% [14]. Therefore, it is of interest to know how many hours per year approximately these weather conditions are met.

The Norwegian Meteorological Institute runs several weather stations around Norway. The station Trondheim - Voll with code SN68860 was chosen as a reference to take these weather data conditions due to its proximity to the area where the project takes place and the availability of the variables measured in that station. The three main variables relevant to this analysis were the minimum temperature, maximum temperature, and relative humidity obtained on an hourly basis from January 1st, 2022 until December 31st, 2022. With this information, an average temperature was obtained for each hour and according to this new temperature and the relative humidity it was examined which hours could potentially present frost conditions.

Overall, a total of 2,353 hours of the year will present frost conditions, this is equivalent to 98 days or 3.25 months. It was found that between September and June, there may be some level of frost growth. The most severe cases are the months of January, February, April, and December as shown in Figure 4.1. Additionally, it was realized that even though the temperature usually rises as the morning goes by, in many cases this temperature increase was not enough to surpass the 5°C, meaning the ASHP could be working continuously under frosting conditions for several hours or even days. Additional graphs and material about the weather analysis can be found in Appendix A.

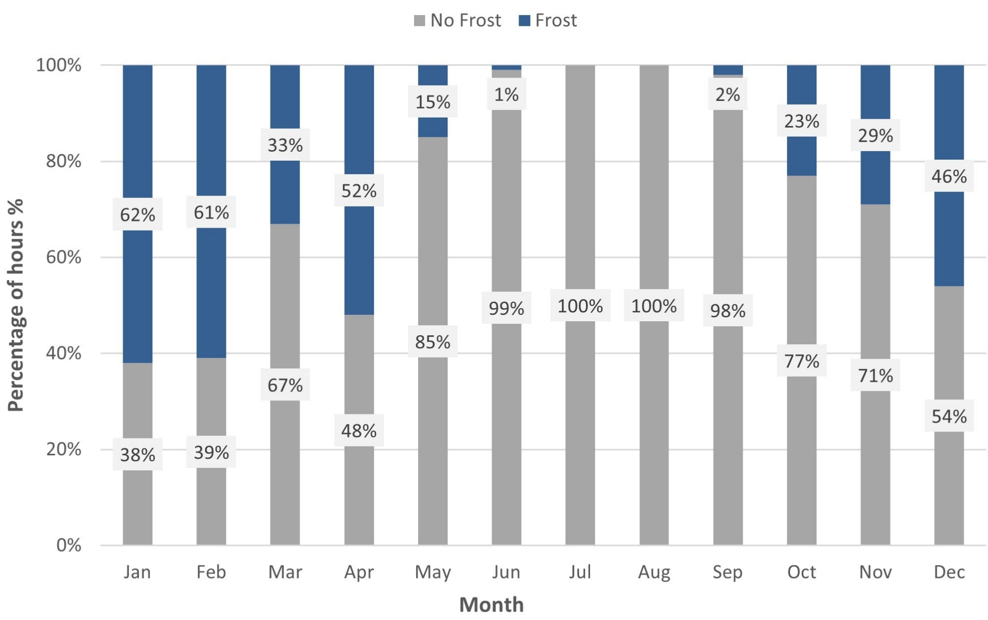


Figure 4.1: *Percentage of hours with frost conditions.*

4.3 Defrosting system

There are different methods to control frost formation, its two main categories being [17]:

- Passive methods: Uses surface treatments to reduce frost growth without additional energy consumption.
- Active methods: Removes the frost periodically with energy consumption.

For this project, an accumulator tank with warm media will be used as a defrosting system, an example of an active defrost method. The media for the defrost system is a 37% ethylene glycol mixture which warms up with the excess heat of the buildings. Afterward, it is stored in the accumulation tank which serves as the thermal storage.

Figure 4.2 shows a general overview of the defrost system. As explained in chapter 3, there are four dry cooler units with a 600 kW capacity each. The capacity of the defrosting system is 21 liter/s and the difference between the inlet and outlet temperature of the 37% ethylene glycol is assumed to be 23°C. The heat capacity of the system is estimated to be 1,834.6 kW. The space between the fins of the dry coolers is 4 mm, therefore frost should be melted before it blocks this space, giving a maximum frost layer thickness of 2 mm on the fins.

The defrosting of dry cooler units will be done individually, not simultaneously, with two accumulator tanks installed to store the energy needed to defrost the coils. The most important variables of the defrosting system are listed in Table 4.1. In order to estimate the accumulator tanks' size, the frost growth and melting rate should be estimated for the conditions of the project.

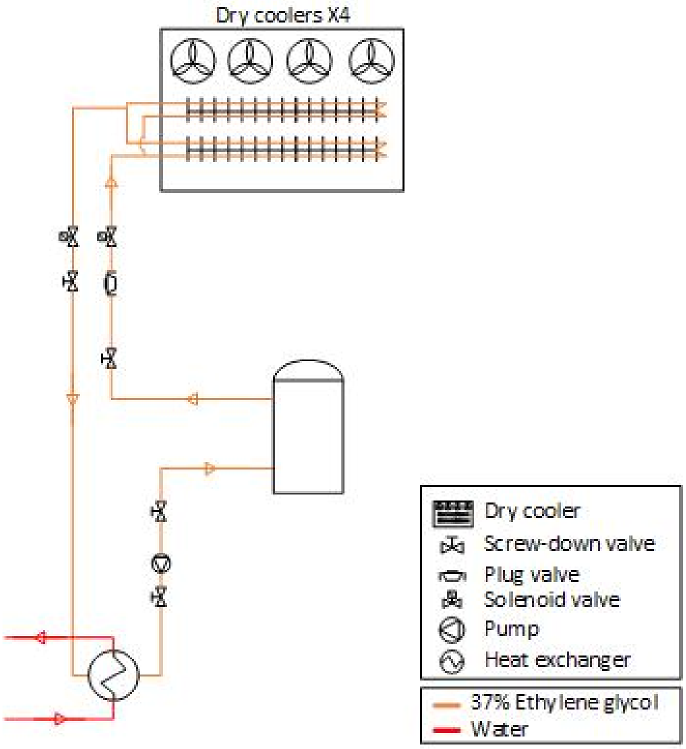


Figure 4.2: *Defrosting system of dry coolers*

Table 4.1: *Input data used to estimate the frost growth rate.*

Parameter	Value
Dry Cooler Unit	
Capacity [kW]	600
Surface area [m^2]	3,451
Fin spacing [mm]	4.0
Airflow [m^3/h]	255,336
Air velocity [m/s]	1.5
Air Density [kg/m^3]	1.2
Frost	
Frost Density [kg/m^3]	1000
Latent heat of ice melting at 0°C [kJ/kg]	334
Defrosting System	
Volume capacity [Liter/s]	21
Inlet/outlet temperature difference [°C]	23
Length of pipes [m]	80
Inner diameter [mm]	211.1

4.4 Frost growth rate

For the frost growth rate, a study done by Long Zhang and his colleagues was reviewed due to the similarities in its conditions to this project [18]. They developed a numerical model to predict frost growth and validated it with experimental data from the literature. For this numerical study, the air temperature and air moisture content are assumed to be 2°C and 3.74 g/kg respectively. The result shows that when the air velocity is about 1.5 m/s, the frost growth rate is approximately 0.0004 mm/s [18].

Additionally, a psychometric chart is used to determine the frost growth rate. It is assumed that the air inlet and outlet temperatures are 3 and -2°C respectively and relative humidity is 80%. Using the Mollier Sketcher software [19], the inlet and outlet air moisture content is found to be 3.75 and 2.55 g/kg respectively, as shown in Figure 4.3. Then, the frost growth rate is calculated with the following equation, resulting in a frost growth rate of approximately 0.00003 mm/s:

$$frost\ growth\ rate = \frac{(\mathrm{X_{in}} - \mathrm{X_{out}}) \cdot \rho_{\mathrm{air}} \cdot \dot{\mathrm{V}}_{\mathrm{air}}}{\rho_{\mathrm{ice}} \cdot \mathrm{A_{HE}} \cdot 3600} \tag{4.1}$$

where:

- $\mathrm{X_{in}}$: inlet air moisture content (g/kg);
- $\mathrm{X_{out}}$: outlet air moisture content (g/kg);
- ρ_{air}: average density of air (kg/m^3);
- $\dot{\mathrm{V}}_{\mathrm{air}}$: air flow rate (m^3/h);
- ρ_{ice}: average density of ice (kg/m^3);
- $\mathrm{A_{HE}}$: area of heat exchanger (m^2);

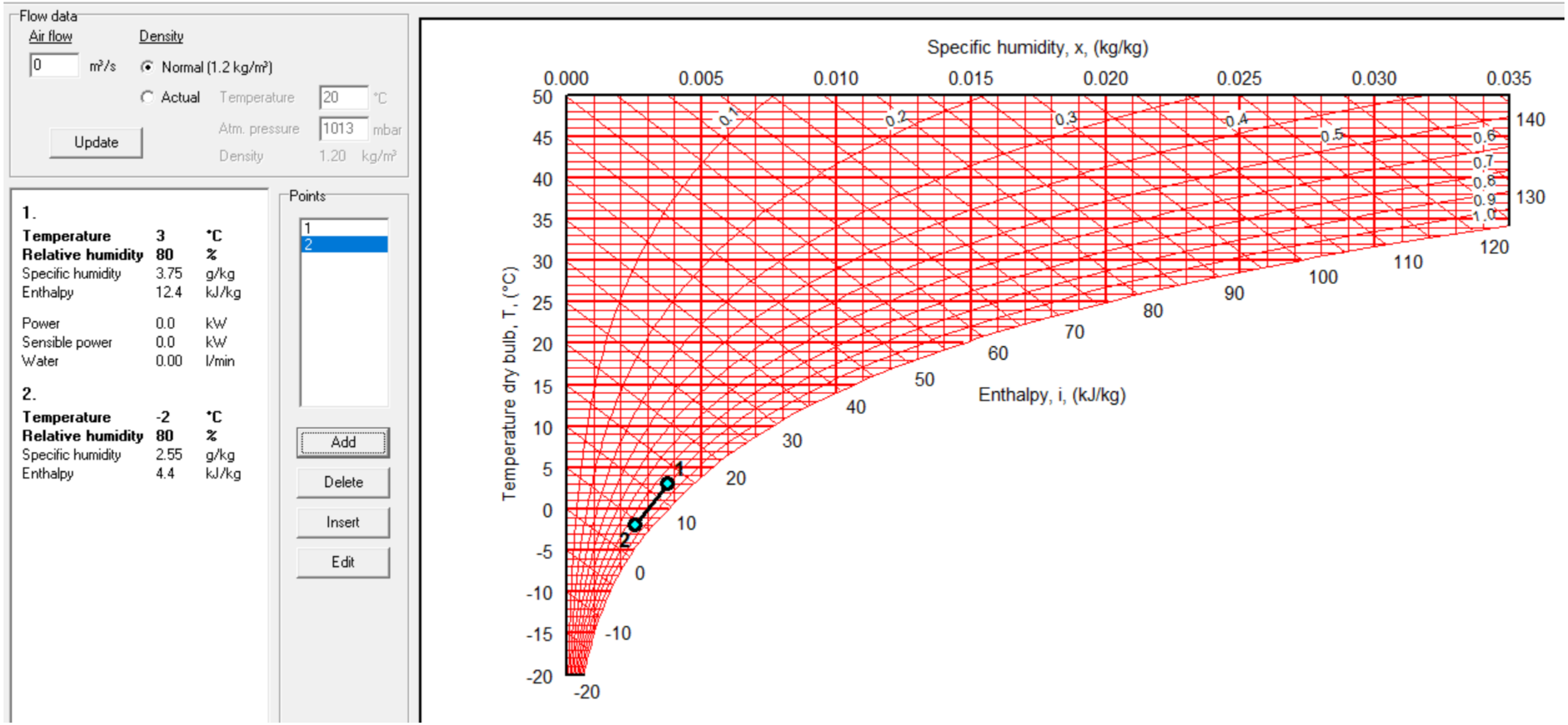


Figure 4.3: *Variations in air moisture content to air temperature at 80% relative humidity*

4.5 Frost melting rate

The melting rate depends on the frost thickness and capacity of the defrosting system. Since the fin space is 4 mm, to avoid blocking this space, the frost should be melted before reaching a 2 mm thickness. The following assumptions were made to estimate the tank sizing for the defrosting system:

- The frost growth rate considered for this project is 0.0004 mm/s to have a safety margin.
- The frost melting rate is estimated for four scenarios with 0.5, 1, 1,5, and 2 mm of frost thickness.
- The melting of the dry cooler units will be done individually, not simultaneously.
- The difference between the inlet and outlet temperature of the 37% ethylene glycol mixture at dry cooler units is constant and equal to 23°C during the defrosting period.
- The volume flow rate of the defrosting system is considered to be 21 Liter/s. Then the heat capacity of the system is calculated based on the 23°C temperature difference giving 1,834.6 kW as a result.
- According to Kelvin's estimation, about 300,000 kJ of additional energy is available from the plate heat exchanger working between the 37% ethylene glycol mixture and water.
- The piping of the defrosting system has a length of 80 m and a 211.1 mm inner diameter which gives 2,799 liters of volume available that can work as thermal storage just like the accumulator tank.

4.6 Accumulator tank sizing

The calculation of the accumulator tank size is done for four scenarios: 0.5, 1, 1.5, and 2 mm frost thickness limit since the space available for the airflow is 4 mm and frost can be accumulated on both surfaces. The time to reach the set frost layer thickness for each scenario is calculated and shown in Table 4.2.

Table 4.2: *Time to reach different frost layer thickness at a growth rate of 0.0004* mm/s.

Frost thickness [mm]	Time [minutes]
0.5	23.52
1	47.03
1.5	70.57
2	94.08

The frost melting time for each scenario is calculated using the latent heat of ice, the heat capacity of the system, and the additional energy available from the plate heat exchanger in the system. Next, the volume of the tank is calculated based on the volume capacity of the defrosting system and taking into consideration the volume of the pipes as storage as well. This is shown in detail in Table 4.3.

