Optimizing the heating and cooling system at Teknostallen

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***Abstract*—The Teknobyen project will consist of an energy center for the cooling, heating, and domestic hot water demands of three buildings situated in Trondheim, Norway. The goal of this study is to evaluate the performance of the propane heat pump and the overall ethylene glycol system to be installed in the Teknobyen project during the winter time. To achieve this, simulation models were developed in Dymola Modelica that resemble adequately the systems proposed for the project. The models made represent the propane heat pump unit and the overall ethylene glycol system that will obtain heat from the ambient temperature through a series of dry coolers.**

I. INTRODUCTION

Teknobyen is a multipurpose complex composed of several buildings located in Trondheim, Norway. Their newest project involves the construction of a new building named Teknostall and the creation of a new energy center that will provide heating, cooling, and domestic hot water for three of their buildings; Teknostall, Professor Brochs gate 2, and Abels gate 5. This energy will be provided by a propane heat pump unit, a CO_2, and a large ethylene glycol system that obtains energy from four dry coolers. This project will focus on the propane heat pump unit and a simplified version of the ethylene glycol system.

II. THEORY

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.

A. Heat pump components

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 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 and 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 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].

B. 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]. However, the two main downsides of these heat pumps are that the temperature level of the heat source is lower than other types and that the temperature varies depending on the season [5].

C. 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.The most common parameter to them is the Coefficient of Performance (COP). This parameter will be used to evaluate the system and it is calculated as stated below [1]:

$$COP = \frac{\text{Heat output}}{\text{electrical energy input}} \tag{1}$$

The higher the COP the better the system performs since it provides more energy than the one it is consuming. Air source heat pumps generally have a COP between 2 and 4 [1].

D. Working fluid: propane

Propane 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 [6]. It is an attractive substitute for Chlorofluorocarbons (CFCs) in domestic refrigeration systems given that it has a similar performance with a lower charge and with minor to no changes in the design or system optimization [7]. It has been in use since the 1930's and it is a common refrigerant for heat pumps, air conditioners and commercial refrigeration systems [8].

III. PROPOSED SYSTEM DESCRIPTION

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.

A. General System description

The heating and cooling demands of the buildings will be covered mainly by an air-source heat pump (ASHP). An ASHP unit extracts heat from air at a lower temperature through four dry coolers that will be installed on the roof of the building. Additionally, the glycol stream recovers waste heat from the buildings and delivers it to the propane and CO_2 through the heat pump systems. 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, space heating for AG5, and the defrosting of the dry coolers. The system will operate in two modes: cooling and heating, depending on the weather conditions and the requirements of the buildings. The Piping and Instrumentation Diagram (P&ID) of the system can be found in the Appendix.

B. Propane heat pump

The propane system is composed of two heat pumps with four circuits each. Each circuit includes an evaporator, compressor, condenser, and expansion valve. As mentioned before, this heat pump heats the water that runs through the different buildings for heating purposes and the water temperature before and after the condenser varies according to the cooling or heating mode in which the system is set.

IV. SIMULATED SYSTEM DESCRIPTION

A. Propane Heat Pump Simulation

The general cycle consists of an evaporator, compressor, condenser, expansion valve, volume tank, and two PI controllers, as shown in Figure 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 [9]. 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 [10] in Excel and Coolpack [11]. The result was 19.95 bar for the high-pressure side and 2.19 bar for the low-pressure side. The model will simulate the operation of the heat pump in heating mode, that is, during the cold months of the year.

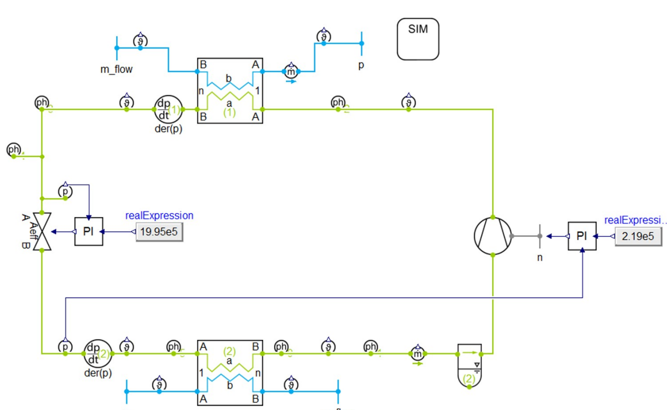


Figure 1: Propane Heat Pump Model

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, made of stainless steel. The heat transfer model was considering a constant alpha of 1,906 $\mathrm{W/m^2K}$ for both sides. Additionally, a quadratic flow-dependent pressure drop model was selected for this model.

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 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 was considering a constant alpha of 1,654 $\mathrm{W/m^2K}$ for both sides. To calculate the pressure drop in this equipment, a quadratic mass flow-dependent pressure drop has been selected.

