

HØGSKULEN PÅ VESTLANDET

BACHELOR THESIS

**Control and system engineering to optimize
energy transfer from natural refrigerant
heat pump systems**

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*“I would like to thank everyone who have supported me through my education.
Especially all the enthusiastic teachers and lecturers who have provided me with a solid education.
I can now call myself an engineer.”*

Daniel Vikøyr

“My dearest thanks go to all the people who have helped me in this journey - parents, teachers and friends.”

Matiss Macs

Abstract

Background: Heat pump is a popular alternative to conventional heating sources and projections show that increased adoption is to be expected. Chlorofluorocarbon refrigerants (R-12, R-22, etc.) used in heat pumps are strictly regulated because of their harmful effect on the ozone layer. Hydrofluorocarbons (R-134a, etc.) are commonly used replacements. However, they are in the process of being phased out because of their high global warming potential. This has resulted in increased awareness and research about use of natural refrigerants, such as R-290 (propane), in heat pumps.

Objective: Heating systems are typically designed from HVAC engineer perspective. Our contracting entity is interested in seeing analysis of R-290 heat pump system from the perspective of automation engineering. The main points of focus being control aspects of energy transfer and savings. We research suitable simulation software (TRNSYS) that could verify our hypothesis and could be utilized by Rambøll's engineers for future designs.

Methods: We explain the foundational theory that helps to highlight the advantages and the disadvantages regarding R-290 refrigerant based heat pump systems. We analyse the system itself from a theoretical point of view and its suitability as a replacement for more traditional systems. Furthermore, after defining the system we suggest potential modifications from the perspective of an automation engineer. We employ TRNSYS software to both analyse the usefulness of the software itself and our proposed scenarios. Finally, we touch the subject of control engineering by analysing various properties of heat pump condenser's state space representation.

Results: We conclude that, besides the environmental benefits, the working properties of R-290 are similar to R-22, thus an existing heating system design principles can be applied. Disadvantages are related to increased complexity with regards to safety precautions. We present the necessary control hardware. It is concluded that TRNSYS simulation software can be successfully used for our purposes, and we discuss the potential issues that need to be addressed if it were to be used by Rambøll. Our suggested system modifications are analysed. Finally, as an extra supplement, a MATLAB program was created to showcase potential controllability breakpoints and system dynamics.

Conclusion: The initial objectives of this thesis were met. In the process we have proposed answers to central questions regarding the viability of R-290 use and the ways how to design such a system for additional energy savings and optimal control. Some of our proposed solutions had energy savings up to 20%. However, it has to be noted that results are simulation based and they have not been empirically verified. This could be a basis for further research utilising a physical test rig.

We feel that through the many hours of research, programming, mathematical and system analysis we have immersed ourselves in the intricacies of heat pumps. Yet we still have to acknowledge how complex and overwhelmingly large the topic is. Nevertheless, our aim is to give a concise and comprehensible explanation. Whether you are our supervisor, a Rambøll employee, our classmate or just someone curious, we hope you will enjoy the work that we have put into this thesis.

Declaration of Authorship

Daniel A. M. Vikøyr and Matiss Macs, declare that this thesis titled, “Control and system engineering to optimize energy transfer from natural refrigerant heat pump systems” and the work presented in it are their own. We confirm that:

- This work was done while in candidature for a bachelor’s degree at Western Norway University of Applied Sciences
- Where we have quoted from the work of others, the source is always given. With the exception of such quotations, this thesis is entirely our own work.
- We have acknowledged all main sources of help.
- Where the thesis is based on work done by ourselves jointly with others, we have made clear exactly what was done by others and what we have contributed ourselves.

Division of Labour

Both authors, Daniel A. M. Vikøyr and Matiss Macs have equally contributed to this thesis. For more efficient use of time Daniel A. M. Vikøyr has focused on tasks regarding mathematical model and MATLAB. Likewise Matiss Macs has focused on simulation and TRNSYS. Nevertheless, every chapter has been written in cooperation between the authors and both have devoted equal time to this thesis and its content.