Table 4.3: *Variation of frost melting time and volume of tanks.*

Frost thickness limit [mm]	**0.5**	**1**	**1.5**	**2**
Required heat for melting [kJ]	228,483	756,965	1,285,448	1,813,931
Melting time [minutes]	2.075	6.88	11.68	16.48
Tank volume [Liters]	183	5,866	11,915	17,964

Due to the time it takes to melt the frost layer and the size requirement of the tank, it was decided that a limit of 1 mm thickness would be preferred to avoid long defrosting cycles and large accumulating tanks since the system must stop working during this defrosting period and due to space limitations. As shown in Table 4.3, this frost thickness will require a tank with an estimated volume of 5,866 liters, a round-up value of 6,000 liters.

Based on the obtained results, the defrost system would be programmed to melt the ice after 47 minutes, melting a frost layer that is 1 mm in thickness. This defrosting process would last approximately 7 minutes, giving enough time to reload the storage tank before the next dry cooler requires defrosting. The required volume of the tank to melt 1 mm of ice is 5,866 liters, therefore a 6,000-liter tank is proposed to meet this demand. Since it is planned to install two accumulator tanks, hydraulically coupled in series, the volume of each tank can be 3,000 liters.

CHAPTER 5

System simulation

This chapter will describe in detail the simulations of the systems, the input parameters for each of their components, and a brief description of the software used for the simulation and data visualization of the results.

5.1 Software simulation: Modelica

Modelica is a software by Dymola that allows the simulation or modeling of mechanical, electrical, and thermal process systems. There are multiple libraries to choose from depending on the type of system to simulate, and the component's behavior is defined by mathematical equations [20]. This software was essential for this project since it was the program used to perform all the models.

5.2 Software visualization: DaVE Data visualization and Simulation Environment

DaVE is a visualization and simulation environment software that has the ability and tools to display different graphs and diagrams of thermal systems such as state diagrams for refrigerants and other fluids [21]. This program was of particular importance to the project to further understand the results obtained from Modelica. It was used to create a simple visualization of the system's logarithmic-pH diagram for the propane heat pump model.

5.3 Software visualization: CoolProp

CoolProp is a C++ library that provides properties of incompressible fluids as well as equations of state and transport properties for fluids. It can be used with Microsoft Excel, EES, and MATLAB, among other software. For this project, it was used in conjunction with Microsoft Excel to calculate the properties of the refrigeration fluid in the propane heat pump cycle [22].

5.4 Software visualization: Coolpack

CoolPack is a software that supplies a library of simulation models for refrigeration systems. It facilitates the analysis of refrigeration cycles, component dimensioning, and overall optimization of the process [23]. In this project, the software was used to model the basic propane heat pump cycle and use it as a reference to compare it with the results obtained from the heat pump modeled in Modelica.

5.5 System description: Propane Heat Pump Simulation

An initial model of the propane heat pump unit was made during the specialization project which served as a basis to understand and simulate broadly the operation of the one to be installed in Teknobyen. This section will present briefly the improvements made to this initial model to achieve closer conditions to the ones in real life.

The general cycle consists of an evaporator, compressor, condenser, expansion valve, volume tank, and two PI controllers, as shown in Figure 5.1. The evaporator and the condenser are simulated as plate heat exchangers and the fluids utilized were propane and water to simplify the simulation in Modelica. The first step prior to making this model was to obtain the high and low pressure and the enthalpy for each point of the heat pump cycle, this was done with CoolProp in Excel and Coolpack. The model will simulate the operation of the heat pump in heating mode, that is, during the cold months of the year.

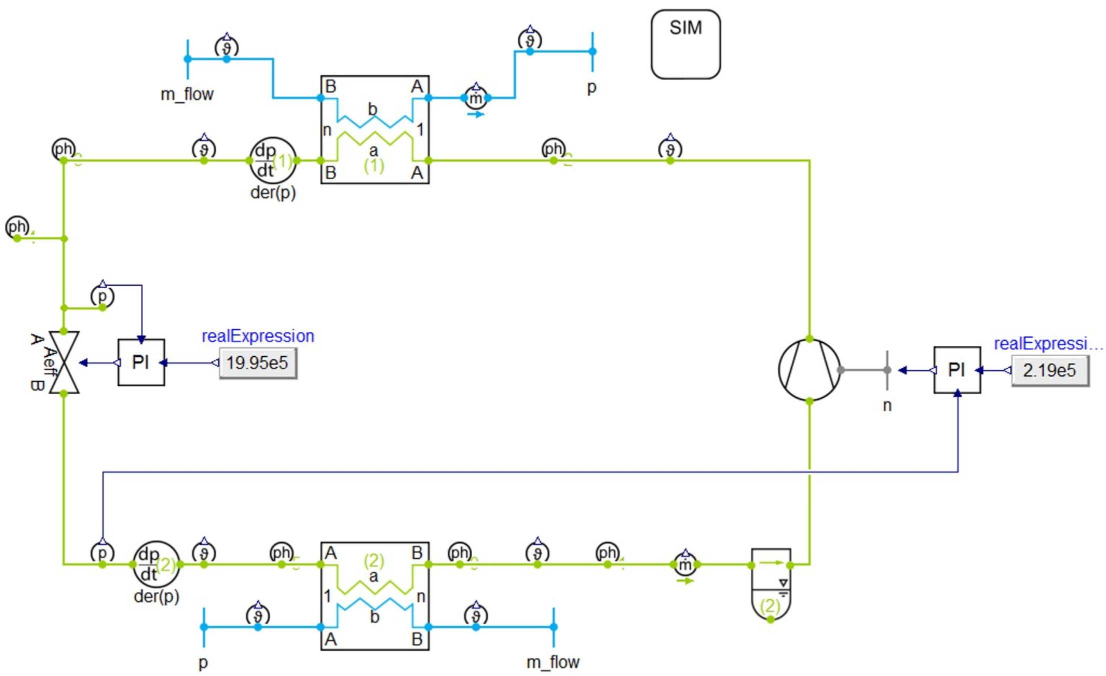


Figure 5.1: *Propane Heat Pump Model*

Evaporator

The evaporator has a geometry of a plate heat exchanger with a number of cells of 5, a water mass flow rate of 4.108 kg/s, and a temperature of -16°C at the inlet of the evaporator. The material used for this equipment is stainless steel. The heat transfer model for the propane and water side was considering a constant alpha of 1,906 W/m^{2}K and considering the following data for the plate heat exchanger.

Table 5.1: *Data input to evaporator*

Parameter	Value
Number of Plates	150
Length [m]	0.632
Width [m]	0.322
Pattern angle [°]	35
Wall thickness [mm]	0.7
Pattern amplitude [mm]	2
Pattern wave length [mm]	12.6

A quadratic mass flow-dependent pressure drop model was selected with the following input as given by the technical data sheets of the system for the propane side and the water side.

Table 5.2: *Data input to pressure drop in evaporator*

Parameter	Propane side	Water side
Nominal mass flow rate [kg/s]	0.2691	4.108
Nominal pressure drop [kPa]	18.8	5.84

Compressor

The compressor utilized by the model was a simple compressor with a PI controller to regulate the flow rate of propane in the system to achieve the desired low pressure of the system, which is 2.19 bar. The following table summarizes the input data to this equipment and additional information on the PI controller input can be found in Appendix B. A separator tank was added between the evaporator and the compressor to facilitate the simulation of the model. The dimension of this tank is 0.050 m^3 and an initial filling level of 30% .

Table 5.3: *Data input to compressor*

Parameter	Value
Displacement [m^3]	0.00136
Volumetric Efficiency	0.87
Isentropic Efficiency	0.73

Condenser

The condenser has a geometry of a plate heat exchanger with a number of cells of 5, having a water mass flow rate of 2.44 kg/s and a temperature of 45°C at the inlet of the condenser and it is made of stainless steel. The heat transfer model for the propane and water side was considering a constant alpha of 1,654 W/m^{2}K and considering the following data for the plate heat exchanger.

Table 5.4: *Data input to condenser*

Parameter	Value
Number of Plates	180
Length [m]	0.632
Width [m]	0.322
Pattern angle [°]	35
Wall thickness [mm]	0.75
Pattern amplitude [mm]	2
Pattern wave length [mm]	12.6

To calculate the pressure drop in this equipment, a quadratic mass flow-dependent pressure drop has been selected, and Table 5.5 presents the input data according to the information given by the equipment producer.

Table 5.5: *Data input to pressure drop in condenser*

Parameter	Propane side	Water side
Nominal mass flow rate [kg/s]	0.2691	2.44
Nominal pressure drop [kPa]	0.0406	0.7

Expansion valve

A regular heat pumping system has an expansion valve to lower the pressure between the condenser and evaporation. In the previous model developed during the specialization project, this was done in two steps with two expansion valves. This was improved in the latest model developed during spring 2023 with a one step expansion. In this case, an expansion valve was implemented with a PI controller to regulate the flow and maintain the high pressure at the set point of 19.95 bar. The PI control has as input the pressure after the condenser and it gives the expansion valve an effective flow area depending on the pressure set point defined. The PI control on the low-pressure side, connected to the compressor, works in the same way except the pressure sensor is located between the expansion valve and the evaporator and the set pressure for this control is 2.19 bar.

Simulation assumptions and limitations

As previously mentioned, in order to simplify the simulation of this complex system, some assumptions were made:

- The fluids involved in the system were only propane and water, not propane, water, and the ethylene glycol mixture with water at 37%.
- The dimensions input into the condenser and evaporator follow the ones specified in the manufacturer information sheet on the number of plates, length, and width. Due to a lack of information on the remaining dimensions required by Modelica, the suggested values by the program were used.

Considering the latter, the results may differ to some degree from the ones expected from the real-life system. The comparison and discussion of these results will be presented further in chapter 6 and chapter 7.

5.6 Ethylene glycol system

In order to simulate how the overall system would perform under different weather scenarios, an ethylene glycol model had to be developed since this working fluid is going through the dry coolers and obtaining heat from the ambient temperature. Additionally, this working fluid gains heat from several heat exchangers in the different buildings as explained in chapter 3. This model consists of seven tubes that behave as heat exchangers and one heat exchanger that represents the four dry coolers in the system. Figure 5.2 shows the overall diagram of this cycle.

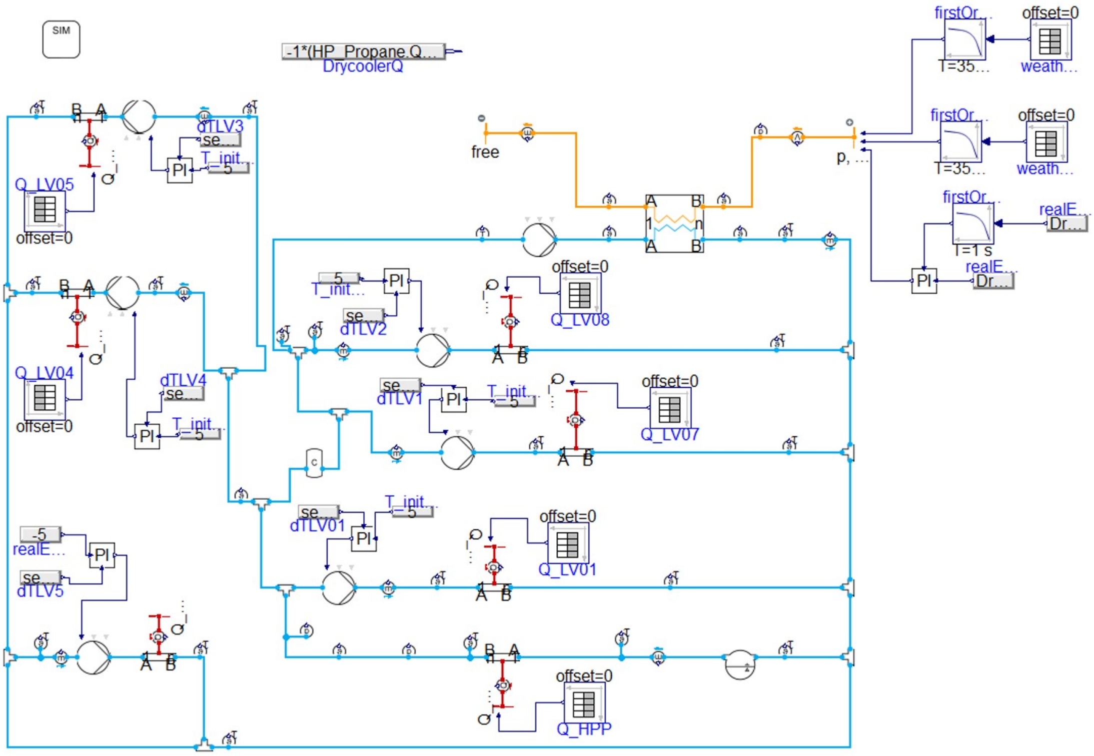


Figure 5.2: *Ethylene glycol model diagram.*

Weather data and case scenarios

The ethylene glycol model will be run under two case scenarios with different temperature and relative humidity data. The first case will have air temperatures between -7.8 and 6.8°C and relative humidity values between 51 and 96%. The second case scenario will work under temperatures between -16.9 and -4.7 °C and relative humidity ranging from 70 to 91%. The model will simulate for 5 consecutive days to define how the system performs for several hours under the described conditions. It is relevant to mention that the weather data used for the simulations was retrieved from the Norwegian Meteorological Institute and the Norwegian Broadcasting Corporation [16], the specific data can be found in Appendix B.

Heat exchangers and heat pump units

As stated before, the model has several tubes that behave as the different heat exchangers where the ethylene glycol mixture can go through, two of them being the propane and CO_2 heat pumps and five of them being heat exchangers that cool down data centers, kitchen facilities or existing machinery. Table 5.6 summarizes the name, capacity, and function of each unit. The tube that simulates the propane heat pump has a total capacity of 1,854.4 kW due to the fact that the system includes two heat exchangers with four circuits of 231.8 kW each.