A regular heat pumping system has an expansion valve to lower the pressure between the condenser and evaporation. 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.

In order to simplify the complexity of this model, some assumptions were made which lead to certain limitations to this simulation. The following 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 in further sections.

B. Ethylene glycol model

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. 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 2 shows the overall diagram of this cycle.

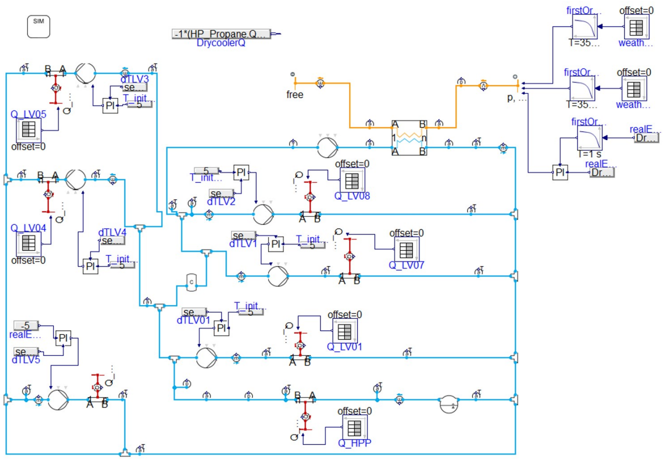


Figure 2: Ethylene glycol model

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 [12].

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 I summarizes the name, 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 II summarizes the units and their total capacity.

Table I: Description of heat exchanger units in ethylene glycol model

Unit Code	Description
HP-P	Evaporator to the propane heat pump
HP-CO2	Evaporator to the CO_2 heat pump
LVO1	Cooling for the building, kitchens, and data centers in TST
LV04	Cooling machine in PB2
LV05	Cooling data centers in PB2
LVO7	Building cooling in AG5
LVO8	Cooling of existing machinery in AG5

Table II: Capacity of heat exchanger units in ethylene glycol model

Unit Code	Capacity [kW]
HP-P	1,854.4
HP-CO2	50
LVO1	1200
LV04	400
LV05	200
LVO7	416
LVO8	140

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. 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 displayed in III. Additional information and sketches on the dry coolers can be found in the Appendix.

Table III: 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.

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 is made with the following equation.

$$Q_{Drycooler} = Q_{HPP} + Q_{HPCO2} + Q_{LV01} + Q_{LV04} + Q_{LV05} + Q_{LV07} + Q_{LV08} \quad (2)$$

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;

It can be understood from the previous equation that 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.

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 around 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. 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 IV: 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 V.

Table V: Energy requirements propane heat pump

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

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. 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 work better for the system since 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, trying 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 2 of the dry coolers. For the second scenario, a fixed load of 62 kg/s was used, equivalent to the load of 3 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.

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 n the following sections.

V. Results

A. *Expected results: Propane heat pump*

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 [10] and Coolpack [11]. In addition, the equipment manufacturing information was taken into consideration as the ideal or expected results to obtain. This information is summarized in Table VI.

Table VI: 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

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 the enthalpy from Table VI, 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 necessary enthalpy to reach after the condenser is 355 kJ/kg without the subcooling.

B. Model results: Propane heat pump

The results obtained from the propane heat pump model are summarized in Table VII for each relevant point mentioned before and Figure 3 depicts the resulting logarithmic-pH diagram for the cycle. This model was simulated for 1 hour time.

Table VII: 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

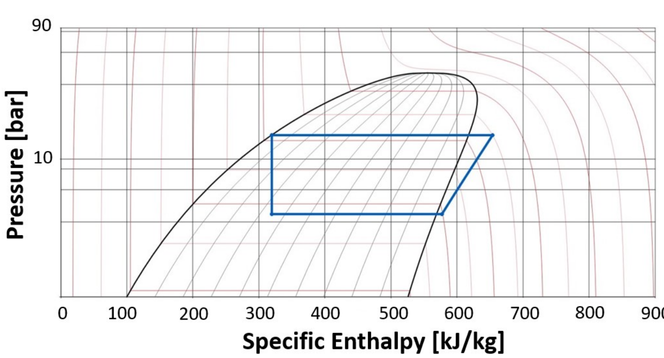


Figure 3: Propane System Results

From Figure 3 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 3 and Table VII it can be perceived that the enthalpy outside the evaporator surpasses slightly the estimated value of 555.77 kJ/kg since 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 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.

C. 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, which 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 in case it is necessary, 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.

D. Model results: 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.

Figure 4 and Figure 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.