Signed: 

Date and place: *June 1, 2021* *Bergen, Norway*

Contents

Acknowledgements	II
Abstract	III
Abbreviations	VIII
Nomenclatures table	IX
1 Introduction	1
1.1 Contracting entity	1
1.2 Problem description	1
2 Specification of requirements	1
3 The Fundamental Theory	2
3.1 Cold side	2
3.1.1 Air / atmosphere	3
3.1.2 Seawater	3
3.1.3 Groundwater	3
3.1.4 Gray-water	3
3.2 Heat pump	4
3.2.1 Heat exchanger	4
3.2.2 Tube heat exchanger	4
3.2.3 Plate heat exchanger	5
3.2.4 Microchannel heat exchangers	5
3.2.5 Propane	6
3.2.6 R-290	8
3.2.7 CO ₂	9
3.2.8 Compressor	10
3.3 Warm side	11
3.3.1 Ventilation	11
3.3.2 Underfloor/ Outside under-road heating	11
3.3.3 Radiators	11
3.3.4 Electric boiler used in the temperature control	11
3.3.5 Thermal and electric boiler	12
3.4 Control hardware	12
3.4.1 PLC	12
3.4.2 Sensor system	12
3.4.3 Valve control	13
3.4.4 Propane safety	13
4 Potential solutions	14
4.1 Description of possible solutions	14
4.2 The approach for a physical solution	14
4.3 Proposed changes of the system	16
4.3.1 First proposed modification: recycling water loop	16
4.3.2 Second proposed modification: continuous flow for the recycling water loop	18
4.3.3 Third proposed modification: Parallel piping	18
4.4 Problem analysis conclusion	18
5 Defining The System	19
5.1 Mathematical models	19
5.2 Mathematical model ground source heat pump system	23
5.3 Mathematical model air-conditioner/ radiators	24

6	Testing and Simulation	25
6.1	Simulation - TRNSYS	25
6.1.1	Modularity and cost	26
6.1.2	Learning	26
6.1.3	Interface	26
6.1.4	Custom components	27
6.1.5	Simulation of variable speed heat pumps (VSHP)	27
6.1.6	Control options in TRNSYS	28
6.2	Simulation process	28
6.2.1	Heat pump	28
6.2.2	Additional components	30
6.2.3	Characteristics of a typical system	30
6.3	Simulation of proposed solutions	32
6.3.1	Results	33
6.3.2	Power	34
6.3.3	COP	35
6.3.4	Heat transfer rate	36
6.3.5	Tap-water temperature	37
6.3.6	Floor temperature	37
6.3.7	Radiator temperature	37
7	Control system engineering	39
7.1	Discussion about the mathematical system	39
7.2	Regulation of the system	39
7.3	The Matlab script	39
7.4	Proposal of potential thesis "Compressor speed control with continuous feedback regulation."	40
8	Discussion	41
9	Conclusion	41
	Bibliography	42
	Appendix A - Tables & Diagrams	44
	Appendix B - TRNSYS simulation layouts	45
	Appendix C - Chapter 7 supplement - MATLAB showcase	49
	State-Space	49
	Transfer function	50
	Zeros and poles	51
	Observability	51
	Reachability	52
	Controllability	52
	Determinants, eigenvalues and trace	52
	Appendix D - Matlab code	53

List of Figures

1	Schematic presentation - working principle of a heat pump system. The whole system is composed of 3 parts, the heat pump itself, the cold side (left), the heat pump and the warm side (right). [3](Zheng, 2018)	2
2	Boyle's, Charles's and Avogadro's gas laws [5](chem.libretexts, 2017)	4
3	Representation of how a tube heat exchanger works. [6](Comsol, 2021)	5
4	How the inside of a microchannel heat exchanger looks [7](Trumony, 2021)	5
5	Propane phase diagram [8](EngineeringToolbox, 2021)	6
6	Thermal conductivity of propane in different phases and temperatures [9](EngineeringToolbox, 2021)	7
7	Properties of different refrigerants	7
8	Reference: [12](Park and Jung, 2008)	8
9	Carbon dioxide phase diagram - Reference: [14](EngineeringToolbox, 2021)	9
10	Various types of compressors used in refrigeration	10
11	Electric heater	11
12	System to pasteurize and heat the warm water.	12
13	Cabinet housing PLC units for a propane based heat pump system. Picture taken from the field work at our reference system location.	12
14	Safety related components. Pictures taken at the location of our reference system.	13
15	Two ENRAD Chiller VKA. Picture courtesy of [22](Galaasen, 2019)	15
16	Graphical user interface for manual observation and control of individual rooms in our reference facility.	16
17	Part of the piping and instrumentation diagram (P&ID) that showcases the floor and radiator heating where the marked temperatures illustrate where we propose the series connection.	17
18	Basic graphs for concept understanding of entropy and enthalpy in the heating/cooling cycle.	19
19	(Left) Vapor pressure and latent heat, (Right) Liquid density and compression ratio [23](Paradeshi, Mokka, and Jayaraj, 2017)	20
20	The heat capacity of water	22
21	The super heated (system 1) and the phase shifting system (system 2)	24
22	Air's isochoric and isobaric heat capacity [26](EngineeringToolbox, 2021)	24
23	TRNSYS interface	25
24	Component creation in TRNSYS	27
25	Heat pump temperatures	29
26	Heat pump power usage characteristics	29
27	Additional components	30
28	Properties of the radiator component	30
29	Heat pump characteristics	31
30	Temperature comparison	31
31	TRNSYS component layout for the proposed changes	32
32	Heat pump inlet, outlet and electric heater outlet temperatures	33
33	Power consumed and delivered by the heat pump and the power consumed by the electric heater	34
34	Coefficient of performance	35
35	Heat transfer rates	36
36	Temperature lost heating tap water	37
37	Temperature lost heating floors	38
38	Temperature lost heating radiators	38
39	Illustration of the conditional interface	40
40	The magnitude and phase of the transfer function of the mass flow rate from compressor to the output temperature from compressor	40
41	Standard system	45
42	Series system	46
43	Series with bypass	47
44	Parallel with heated tap water	48
45	A-matrix	49