Table 5.6: *Heat exchanger units in ethylene glycol model*

Unit Code	Description	Capacity [kW]
HP-P	Evaporator to the propane heat pump	1,854.4
HP-CO2	Evaporator to the CO_2 heat pump	50
LVO1	Cooling for the building, kitchens, and data centers in TST	1200
LV04	Cooling machine in PB2	400
LV05	Cooling data centers in PB2	200
LVO7	Building cooling in AG5	416
LVO8	Cooling of existing machinery in AG5	140

Dry cooler

The dry coolers are represented by a heat exchanger in the model. In the original design there will be four dry coolers, yet, to simplify the complexity of the model, a large heat exchanger with the equivalent dimensions of the four smaller heat exchangers was used.

Figure 5.3: *Pictures of the dry coolers of the system.*

The original dry coolers have a length of 11.489 m and a height of 2.532 m. Each dry cooler is composed of 2 rows of 84 tubes that pass three times through the equipment before leaving it. As mentioned before, one large heat exchanger was considered for the model in Modelica, which resulted in the following dimensions for the heat exchanger. Additional information on the dry coolers can be found in Appendix B.

Table 5.7: *Data input to dry cooler*

Parameter	Value
Finned tube length [m]	11.489
N serial tubes	672
Serial tube distance[m]	$22e-3$
N parallel tubes	3
Parallel tube distance [m]	$25.4e-3$
Fin thickness [m]	$0.2e-3$
Fin pitch [m]	0.004
Tube inner diameter [m]	0.005
Tube wall thickness [m]	0.0015
N tube side parallel hydraulic flows	672

This dry cooler obtains an air flow stream that is connected to the weather data previously mentioned, having air temperature and relative humidity values per hour for the five days the simulation takes place. First order elements were added to make the transition between temperatures and relative humidity smoother and gradual instead of instantaneous. These first order elements are set to distribute the temperature and relative humidity change through 3,500 seconds or 58 minutes. The airflow rate is controlled by a PI controller which will be further explained in this chapter.

Overall energy balance

A general energy balance was made in the system to calculate how much heat would the dry coolers have to provide to the ethylene glycol mixture to keep the equilibrium between the heat flow in the heat exchangers presented before. This energy balance was made with the following equation.

$$Q_{Drycooler} = Q_{HPP} + Q_{HPCO2} + Q_{LV01} + Q_{LV04} + Q_{LV05} + Q_{LV07} + Q_{LV08} \tag{5.1}$$

where:

- $Q_{\mathrm{Drycooler}}$: Heating required from the dry cooler
- Q_{HPP}: Heating required by the propane heat pump
- Q_{HPCO2}: Heating required by the CO_2 heat pump
- Q_{LV01}: Heating obtained from the building cooling, kitchen, and data centers in TST
- Q_{LV04}: Heating obtained from the cooling machine in PB2
- Q_{LV05}: Heating obtained from the cooling of data centers in PB2
- Q_{LV07}: Heating obtained from the building cooling in AG5
- Q_{LV08}: Heating obtained from the cooling of existing machinery in AG5

From Equation 5.1 it can be understood the propane and CO_2 heat pump require a certain amount of energy, part of it being delivered by the heat exchangers LV01, LV04, LV05, LVO7, and LV08. The rest of the energy required must be provided by the dry cooler, partially or in its totality.

Peak hours

With the purpose of representing more accurately the energy demand of the buildings, assumptions were made to simulate the peak and off-peak hours of the buildings. Since the buildings will be used for office and commercial use, their highest energy consumption would be between 7 am to 7 pm. This time range was chosen and based on the function of each heat exchanger unit, a percentage of their capacity was selected for their high and low energy requirements. The only heat exchanger that was not subjected to different values as on and off-peak hours is the CO_2 heat pump for DHW. This heat pump was working at full capacity regardless of the time of the day. The following table contains the capacity used for each heat exchanger during the peak and off-peak hours of the day for the first and second case scenario. These values were introduced in the model as inputs for the tubes that represent the heat exchangers and heat pump units.

Table 5.8: *Energy requirements peak and off-peak hours first and second case*

Unit Code	Peak Consumption [kW]	Off-peak Consumption [kW]
HP-CO2	50	50
LVO1	500	400
LV04	240	160
LV05	160	120
LVO7	249.6	166.4
LVO8	112	84

The propane heat pump demand varies between the first and second case since the second case presents lower temperatures and therefore will have a higher energy demand than the first case. The requirements for the propane heat pump for the first and second cases can be found in Table 5.9.

Table 5.9: *Energy requirements propane heat pump*

Case	Peak Consumption [kW]	Off-peak Consumption [kW]
1	1,483	1,112
2	1,600	1,200

PI Controllers and mass flow rates

There are several PI controllers in this simulation, all of them controlling the mass or volumetric flow rate that goes through the different parts of the system. In this section, a detailed description will be given of the different controllers, the variables they are considering as input, and their set points.

The PI controllers prior to the tubes that represent the heat pump and heat exchangers in the system regulate the mass flow rate that goes through that particular area of the cycle. These controllers have as reference the difference in temperature before and after the tube and their set point is to make this difference 5 °C. Initially, a fixed desired temperature was used as a set point, yet, working with a temperature difference instead proved to be more efficient and overall worked better for the system. This is because the temperature of the ethylene glycol in the system will change depending on the seasons and weather conditions.

In the case of the dry cooler, the PI is regulating the volumetric air flow rate that enters the dry cooler based on the overall energy balance equation mentioned before and the heating that the dry cooler is providing. It aims to match them to provide the necessary energy to the system. The reading of this energy balance also has a first order element to facilitate the regulation of the simulation, yet this is only set for 1 second unlike the other first order elements added to the air temperature and the relative humidity data from the weather conditions.

The mass flow rate of the glycol that goes through the dry cooler heat exchanger is fixed at 41 kg/s for the first case, which is the load of two of the dry coolers. For the second scenario, a fixed load of 62 kg/s was used, equivalent to the load of three dry coolers. This was done to simplify the complex model yet it can be further modified to make it dynamic as the flow rate of air going into the dry cooler.

Simulation assumptions and limitations

With the purpose of simplifying the complexity of the system, the following assumptions were made:

- The fluid utilized in the system was water instead of 37% ethylene glycol mixture in water.
- The pressure drop was neglected in the dry cooler heat exchanger.
- The dry cooler heat exchanger was simulated as one large equipment with the dimensions of four times the dimensions of one dry cooler instead of simulating four individual dry cooler heat exchangers.
- The ethylene glycol mass flow rate through the dry cooler was fixed in both scenarios instead of being dynamic and regulated by a PI controller.
- Instead of plate heat exchangers, tubes with heat port connections were used to represent the heat exchangers and heat pumps.
- It is assumed that there is no heat loss from the heat exchanger tubes.

Due to these assumptions and limitations, the results obtained from this model could differ from the ones obtained with the current equipment installed in Teknobyen. Further comparison and discussion of these results will be given in chapter 6 and chapter 7.

CHAPTER 6

Results

This chapter will cover the results of the simulations previously explained in chapter 5 for the propane heat pump and the ethylene glycol system. The following sections will describe in detail the results obtained in each case, further discussion on the results and the simulation's validity will be presented in the chapter 7.

6.1 Expected results: Propane heat pump simulation

The main factors to consider while working with the propane simulations were the low and high pressure of the system, the enthalpy outside the condenser and evaporator, the water outlet temperature from the condenser and evaporator, the COP, and the quick control of the low and high pressure of the system with the PI controls. With the temperature and pressure from the inlet and outlet of the evaporator and condenser, an approximate of the enthalpy was calculated at each point of the cycle with Coolprop and Coolpack. In addition, the equipment manufacturing information was taken into consideration as the ideal or expected results to obtain. This information is summarized in Table 6.1.

Table 6.1: *Estimated values for the propane cycle*

Parameter	Estimated value
Enthalpy after evaporator [kJ/kg]	555.77
Enthalpy after compressor [kJ/kg]	700.35
Enthalpy after condenser [kJ/kg]	340
Enthalpy expansion valve [kJ/kg]	340
Temperature water outlet condenser [°C]	55
Temperature water outlet evaporator [°C]	-20
High-pressure propane system [bar]	19.95
Low-pressure propane system [bar]	2.19
Coefficient of Performance	2.56

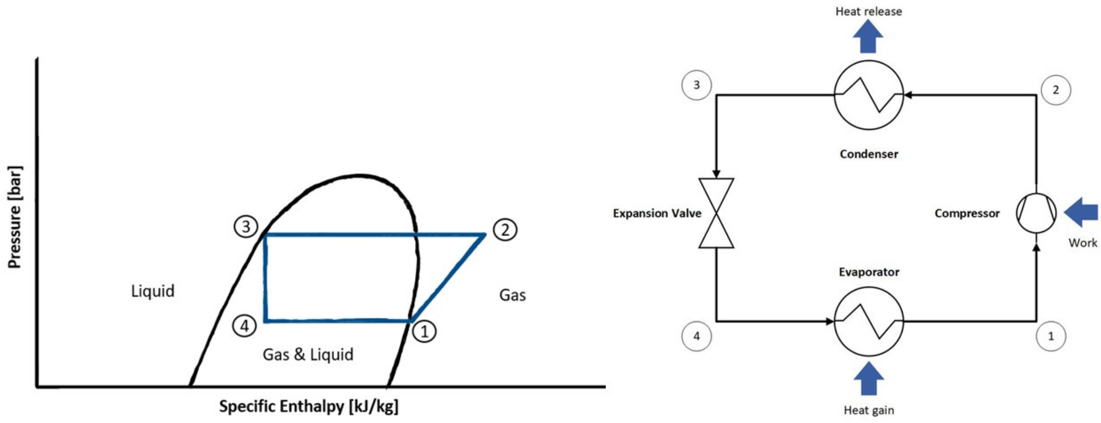


Figure 6.1: *Propane System Cycle*

It is of relevance to mention that the system described by the manufacturer has a 5 K of superheat in the evaporator, which makes it reach an enthalpy of 555.77 kJ/kg as seen in Table 6.1. The actual value without the superheating is around 548 kJ/kg. Similarly, the real system has a subcooling of 4 K in the condenser. The minimum necessary enthalpy to reach after the condenser is 355 kJ/kg without the subcooling.

6.2 Results obtained: Propane heat pump simulation

The next sections will describe in detail the results obtained, the logarithmic-pH diagram, and their PI control for the high and low-pressure sides. The results obtained from the propane heat pump model are summarized in Table 6.2 for each relevant point mentioned before and Figure 6.2 depicts the resulting logarithmic-pH diagram for the cycle using DaVE. The model was simulated for 1 hour.

Table 6.2: *Propane cycle results in Modelica*

Parameter	Obtained value
Enthalpy after evaporator [kJ/kg]	559.9
Enthalpy after compressor [kJ/kg]	712.2
Enthalpy after condenser [kJ/kg]	326
Enthalpy after expansion valve [kJ/kg]	326
Temperature water outlet condenser [°C]	56.56
Temperature water outlet evaporator [°C]	-20.08
High-pressure propane system [bar]	19.95
Low-pressure propane system [bar]	2.19
Coefficient of performance	2.43

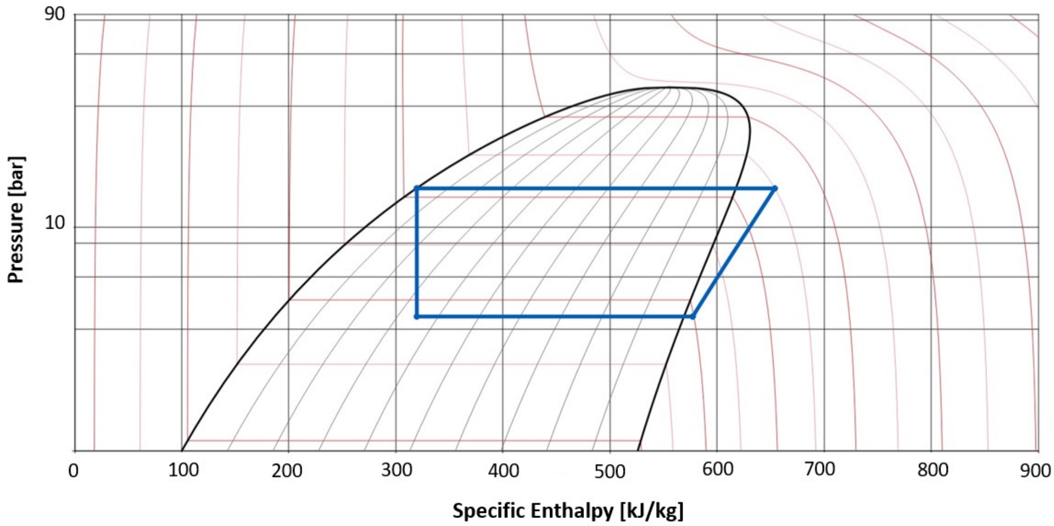


Figure 6.2: *Propane System Results*

From Figure 6.2 it can be observed that the overall behavior of the cycle is correct and it does not present any major change from the one expected. Taking a deeper look at Figure 6.2 and Table 6.2 it can be perceived that the enthalpy outside the evaporator surpasses slightly the estimated value of 555.77 kJ/kg. The enthalpy obtained in the simulation is 559.9 kJ/kg, an approximate deviation of 0.7%. The enthalpy obtained after the compression was 712.2 kJ/kg, only a 1.69% difference from the estimated value of 700.35 kJ/kg. Another important point in the cycle is when the propane leaves the condenser. It is expected to reach an enthalpy of 340 kJ/kg to fully condensate the propane mass flow rate and have the subcooling level defined by the manufacturer. In this simulation the enthalpy reached at this point went slightly further to 326 kJ/kg, giving a deviation of 4.11%. This is of no concern since it means that the fluid is further subcooled at the condensing stage. The enthalpy after the expansion valve also resulted in a 4.11% difference from the estimated one.

Another important factor while working with the simulation is the temperature of the water going outside of the condenser since that water is used for the heating of the three buildings that compose Teknobyen. The temperature for this simulation was 56.56°C, giving a deviation of 2.83%, a 1.56°C difference from the expected value of 55°C. The temperature of the water outside the evaporator was also of relevance to verify the appropriate heat transfer to the propane was given. This temperature was around -20.08 °C, resulting in a deviation of around 0.4% from the expected value of -20°C.