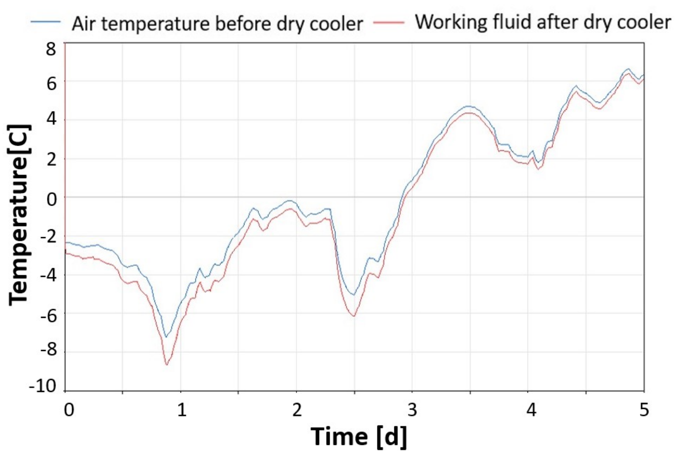


Figure 4: Air and working fluid temperature in dry cooler for case 1

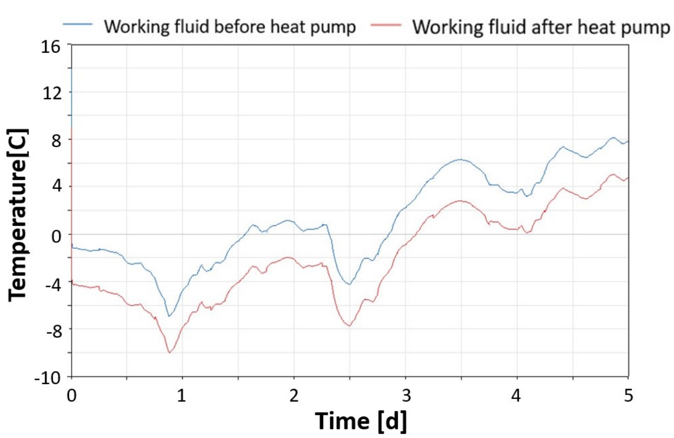


Figure 5: Working fluid temperature in the propane heat pump for case 1

As it can be observed from 4 and 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 other two, 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.

The flow rates of the different heat exchangers vary depending on the time of the day, if it is peak or off-peak time. The flow rate through the CO_2 heat pump is constant since their requirement remains constant as mentioned in previous sections. It is relevant to mention that these controllers achieved the desired set point and additional graphs of them can be found in the Appendix. The respective flow rates for each unit during peak and off-peak hours can be found in VIII.

Table VIII: 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

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. 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.

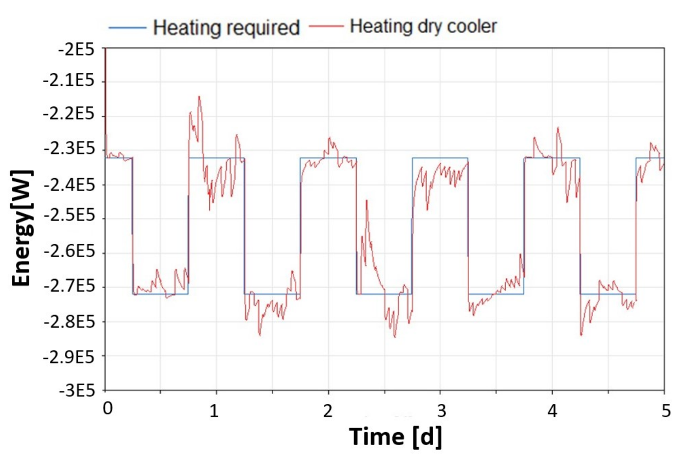


Figure 6: Heating requirements and dry cooler performance for case 1

This controller had a larger range of operation since the heat exchanger represented the four dry coolers in one equipment.

In Figure 7 it can be observed how the air flow rate varied through the five simulated days.

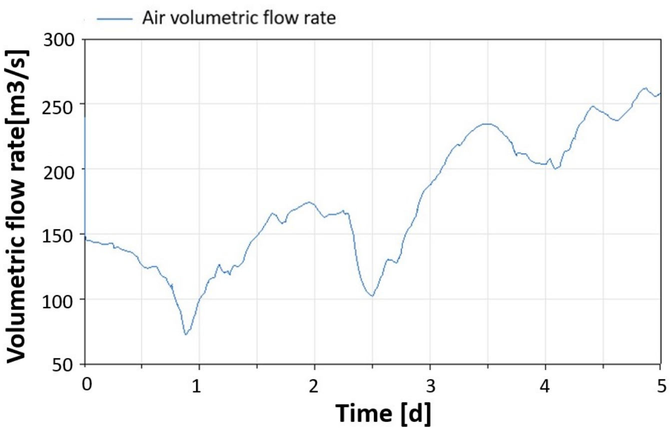


Figure 7: Air flow rate through the dry cooler for case 1

E. Model results: 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%.

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.