The COP of this simulation is 2.43, a 5% deviation from the expected value. The high-pressure side of the propane system achieved a pressure of 19.95 bar as was expected and the low-pressure side reached a pressure of 2.19 bar as expected. The PI control for the low-pressure side controlled the system around the second 191, or a little over 3 minutes. However, the high-pressure side control took around 833 seconds or 13.8 minutes to control the pressure and reach the set point value.

6.3 Expected results: ethylene glycol system

The main relevant variables to analyze regarding the ethylene glycol system are the temperature of the working fluid, the heat provided by the dry cooler heat exchanger, and the flow rates through the different tubes and the dry cooler. These flow rates are regulated by their respective PI controller.

The temperature of the working fluid should not go lower than -20°C since the ethylene glycol freeze point is around -22°C. It is expected that the working fluid temperature will vary as the air temperature changes over time. Additionally, it is a requisite that the dry cooler manages to provide the necessary heating that is required by the buildings. In the case the dry cooler cannot deliver all this heating, an electrical boiler has been installed to further heat the water that heats the buildings, nevertheless, the purpose of the dry cooler is to obtain as much energy from the environment as possible. Regarding the flow rates in the tubes that simulate the different heat exchangers and heat pump units, it is important to verify they are within a reasonable range from the mass flow rates set by the manufacturer. Lastly, the airflow rate in the dry cooler must also be within the range given by the information sheets on the heat exchangers.

As mentioned in previous sections, the flow rate through the different heat exchangers in the system is regulated by several PI controllers which set the mass flow rate to achieve a temperature difference of 5°C. The tubes releasing heat into the working fluid should increase the temperature in 5°C and the CO_2 heat pump should diminish the temperature 5°C as seen in Figure 6.3.

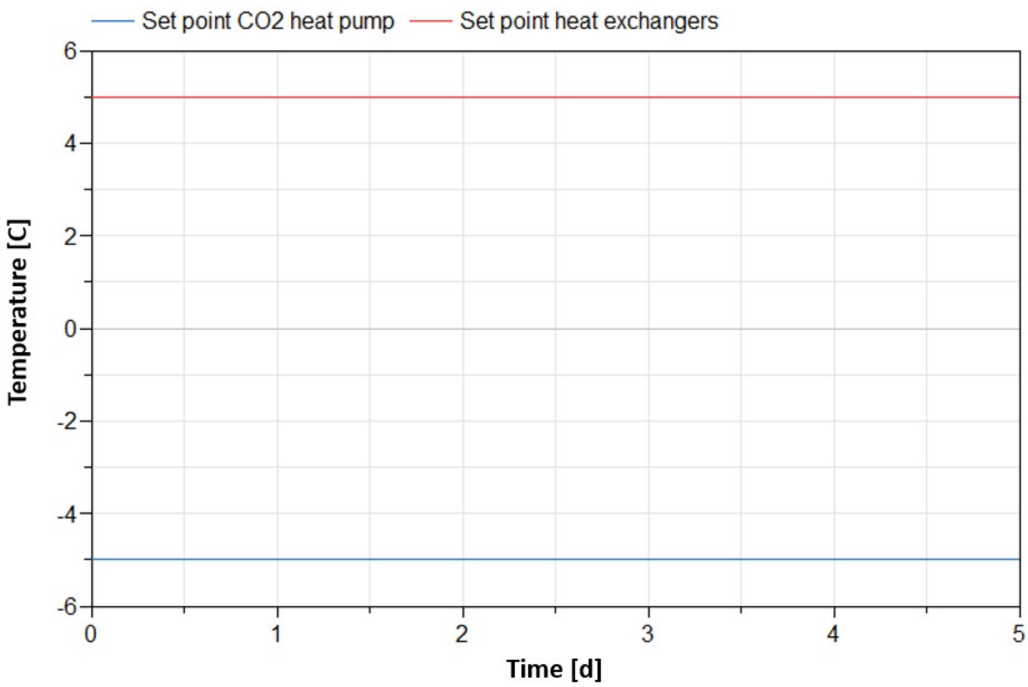


Figure 6.3: *Temperature difference set point for CO_2 heat pump and heat exchangers in case 1 and 2*

6.4 Results obtained: Ethylene glycol system case 1

After running the model for five consecutive days with weather conditions between -7.8 and 6.8°C of temperature and relative humidity between 51 and 91%, the following results were acquired.

Working fluid temperature

Figure 6.4 and Figure 6.5 display the air temperature, the temperature of the working fluid after the dry cooler, and the temperature before and after the propane heat pump.

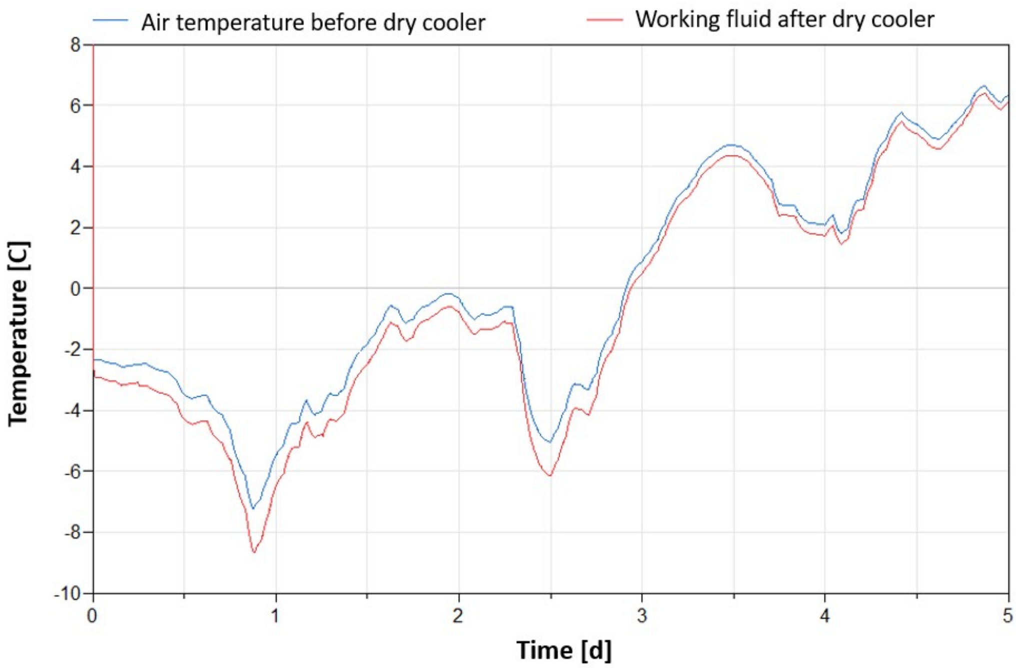


Figure 6.4: *Air and working fluid temperature in dry cooler for case 1*

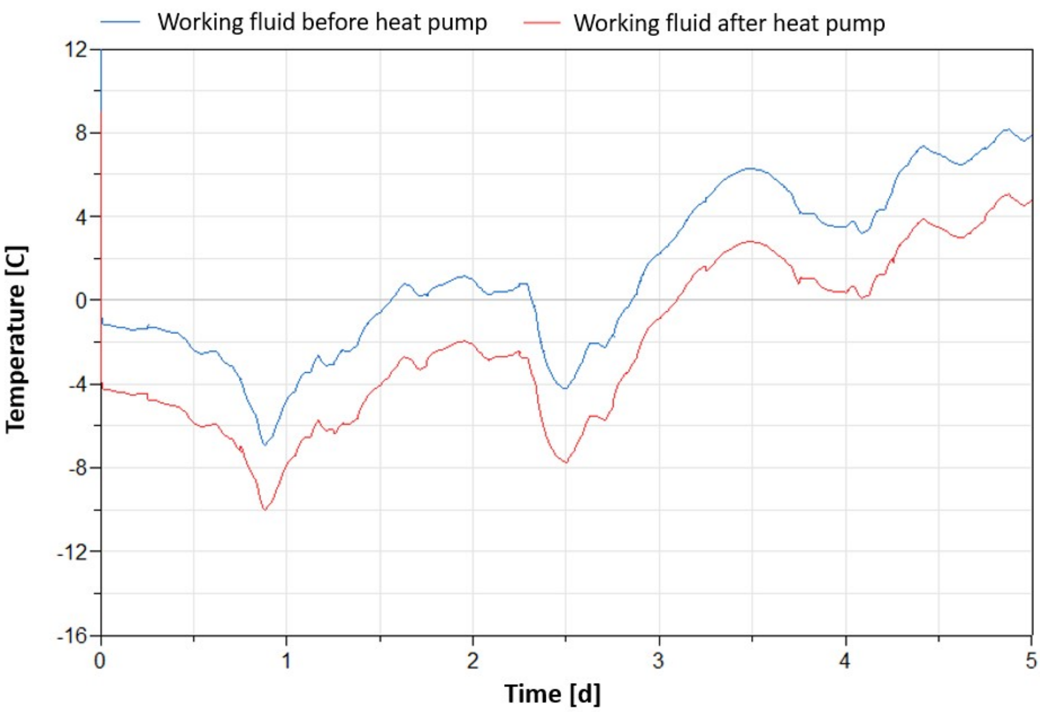


Figure 6.5: *Working fluid temperature in the propane heat pump for case 1*

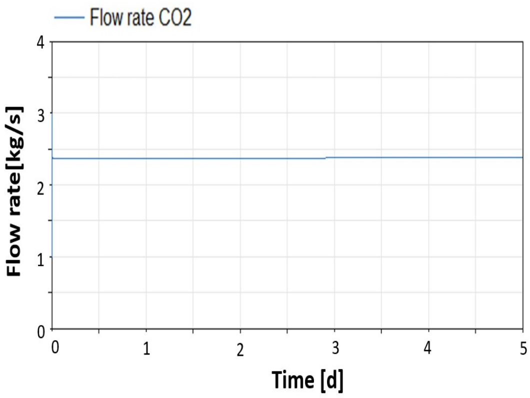
As it can be observed from Figure 6.4 and Figure 6.5, the working fluid's temperature is dependent and varies according to the air temperature. Moreover, the temperature of the working fluid varies through an acceptable range, far from the freezing point of the ethylene glycol. It can be seen that the temperature of the working fluid before the propane heat pump is higher than the temperature of the air and the working fluid after the dry cooler. This is due to the heat exchangers that deliver heat from other parts of the building to the working fluid. This energy is not constant throughout the day, since there are peak and off-peak hours loads to the heat exchangers. Therefore, during some parts of the day the difference between the working fluid temperature before the propane heat pump and the air temperature is not as large as in other times of the day. The lowest temperature of the working fluid in the system will be the temperature after the propane heat pump, this temperature ranges between 5 to -9.93°C.

Flow rates and PI controllers: heat exchangers

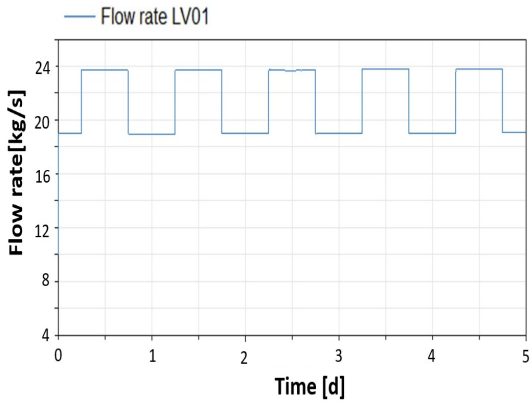
The flow rates of the different heat exchangers vary depending on the time of the day, if it is peak or off-peak time. Since the load of the CO_2 heat pump is constant, then the mass flow rate through it remains constant as well. This can be observed in Figure 6.6 and the respective flow rates for each unit during peak and off-peak hours can be found in Table 6.3.

Table 6.3: *Flow rates through heat exchanger and CO_2 heat pump case 1*

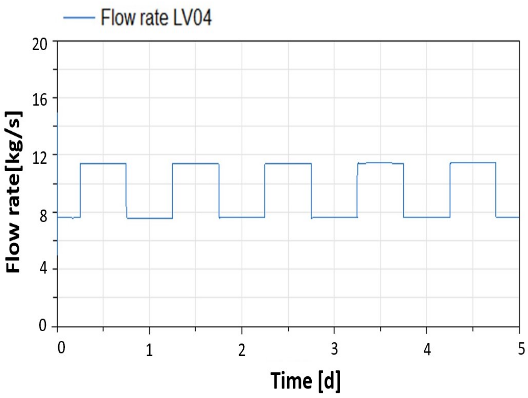
Unit Code	**Peak flow rate [kg/s]**	**Off-peak flow rate [kg/s]**
HP-P	100.82	85.13
HP-CO2	2.37	2.37
LVO1	23.7	18.97
LV04	11.37	7.59
LV05	7.58	5.69
LVO7	11.83	7.89
LVO8	5.31	3.98



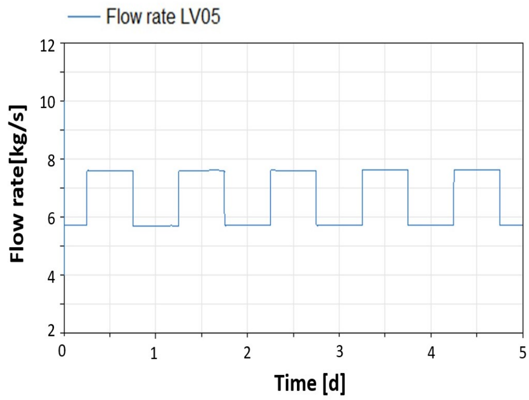
(a) *Flow rate in CO_2 heat pump*



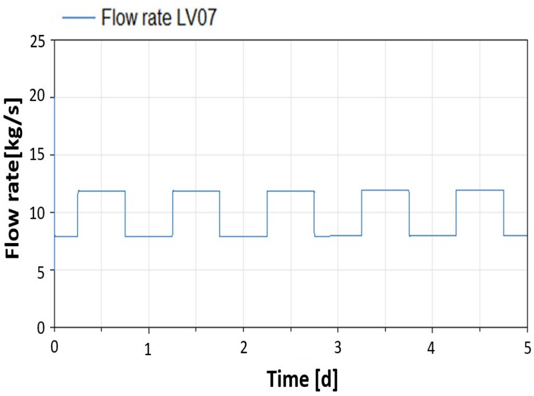
(b) *Flow rate in LV01*



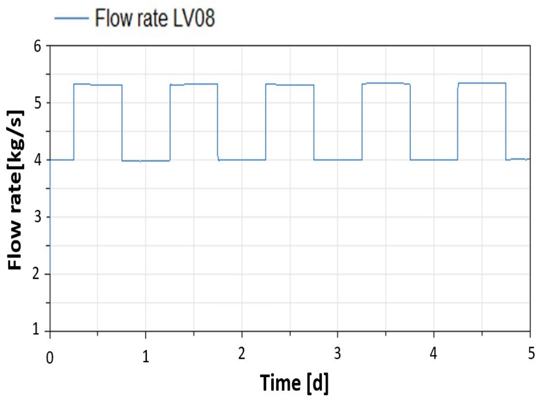
(c) *Flow rate in LV04*



(d) *Flow rate in LV05*



(e) *Flow rate in LV07*



(f) *Flow rate in LV08*

Figure 6.6: *Flow rates regulated by PI controllers case 1*

Flow rate and PI controller: dry cooler

The PI controller in the dry cooler had as objective to match the heating provided by the dry cooler to the heating required by the buildings as explained in previous chapters. The overall performance of the dry cooler can be observed in Figure 6.7. The heating requirements during off-peak hours were around 232 kW and 272 kW for peak hours. Note that the scale is inverted in the next figure due to the flow direction of the energy in the system.