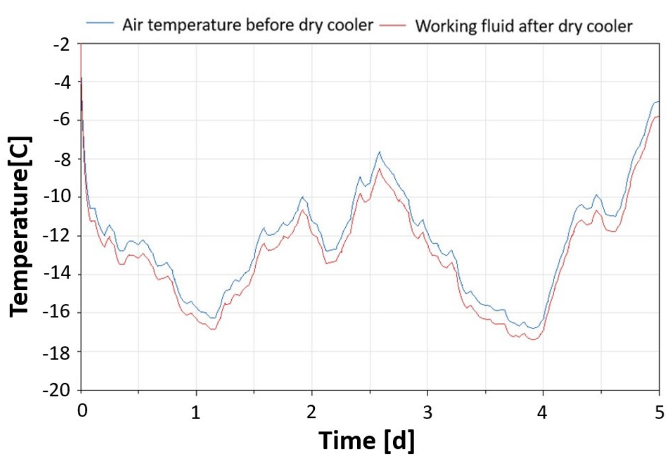


Figure 8: Air and working fluid temperature in dry cooler for case 2

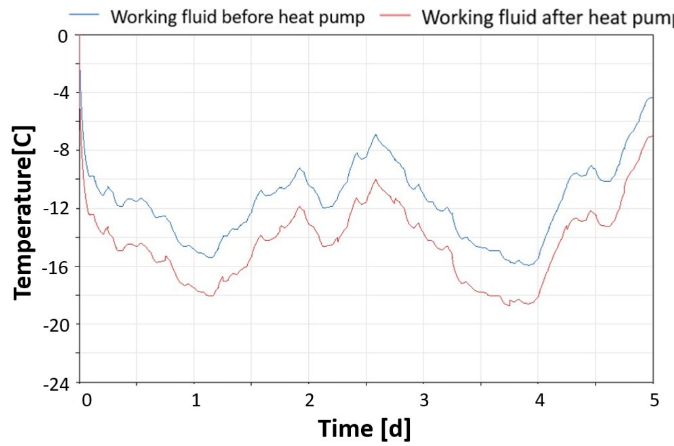


Figure 9: Working fluid temperature in the propane heat pump for case 2

As it can be observed from Figure 8 and Figure 9, 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 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.

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. 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. The respective flow rates for each unit during peak and off-peak hours can be found in IX. The flow rate graphs can be found in the Appendix.

Table IX: 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

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 10. 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.

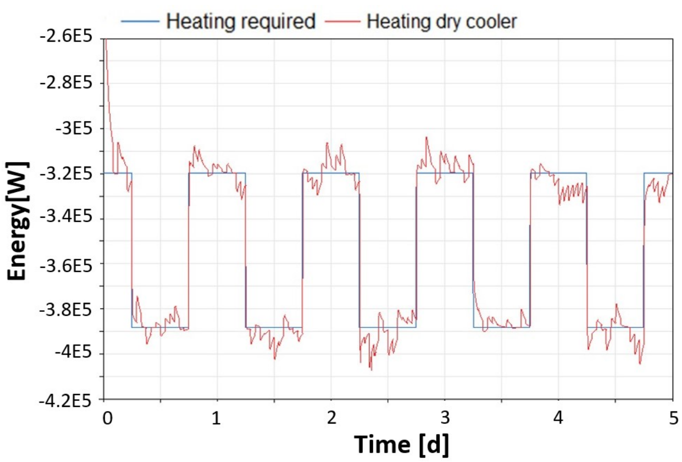


Figure 10: Heating requirements and dry cooler performance for case 2

This controller had a larger range of operation since the heat exchanger represented the four dry coolers in one equipment, the same as the first case scenario. In Figure 11 it can be observed how the air flow rate varied through the five simulated days.

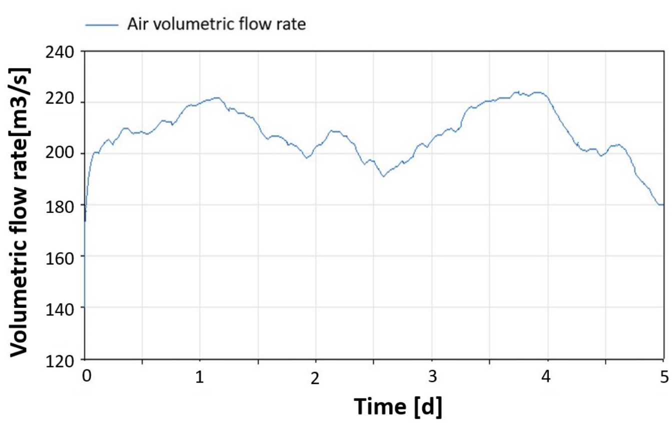


Figure 11: Air flow rate through the dry cooler for case 2

VI. Discussion

A. Propane heat pump model

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

Table X: Comparison table for the propane cycle

Parameter	Process value	Model	Deviation
Enthalpy after evaporator [kJ/kg]	555.77	559.7	0.7%
Enthalpy after compressor [kJ/kg]	700.35	709.8	1.35%
Enthalpy after condenser [kJ/kg]	340	324	4.7%
Enthalpy after expansion devices [kJ/kg]	340	324	4.7%
Water temperature after condenser [°C]	55	55.88	1.6%
Water temperature after evaporator [°C]	-20	-19.87	0.65%
High pressure propane cycle [bar]	19.95	19.95	
Low pressure propane cycle [bar]	2.19	2.19	
Coefficient of Performance	2.56	2.46	3.91%

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.

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 1.35% difference and the temperature nor the pressure at this point were considerably affected by it.

According to the values given by the producer of the equipment, the flow rate of propane at these conditions should be around 0.7 kg/s, nevertheless, the mass flow rate calculated by the model was 0.287 kg/s. This considerable difference between loads 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 2.46, very similar to the expected one and within the COP range for an air source heat pumping system [**?**].

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 0.8°C more than the expected value, which is less than a 5% deviation. The temperature of the water after the evaporator was also very close to the expected value, around 0.13 °C higher than the original value.

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.

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 system will activate and deactivate modules as the heating requirement changes through the day or seasons.

B. Ethylene glycol model

As mentioned in previous sections, 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.

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 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.

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.

VII. Conclusion and future works

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 and 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.

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 period of the year.

References

[1] 1. Dinçer, I.; Kanoğlu, M.(2010). "Refrigeration Systems and Application".
[2] 2. Langley, B. (2002) "Heat pump technology".
[3] 3. Bonin, J. (2015). "Heat pump planning handbook".
[4] 4. Vieira, A.; Stewart, R.; Beal, C. (2015). "Air source heat pump water heaters in residential buildings in Australia: Identification of key performance parameters".
[5] 5. Liu, X.; Lau, S; Li, H. (2014). "Optimization and analysis of a multi-functional heat pump system with air source and gray water source in heating mode".
[6] 6. Eckert, M; Kauffeld, M; Siegismund, V. (2022)."Natural Refrigerants: Applications and Practical Guidelines".
[7] 7. James, R; Missenden, J. (1992). "The use of propane in domestic refrigerators".
[8] 8. Palm, B. (2008). "Hydrocarbons as refrigerants in small heat pump and refrigeration systems – A review".
[9] 9. (2021). "Modelica". https://modelica.org/modelicalanguage.html
[10] 10. (2020). "Welcome to CoolProp". http://www.coolprop.org/
[11] 11. (2022). "CoolPack". https://www.ipu.dk/products/coolpack/
[12] 12. Skålin, R.; Haugen, V.F.; Jensen, I.S. (2022). "Trondheim". https://www.yr.no/en/statistics/table/1-211102/Norway/Tr%C3%B8ndelag/Trondheim/Trondheim,