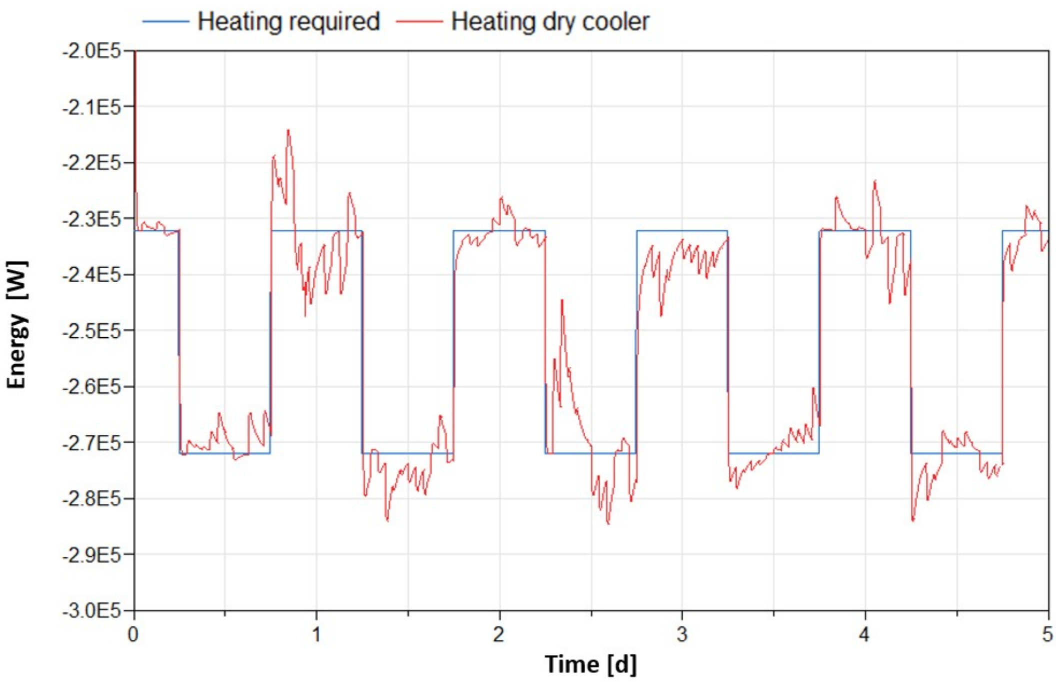


Figure 6.7: *Heating requirements and dry cooler performance for case 1*

This controller had a large range of operation since the heat exchanger represented the four dry coolers in one equipment. In Figure 6.8 it can be observed how the air flow rate varied through the five simulated days.

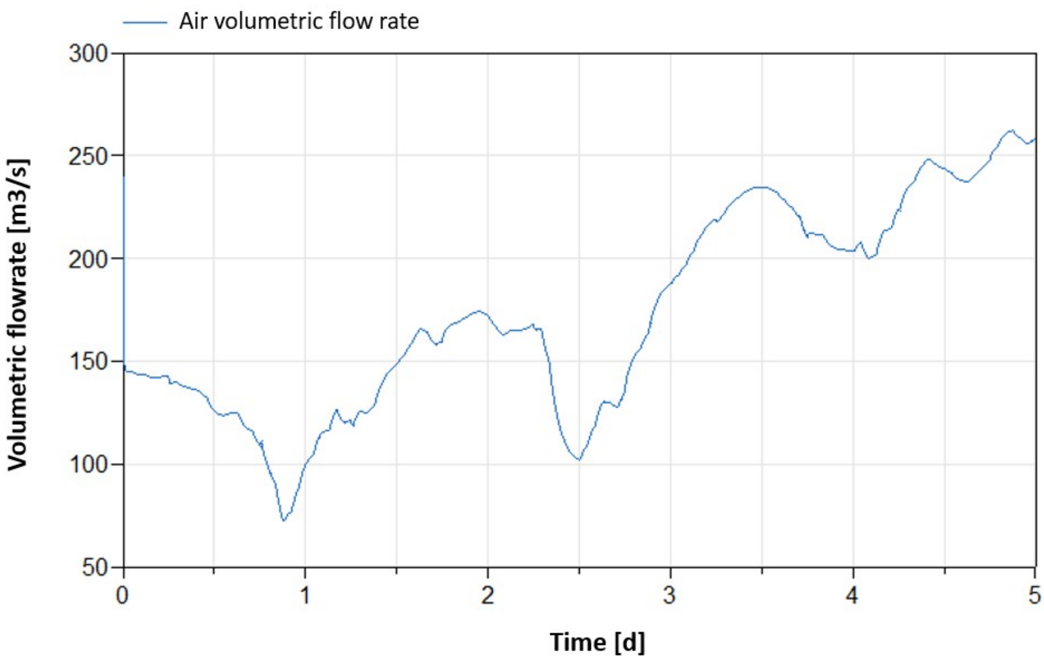


Figure 6.8: *Air flow rate through the dry cooler for case 1*

6.5 Results obtained: Ethylene glycol system case 2

The following results were obtained after simulating the model for five consecutive days with weather conditions between -16.9 and -4.7°C of temperature and relative humidity between 70 and 96%.

Working fluid temperature

The following images show the air temperature, the temperature of the working fluid after the dry cooler, and the temperature of the working fluid before and after the propane heat pump.

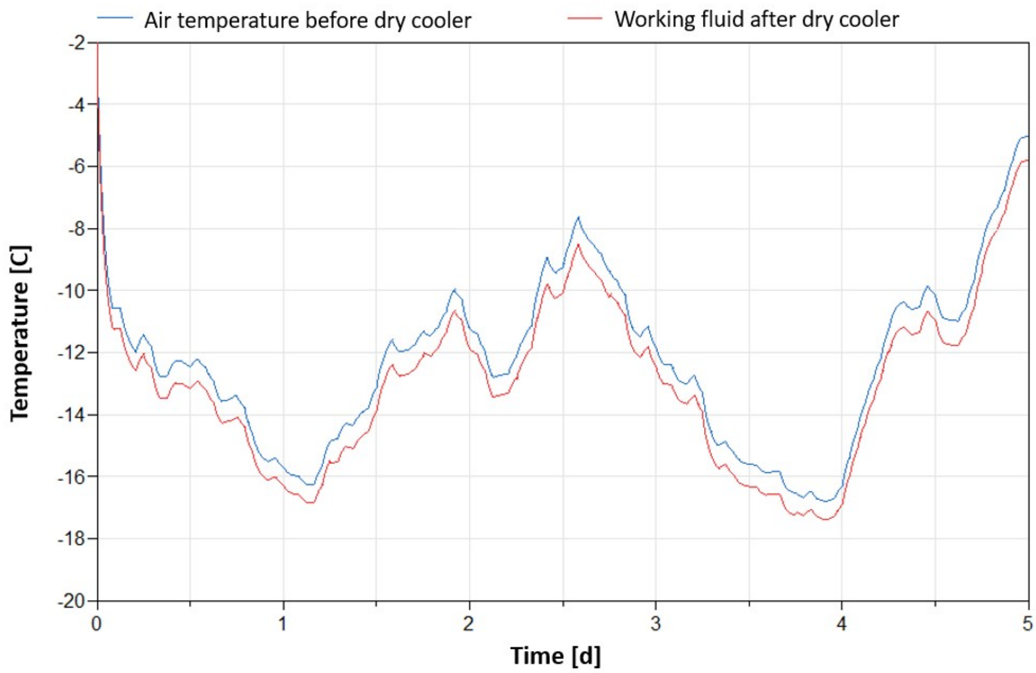


Figure 6.9: *Air and working fluid temperature in dry cooler for case 2*

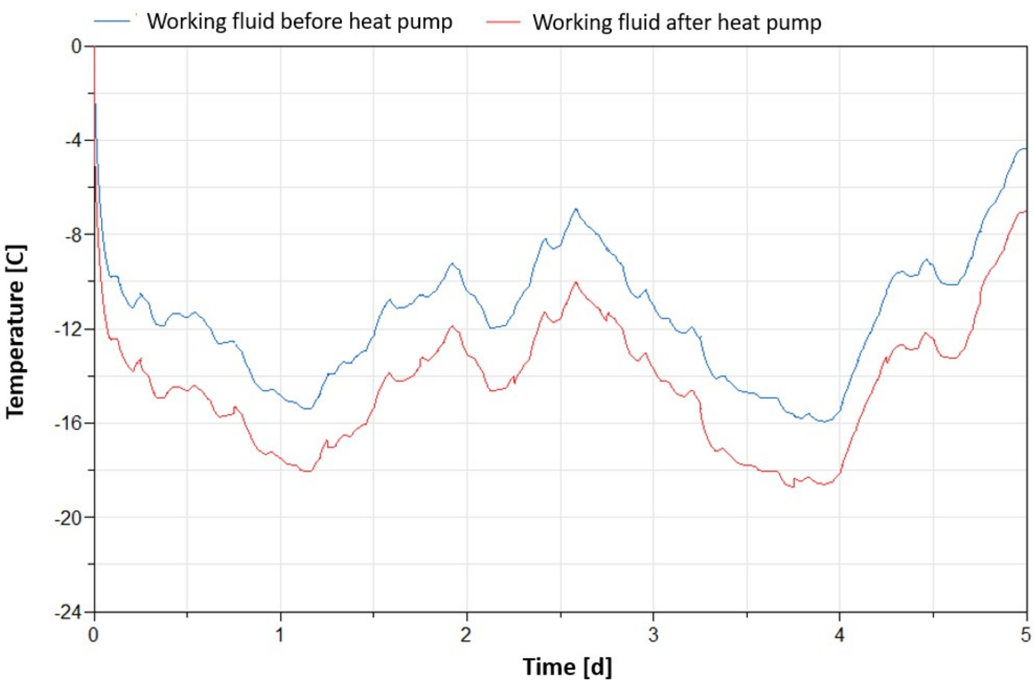


Figure 6.10: *Working fluid temperature in the propane heat pump for case 2*

As it can be observed from Figure 6.9 and Figure 6.10, the working fluid's temperature is dependent and varies according to the air temperature. The temperature of the working fluid varies through an acceptable range, closer to the freezing point compared to the first case study yet still above the freezing point of ethylene glycol. As previously mentioned in the first case study, the temperature of the working fluid before the propane heat pump is higher than the temperature of the ethylene glycol after the dry cooler due to the heat exchangers that deliver heat from other parts of the building to the working fluid. This energy fluctuates with the peak and off-peak hours loads to the heat exchangers. The lowest temperature of the working fluid in the system will be the temperature after the propane heat pump, this temperature ranges between -7 to -18.7°C.

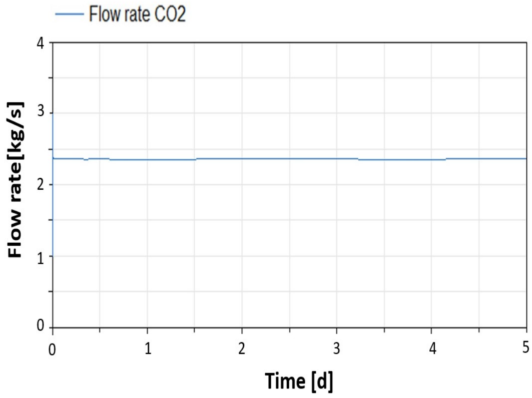
Flow rates and PI controllers: heat exchangers

For the second scenario, the flow rate is regulated in the same manner as in the first scenario: several PI controllers set the mass flow rate to achieve a temperature difference of 5°C in the waste heat recovery heat exchangers and -5°C in the CO_2 heat pump. This can be seen in Figure 6.3.

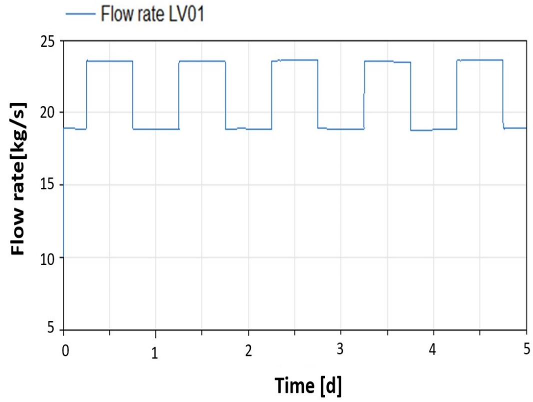
The flow rates of the different heat exchangers vary depending on the time of the day as it did for the previous scenario. Since the load of the CO_2 heat pump remains constant, the mass flow rate through it is constant as well as it was during the first case. This can be observed in Figure 6.11 and the respective flow rates for each unit during peak and off-peak hours can be found in Table 6.4.

Table 6.4: *Flow rates through heat exchanger and CO_2 heat pump case 2*

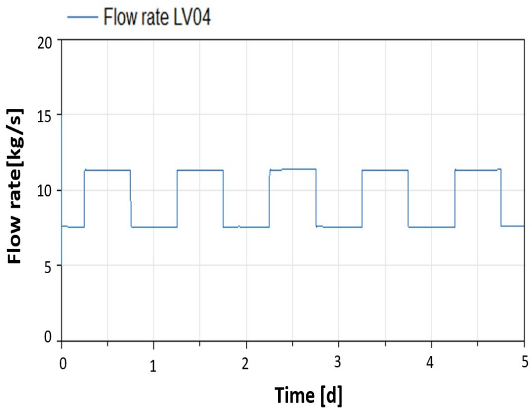
Unit Code	Peak flow rate [kg/s]	Off-peak flow rate [kg/s]
HP-P	121.47	105.92
HP-CO2	2.35	2.35
LVO1	23.57	18.88
LV04	11.31	7.55
LV05	7.54	5.66
LVO7	11.76	7.85
LVO8	5.28	3.97



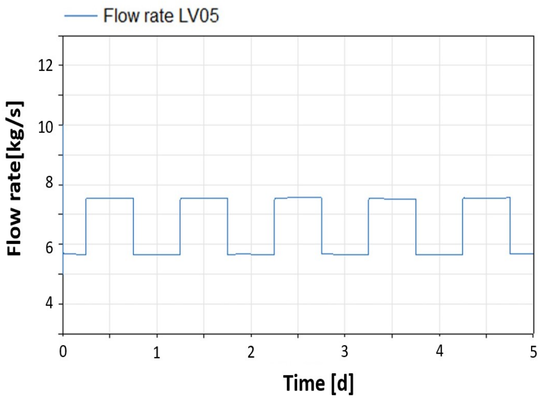
(a) *Flow rate in CO_2 heat pump*



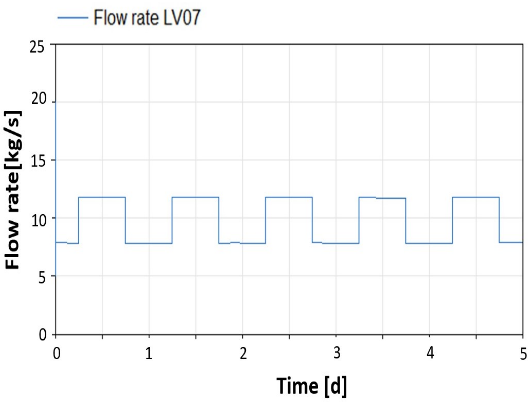
(b) *Flow rate in LV01*



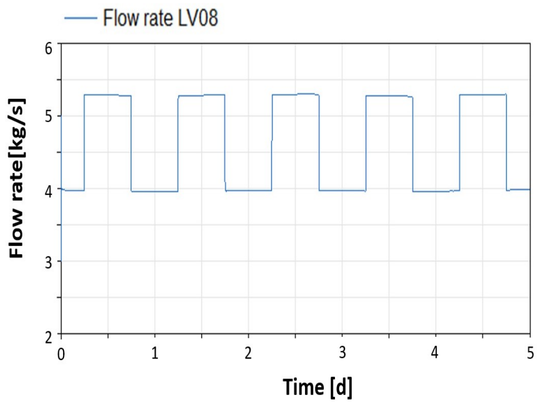
(c) *Flow rate in LV04*



(d) *Flow rate in LV05*



(e) *Flow rate in LV07*



(f) *Flow rate in LV08*

Figure 6.11: *Flow rates regulated by PI controllers case 2*

Flow rate and PI controller: dry cooler

The PI controller in the dry cooler had as set point to match the heating provided by the dry cooler to the heating required by the buildings as explained in previous chapters. The overall performance of the dry cooler can be observed in Figure 6.12. The heating requirement during off-peak hours was 319 kW and 388 kW during peak hours. Note that the scale is inverted in the next figure due to the flow direction of the energy in the system. In Figure 6.13 it can be observed how the air flow rate varied through the five simulated days.

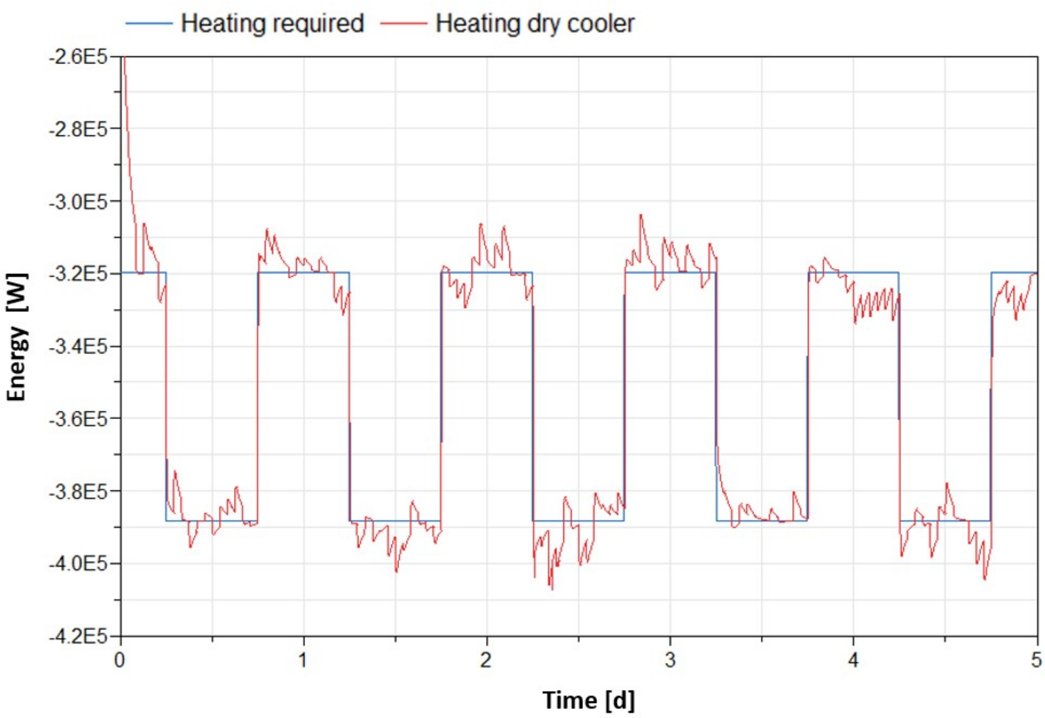


Figure 6.12: *Heating requirements and dry cooler performance for case 2*

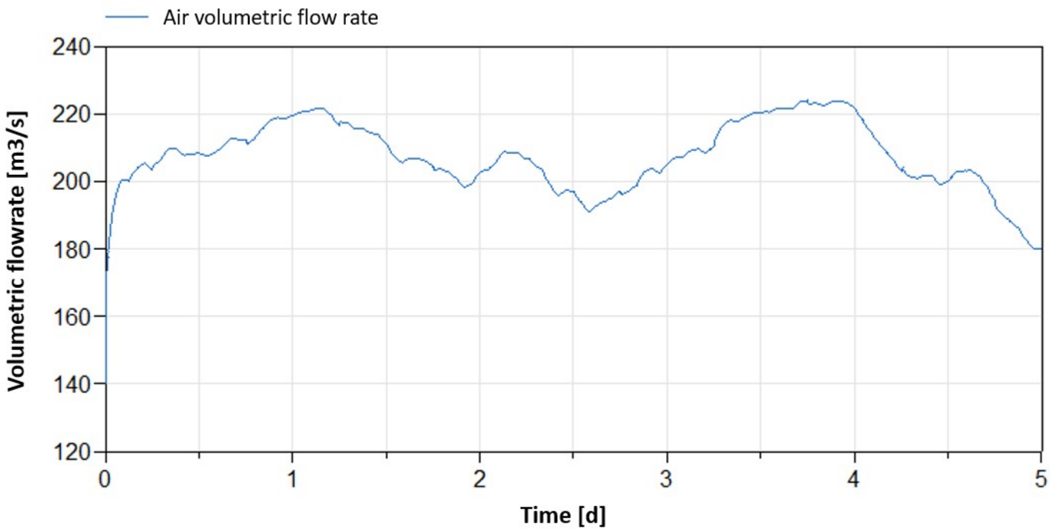


Figure 6.13: *Air flow rate through the dry cooler for case 2*

CHAPTER 7

Discussion

This section will discuss in detail the results obtained from the propane heat pump and the ethylene glycol system model and compare them to the expected results defined by the equipment manufacturer of the proposed system. The objective of this is to determine the accuracy of the simulations and their resemblance to the actual system to be implemented.

7.1 Propane heat pump model

A summary of the results obtained and the expected results can be found in Table 7.1.

Table 7.1: *Comparison table for the propane cycle*

Parameter	Process value	Model	Deviation
Enthalpy after evaporator [kJ/kg]	555.77	559.9	0.7%
Enthalpy after compressor [kJ/kg]	700.35	712.2	1.69%
Enthalpy after condenser [kJ/kg]	340	326	4.11%
Enthalpy after expansion devices [kJ/kg]	340	326	4.11%
Water temperature after condenser [°C]	55	56.56	2.83%
Water temperature after evaporator [°C]	-20	-20.08	0.4%
High pressure propane cycle [bar]	19.95	19.95	
Low pressure propane cycle [bar]	2.19	2.19	
Coefficient of Performance	2.56	2.43	5.01%

Enthalpy and overall cycle

It can be observed that the evaporator in the model complied and even surpassed the requirement of phase changing all the propane mass flow rate with a higher value of superheat than the one originally expected. Similarly, the condenser in the model had a higher subcooling value than the one estimated by the manufacturer. Since a higher subcooling was achieved by the condenser this resulted also in a 4.7% deviation from the expected value for the enthalpy after the expansion valve. These parameters can possibly be improved by modifying the plate heat exchangers that work as an evaporator and condenser in the model. Adjusting all the dimensional data input in the simulation can influence the heat transfer to the propane flow and achieve the superheat and subcooling from the original process.

As mentioned in chapter 5, the dimensions that were used for the simulation were the number of plates, length, and width, yet, other factors such as the wall thickness or pattern amplitude were assumed due to a lack of this information. Another factor that might lead the model to have more superheating and subcooling is the mass flow rate of propane in the cycle. In contrast, the estimated enthalpy of the process after the expansion device was achieved and surpassed slightly by the model, which is not concerning since it is only a 1.69% difference and the temperature nor the pressure at this point were considerably affected by it.

Load and COP

According to the values given by the producer of the equipment, the flow rate of propane at these conditions should be around 0.2691 kg/s, nevertheless, the mass flow rate calculated by the model was 0.305 kg/s. This is not a considerable difference between loads, yet, it could be further improved by working on the PI controllers for the low and high-pressure sides. The heating COP of the original process is 2.56 and the obtained COP for the model was 2.43, very similar to the expected one and within the COP range for an air source heat pumping system [1].

Water temperature

A highly relevant parameter to take into consideration for the propane heat pump system is the temperature at which the process is allowed to heat up the water for the heating of the buildings. This temperature was set around 55°C for the proposed system and the model provides 1.56°C more than the expected value, which is a 5% deviation. The temperature of the water after the evaporator was also very close to the expected value, around 0.08 °C lower than the original value.

Pressure and PI controllers

The high pressure in the propane cycle was achieved thanks to the high-pressure PI controller. The high-pressure PI controller stabilized the system in 13.8 minutes, a value which can still be improved by modifying the k and Ti values of the PI controller. The low-pressure controller managed to stabilize in a little over 3 minutes, which seems to be an acceptable time. These stabilization times, although they can possibly be further improved, are satisfactory for the system since the simulation time was one hour and both high and low-pressure sides of the system are controlled at their set points and maintained there after 14 minutes.

It is relevant to mention that this model controlled the system based on the high and low pressure of the system, nevertheless, since the real system is composed of several circuits as the one modeled, the method of controlling the overall heat pump system will differ. The actual control of the entire propane heat pump system will be likely based on the water temperature after the condenser during the heating mode and the ethylene glycol temperature after the evaporator during the cooling mode. This is done to make sure the system delivers hot water at the desired temperature in winter and that the ethylene glycol does not surpass a temperature range in which it is most efficient. The system will activate and deactivate modules as the heating requirement changes through the day or seasons. The focus of this project was on simulating an individual circuit of the propane heat pump to evaluate its performance, for which the applied PI control method worked appropriately.

7.2 Ethylene glycol model

The following section will discuss in detail the results obtained from the ethylene glycol model for both weather scenarios, possible reasons why these results were obtained, and feasible improvements that can be made to the models.

Working fluid temperature range

As mentioned in chapter 6, the temperature of the working fluid was expected to be maintained above -20°C since the ethylene glycol has a freezing point of around -22°C. Since the lowest temperature in the system will be after the propane heat pump, this temperature was analyzed to determine if the temperature range was adequate. For the first case scenario, this temperature ranged from 5 to -9.93°C. For the second case, the temperature varied between -7 and -18.7°C through the five days simulated. Since the second case used weather conditions with lower temperatures and higher relative humidity than the first case study, it is logical the overall range of the working fluid temperatures will also be lower in this simulation. Nevertheless, it is relevant to mention both case scenarios resulted in temperatures within the expected temperature range according to the manufacturer of the equipment.

PI controllers and flow rates: heat exchangers

The mass flow rate controllers for the heat exchangers and CO_2 heat pump worked appropriately for both cases, reaching their temperature difference set points without reaching their respective upper or lower output ranges. The mass flow rate difference between the first and second scenarios is minimal and is within the flow rate range expected for each heat exchanger set by the manufacturer. The mass flow rate that increased considerably between the first and second scenarios was the working fluid through the propane heat pump. This is because the mass flow rate of working fluid was changed from 41 to 62 kg/s through the dry cooler since the temperature of the second case was considerably colder than the one in the first case.