VIII. Appendix

A. General system description

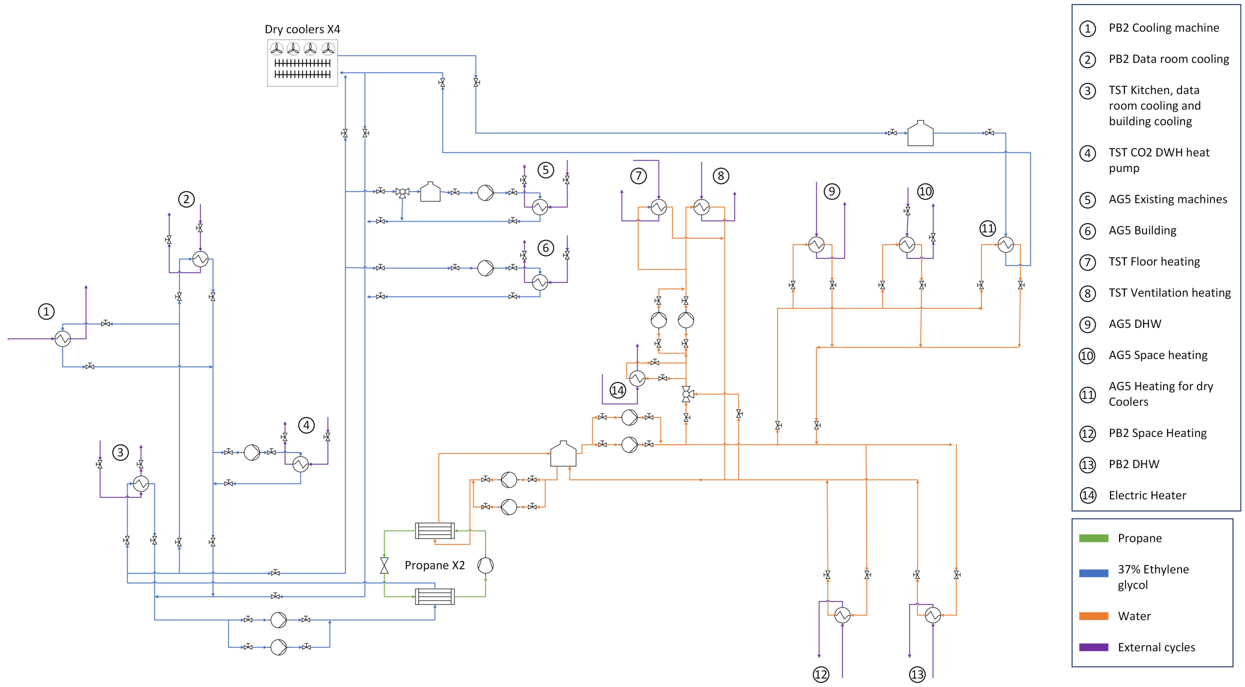
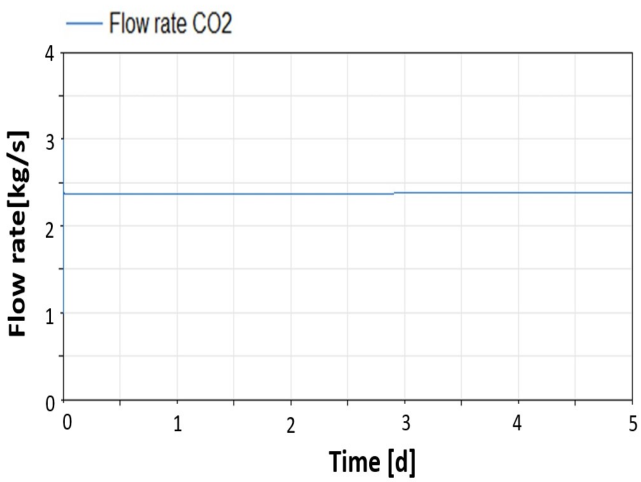
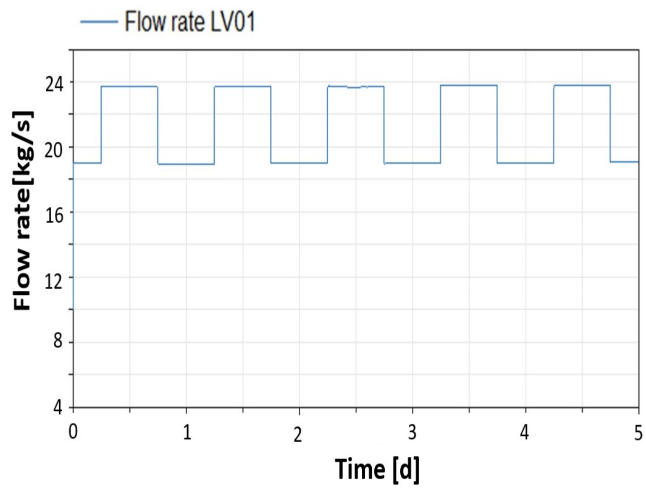


Figure 12: Process Diagram for Teknostallen

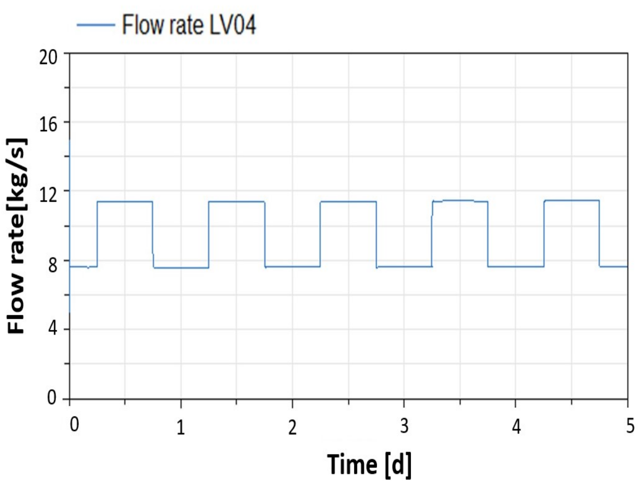
B. Model results: Ethylene glycol system