Dry cooler performance

The dry cooler in both models was capable of delivering most of the energy required during the simulation period. It is relevant to mention that in such cases where the dry cooler was not able to deliver all the energy required, it has been a very small amount of energy that can be easily supplied by the electric boiler in the system. Therefore, its overall performance is compliant and reasonable.

Nevertheless, an important factor to mention is that in both cases, the energy provided by the dry cooler presented a behavior with spikes in it, as opposed to what was seen in other PI controllers of having a smooth output. This can be due to several factors, the main factor being the dry cooler's dimensions. Since the four dry coolers were represented by one single heat exchanger, the dimensions of this equipment were considerably larger and the volumetric air flow rate that passed through it had to be substantially larger as well.

Due to the high volumetric air flow rate, any change in temperature in the air causes a great impact on the energy obtained from it. A feasible solution or improvement to regulate the system better could be to model each dry cooler as an individual heat exchanger and decrease this operating range accordingly in each PI controller. In this manner, the volumetric flow rate going through each heat exchanger will be diminished, the temperature change would cause fewer issues and the system will be closer to the real system installed in Teknobyen. Additionally, the mass flow rate of working fluid through the dry coolers will have to be changed to the value of each unit or it can also be dynamic if the proper PI controller is added to this section. This will be closer to the real system since the operation of the dry coolers is set to be activated as they are needed instead of running constantly and simultaneously. Yet, to simplify the complexity of the system, the approach taken was to simulate the four units in one large equipment.

CHAPTER 8

Conclusion and further work

8.1 Conclusion

It can be concluded from this project that a propane heat pump simulation can be developed using Modelica to resemble an actual heat pump system for the heating and cooling of buildings. The simulations provided an overall adequate representation of a feasible propane heat pump and evaluated the outcomes from a heating mode scenario.

A general cycle of a propane heat pump unit was presented with winter operating conditions with the purpose of better understanding the propane system, its components, and the relevant variables for a suitable operation. Additionally, it was made to determine if the model achieved results similar to the expected ones by the manufacturer of the equipment. The result was a model that simulates the actual process in a considerably accurate manner yet there are some limitations to this model as explained in chapter 5.

The ethylene glycol model was made with the objective to determine how the overall system would behave based on the winter period of the year. Two sets of winter weather conditions were used to evaluate the performance of the system. The ethylene glycol model represents the overall system that provides energy to the propane and CO_2 heat pumps while obtaining energy from different locations in the Teknobyen buildings. The system was simulated for five consecutive days based on two weather data scenarios from 2022. In both scenarios, the model performed adequately and within the expected range of operation, suggesting the installed system will be able to withstand and operate well during the winter time of the year.

8.2 Further work

The simulation of the propane heat pump describes the overall behavior of the actual cycle and facilitates the understanding of the systems and the variables that influence them. Nevertheless, both the propane heat pump and the ethylene glycol models have certain limitations to them due to different factors. The propane heat pump model can be further improved in many manners. An initial improvement could be made by addressing the following recommendations:

- Implement the ethylene glycol mixed with water at 37% in the evaporator instead of water as the working fluid.
- Improve the stabilization time of the PI controllers, particularly the one controlling the high-pressure side.

The ethylene glycol model simulates adequately the system that is installed in Teknobyen, however, further improvement can be made in this model. The areas which could be improved are the following:

- Implement the ethylene glycol mixed with water at 37% in the model instead of water as the working fluid.
- Simulate the dry cooler units separately instead of one large unit.
- Account for the pressure drop in the dry cooler units.
- Model the heat exchangers and heat pumps as heat exchangers instead of tubes with heat port connections.
- Make the flow rate of working fluid through the dry cooler dynamic with a PI controller instead of having it fixed.

Appendices

APPENDIX A

Weather data analysis

The following appendix focuses on the weather data analysis from the 1st of January 2022 until the 31st of December 2022 and its results to determine the hours with frost conditions in Trondheim, Norway during this period.

A.1 Weather data

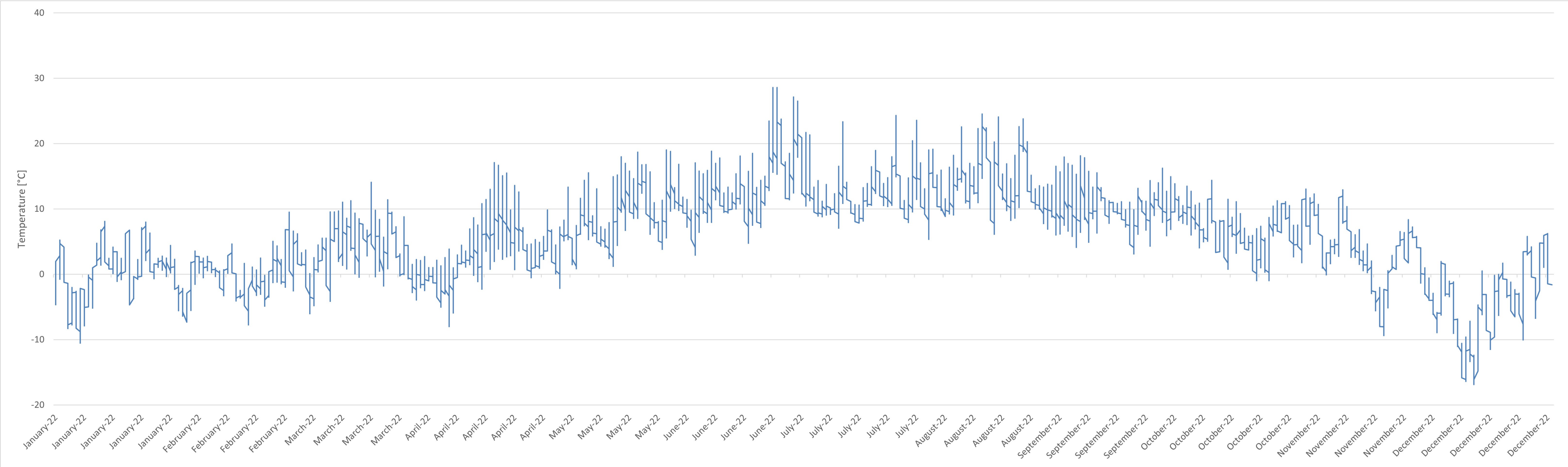


Figure A.1: *Average air temperature in 2022.*

The following graphs will show the frost hours during the four months of the year 2022 that presented the most frost conditions.

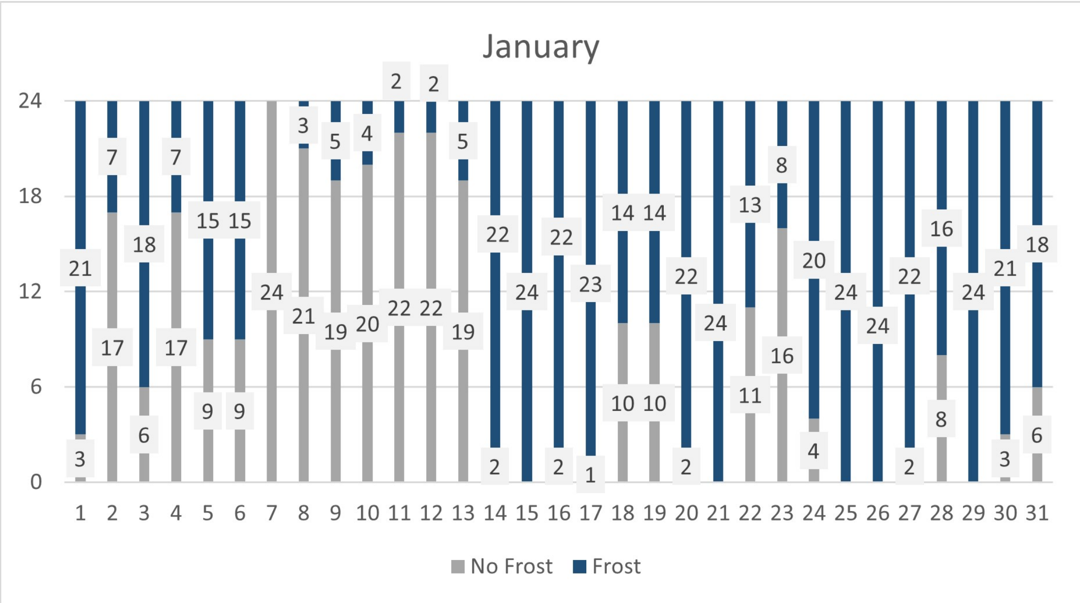


Figure A.2: *Frost hours in January.*

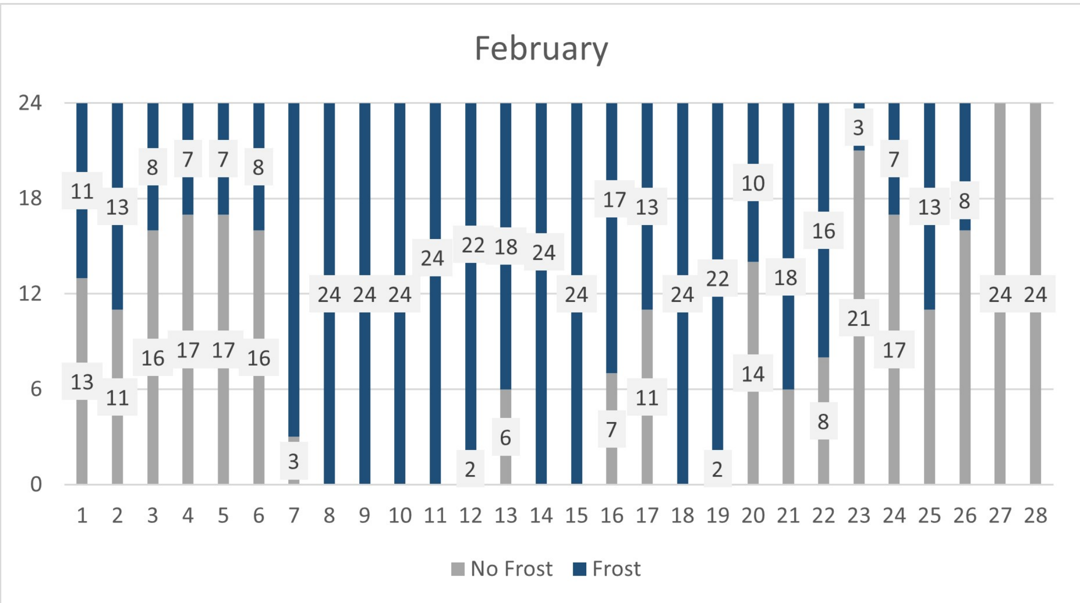


Figure A.3: *Frost hours in February.*

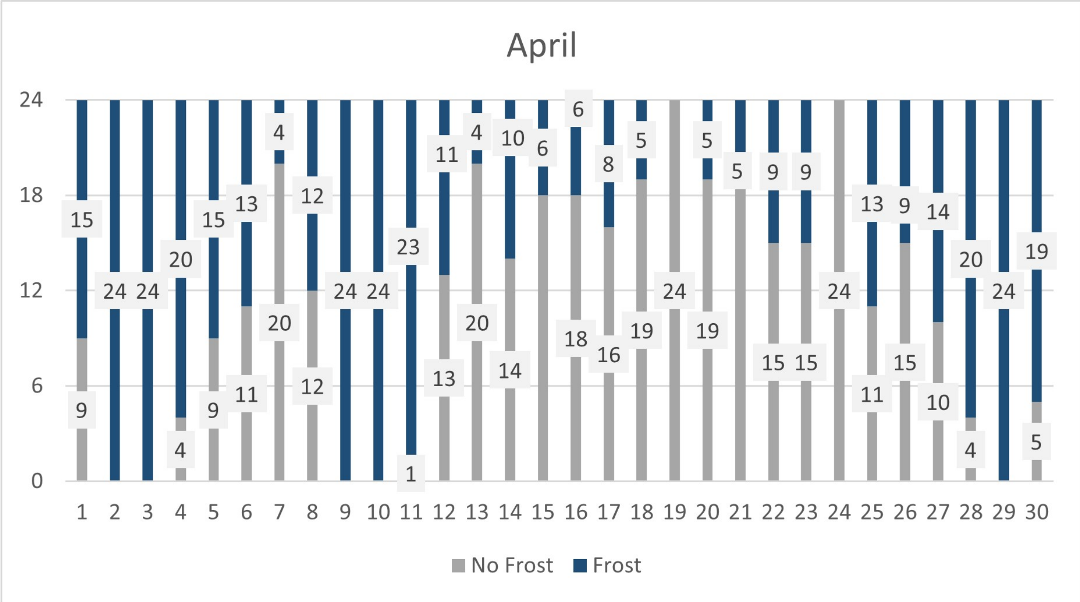


Figure A.4: *Frost hours in April.*

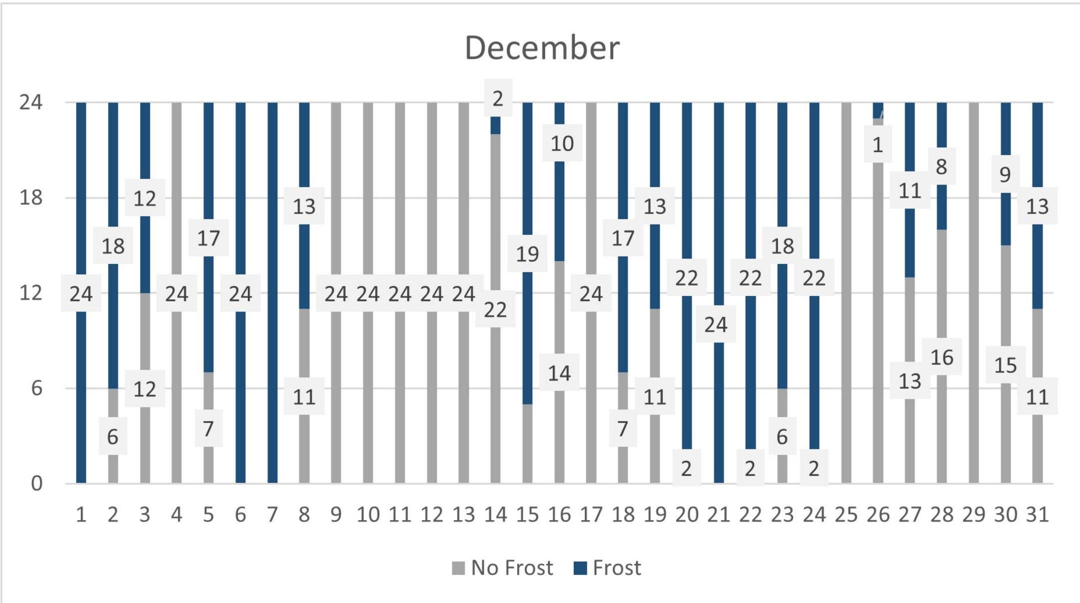


Figure A.5: *Frost hours in December.*

APPENDIX B

Simulation Parameters

The following appendix contains information on the PI controllers for the propane heat pump mode, the weather data used for the ethylene glycol model, and additional information on the dry cooler for the ethylene glycol model in Modelica.

B.1 Propane Heat Pump Model: PI controllers input

Table B.1: *PI control input parameters*

Parameter	Compressor PI	Expansion valve PI
InvertFeedback	true	false
k	2e-2	9e-12
Ti [s]	0.1	20
yMax	100	0.1e-1
yMin	0	50.2e-13

B.2 Ethylene glycol model: Weather data

The following air temperature and relative humidity data were utilized for the first and second case scenario of the ethylene glycol model. This data was retrieved from the Norwegian Meteorological Institute and the Norwegian Broadcasting Corporation [16]. The data was taken from the 8th of January to the 12th of January for the first case and from the 10th of December until the 14th of December for the second case scenario. All the weather data is from the year 2022 and the air temperature used was the average from the maximum and minimum temperature presented by the Norwegian Meteorological Institute.

Table B.2: *Case 1: Weather data part 1*

Value #	Average air temperature [K]
0	0
1	270.8
2	270.65
3	270.65
4	270.5
5	270.65
6	270.65
7	270.7
8	270.45
9	270.4
10	270.35
11	270
12	269.4
13	269.45
14	269.65
15	269.65
16	268.9
17	268.9
18	268.25
19	267
20	266.7
21	265.3
22	266.4
23	267.3
24	268.1
25	268.15
26	269.1
27	268.75
28	269.9
29	268.75
30	269.15
31	270
32	269.6
33	270
34	271.1
35	271.3
36	271.45
37	271.85
38	272.35
39	272.85
40	272.4
41	271.75
42	272.2
43	272.7
44	272.65
45	272.85
46	273.05
47	273
48	272.75
49	272.2
50	271.95
51	272.4
52	272.25
53	272.4
54	272.65
55	272.55
56	270.75
57	268.55
58	268.2
59	268
60	268

Table B.3: *Case 1: Weather data part 2*

Value #	Average air temperature [K]
61	268.85
62	269.55
63	270.45
64	269.95
65	269.7
66	270.7
67	271.75
68	271.85
69	272.5
70	273.8
71	274.1
72	274.15
73	274.55
74	274.95
75	275.55
76	276.15
77	276.5
78	276.55
79	277.05
80	277.5
81	277.6
82	277.8
83	277.9
84	277.85
85	277.75
86	277.55
87	277.15
88	276.9
89	276.5
90	275.45
91	275.85
92	275.85
93	275.2
94	275.2
95	275.25
96	275.2
97	275.75
98	274.6
99	275.2
100	276.5
101	276.1
102	277.3
103	278.25
104	278.2
105	278.95
106	279.1
107	278.5
108	278.45
109	278.25
110	277.95
111	278
112	278.4
113	278.75
114	278.95
115	279.4
116	279.95
117	279.85
118	279.3
119	279.1
120	279.65

Table B.4: *Case 1: Weather data part 3*

Value #	Relative humidity [%]
0	0
1	67
2	63
3	67
4	64
5	64
6	64
7	63
8	65
9	65
10	69
11	68
12	69
13	70
14	67
15	71
16	66
17	68
18	72
19	74
20	80
21	84
22	79
23	75
24	75
25	73
26	71
27	71
28	69
29	77
30	69
31	71
32	69
33	67
34	64
35	65
36	65
37	63
38	58
39	57
40	59
41	60
42	57
43	55
44	55
45	55
46	53
47	53
48	53
49	57
50	55
51	54
52	55
53	53
54	52
55	55
56	64
57	72
58	71
59	72
60	74

Table B.5: *Case 1: Weather data part 4*

Value #	Relative humidity [%]
61	70
62	68
63	65
64	69
65	71
66	65
67	62
68	52
69	53
70	51
71	53
72	54
73	53
74	53
75	51
76	51
77	61
78	57
79	56
80	53
81	54
82	53
83	54
84	55
85	56
86	58
87	61
88	71
89	67
90	73
91	59
92	69
93	59
94	54
95	56
96	57
97	56
98	62
99	59
100	63
101	59
102	55
103	56
104	59
105	67
106	77
107	80
108	84
109	87
110	91
111	92
112	90
113	94
114	96
115	92
116	90
117	92
118	94
119	95
120	93

Table B.6: *Case 2: Weather data part 1*

Value #	Average air temperature [K]
0	0
1	261.25
2	261.4
3	262.6
4	261.1
5	260.9
6	262.05
7	261.15
8	259.85
9	260.35
10	261.15
11	260.85
12	260.6
13	261.1
14	260.5
15	260
16	259.2
17	259.65
18	259.85
19	259.15
20	258.1
21	257.5
22	257.5
23	257.8
24	257.3
25	257.05
26	257.15
27	256.75
28	256.9
29	257.8
30	258.75
31	258.4
32	259.15
33	258.75
34	259.4
35	259.45
36	260.4
37	261.55
38	261.85
39	260.95
40	261.25
41	261.5
42	262.05
43	261.6
44	262.15
45	262.75
46	263.55
47	262.7
48	261.4
49	261.65
50	260.9
51	259.85
52	260.45
53	260.5
54	261.3
55	261.95
56	262.25
57	264.05
58	264.7
59	263.45
60	263.95

Table B.7: *Case 2: Weather data part 2*

Value #	Average air temperature [K]
61	265.2
62	265.95
63	264.6
64	264.45
65	264.15
66	263.45
67	263.3
68	262.8
69	261.3
70	261.5
71	262.2
72	260.95
73	260.45
74	260.75
75	259.95
76	260.05
77	260.6
78	259.55
79	258
80	257.85
81	258.35
82	257.75
83	257.45
84	257.5
85	257.5
86	257.15
87	257.3
88	257.3
89	256.4
90	256.55
91	256.4
92	256.8
93	256.25
94	256.3
95	256.5
96	257.05
97	258.35
98	259.15
99	260
100	260.65
101	261.35
102	262.4
103	263.05
104	262.85
105	262.4
106	262.65
107	263.7
108	262.85
109	261.85
110	262.15
111	262.15
112	262.85
113	263.9
114	265.2
115	265.9
116	266.05
117	266.75
118	267.85
119	268.45
120	268.15

Table B.8: *Case 2: Weather data part 3*

Value #	Relative humidity [%]
0	0
1	89
2	91
3	88
4	88
5	89
6	88
7	86
8	88
9	88
10	87
11	87
12	87
13	88
14	89
15	88
16	88
17	86
18	84
19	86
20	86
21	86
22	86
23	85
24	86
25	85
26	85
27	85
28	86
29	85
30	83
31	85
32	84
33	85
34	83
35	84
36	84
37	79
38	78
39	83
40	83
41	82
42	80
43	82
44	81
45	75
46	73
47	80
48	85
49	83
50	85
51	85
52	85
53	88
54	88
55	90
56	88
57	81
58	83
59	81
60	83

Table B.9: *Case 2: Weather data part 4*

Value #	Relative humidity [%]
61	83
62	79
63	82
64	78
65	83
66	82
67	74
68	76
69	82
70	77
71	78
72	81
73	81
74	82
75	80
76	78
77	81
78	80
79	87
80	85
81	85
82	84
83	85
84	84
85	83
86	81
87	81
88	82
89	82
90	83
91	83
92	82
93	82
94	82
95	81
96	81
97	78
98	81
99	80
100	75
101	75
102	77
103	81
104	80
105	76
106	70
107	73
108	82
109	75
110	78
111	80
112	80
113	80
114	82
115	84
116	85
117	86
118	87
119	85
120	87

B.3 Ethylene glycol model: Dry cooler

This section contains images on the dry cooler of the ethylene glycol system to better understand the parameters necessary for Dymola Modelica for this section and to show images on the actual equipment on the Teknobyen buildings.

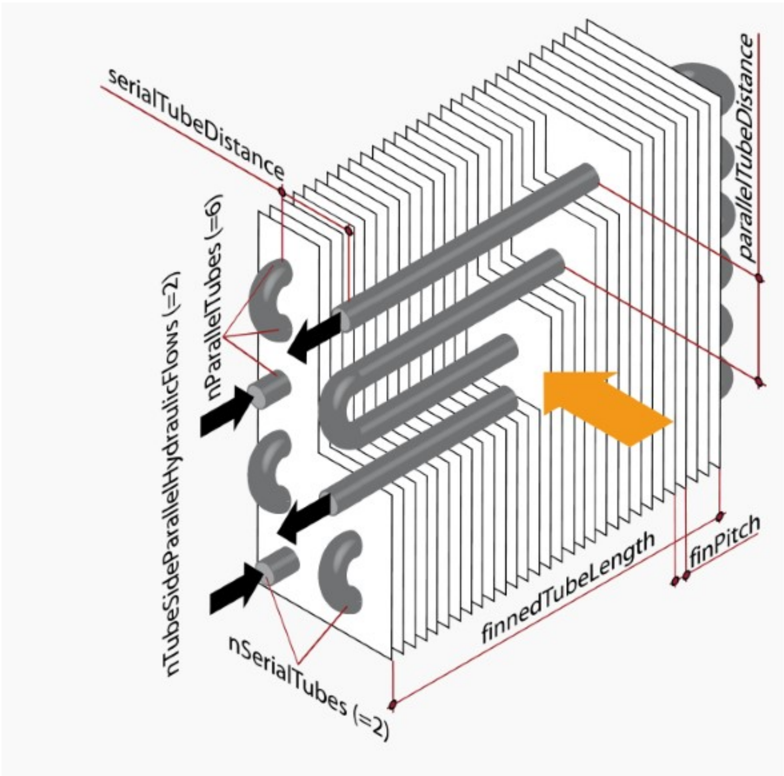


Figure B.1: *Definition of dimensions for Dymola Modelica.*

Figure B.2: *Drycooler on the roof of AG5.*

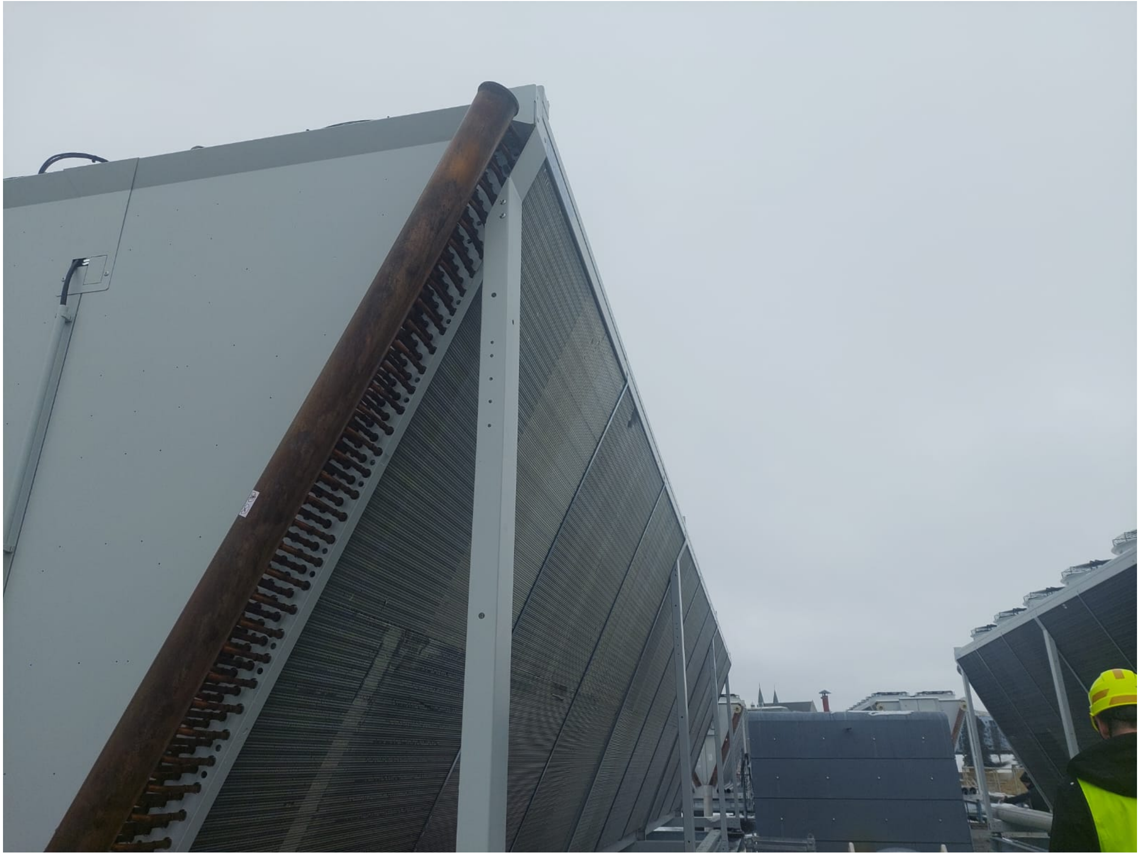


Figure B.3: *Drycooler on the roof of AG5.*

Figure B.4: *Drycooler on the roof of AG5.*

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