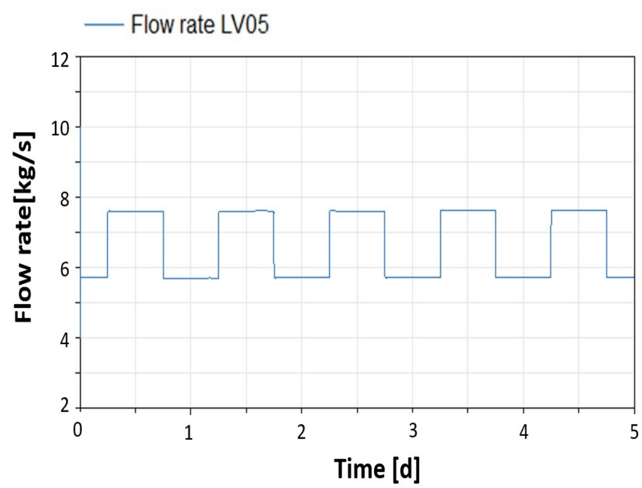
(a) Flow rate in CO_2 heat pump



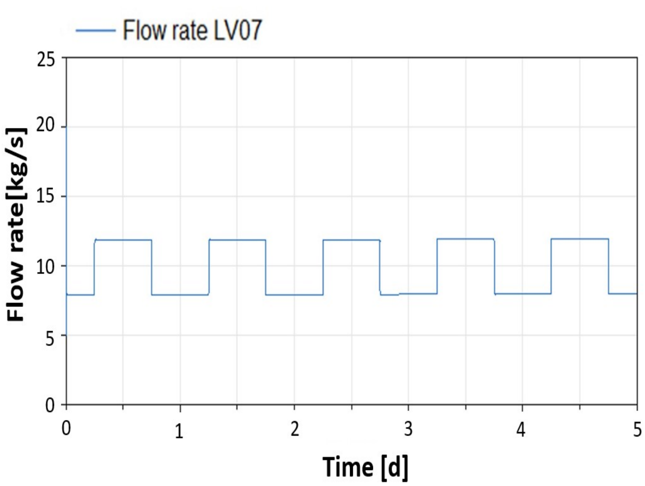
(b) Flow rate in LV01



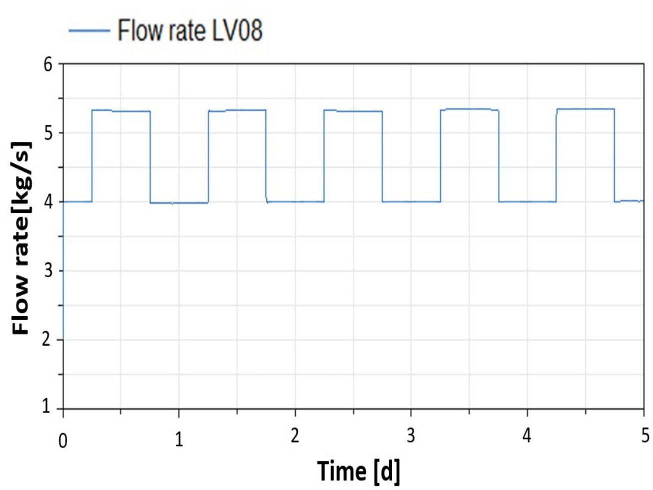
(c) Flow rate in LV04



(d) Flow rate in LV05

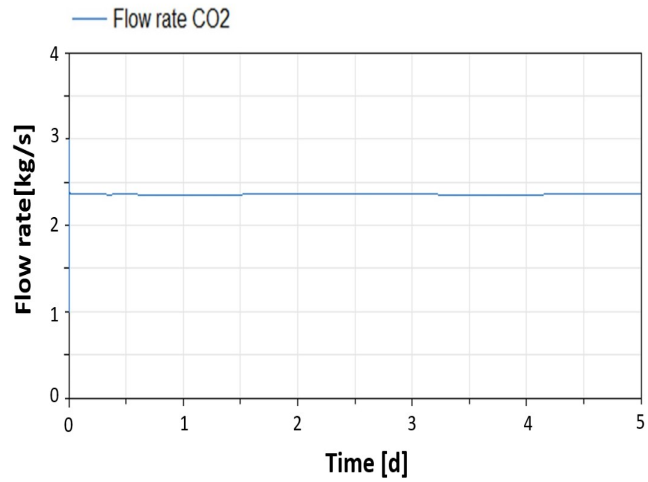


(e) Flow rate in LV07

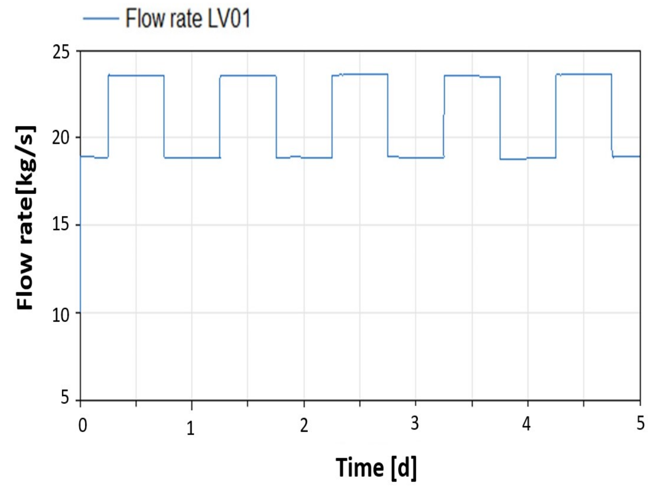


(f) Flow rate in LV08

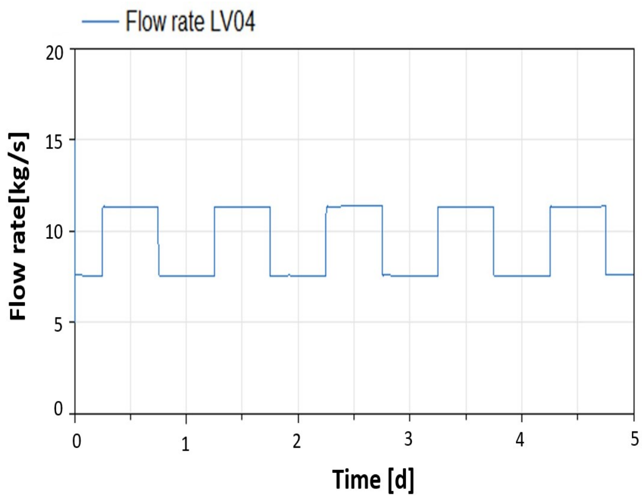
Figure 13: Flow rates regulated by PI controllers case 1



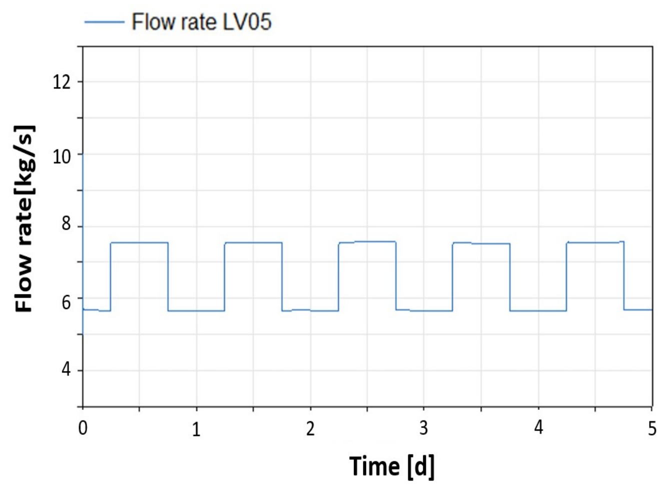
(a) Flow rate in CO_2 heat pump



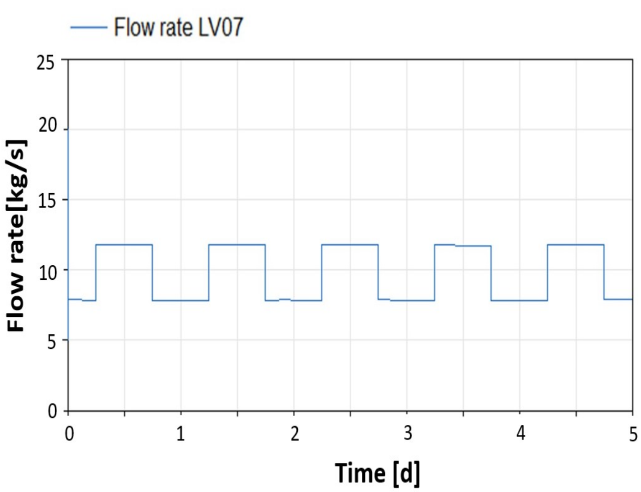
(b) Flow rate in LV01



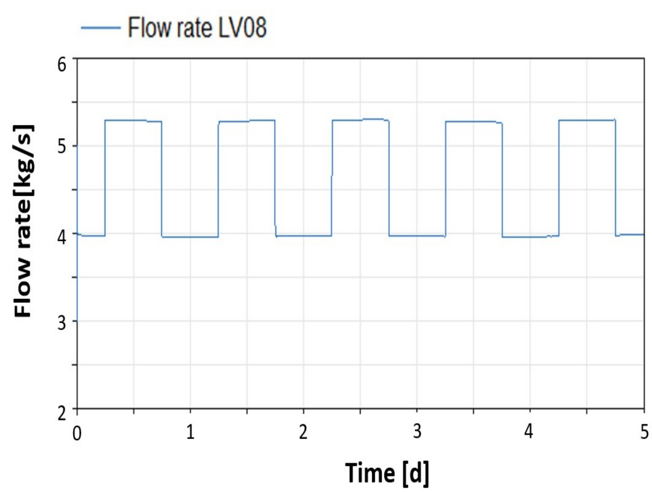
(c) Flow rate in LV04



(d) Flow rate in LV05



(e) Flow rate in LV07



(f) Flow rate in LV08

Figure 14: Flow rates regulated by PI controllers case 2