46	B-matrix	49
47	C-matrix	50
48	D-matrix	50
49	Transfer function	50
50	TF zero/pole representation form. The values of "s" in the brackets in the numerator equal the zero points. The "s" values that gives zeroing in the denominator are the poles of the system.	51
51	The observability matrix	51
52	Eigenvalues	52
53	Trace of the system	52

List of Tables

1	The Specific heat values of water (H_2O)	22
2	The specific heat values of air	24
3	The properties of Propane (C_3H_8)	44

Abbreviations

ODP	Ozone Depletion Potential
GWP	Global Warming Potential
COP	Coefficient Of Performance
PLC	Programmable Logic Controller
VSHP	Variable Speed Heat Pump
RPM	Revolutions Per Minute
P&ID	Piping and Instrumentation Diagram
PWM	Pulse Width Modulation
PID	Proportional Integral Derivative (control principle)
R-22	Chlorodifluoromethane
R-134a	Norflurane
R-290	Propane
R-717	Ammonia
R-744	Carbon dioxide (CO_2)
R-1270	Propylene

Nomenclatures table

Δ	energy difference
\dot{W}	Net work/ energy (kW)
\dot{W}_{net}	Net compressor power (kW)
\dot{W}_{comp}	Compressor power (kW)
\dot{m}_r	Mass flowrate of the refrigerant (kg/s)
\dot{m}_{water}	Mass flowrate of the return water on the warm side system (kg/s)
\dot{m}_{brine}	Mass flowrate of the return brine/ water on the cold side system (kg/s)
\dot{m}_{air}	Mass flowrate of the air (kg/s)
H	Enthalpy (kJ/kg)
$H_{1,2,3,4}$	Enthalpy in point 1 to 4 (kJ/kg)
H_r	The enthalpy of the refrigerant (kJ/kg)
H_{comp}	Refrigerant enthalpy out of the compressor (kJ/kg)
H_{comp_isen}	Refrigerants Isentropic enthalpy out of the compressor (kJ/kg))
H_{cond}	Refrigerant enthalpy out of the condenser (kJ/kg)
H_{expa}	Refrigerant enthalpy out of the expansion (kJ/kg)
η_{elec}	Electrical engine efficiency
η_{mech}	Mechanical efficiency of the compressor
η_{comp_isen}	Isentropic efficiency of compressor
\dot{Q}	Energy capacity (kW)
\dot{Q}_r	Heat transferred from refrigerant (kW)
\dot{Q}_{cond}	Condenser Energy capacity (kW)
\dot{Q}_{evap}	Evaporator Energy capacity (kW)
\dot{Q}_{HP}	Heating capacity of the Heat Pump (kW)
T_{water_inn}	Temperature return water on the warm side system (J/(kg * °C))
T_{water_out}	Temperature of the water going out of the condenser (J/(kg * °C))
T_{brine_inn}	Temperature return brine/ water on the cold side system (J/(kg * °C))
T_{brine_out}	Temperature of the brine/ water going out of the evaporator (J/(kg * °C))
T_{cond_inn}	Temperature of the refrigerant going into the condenser system (J/(kg * °C))
T_{cond_out}	Temperature of the refrigerant going out of the condenser system (J/(kg * °C))
T_{AC_AR}	Temperature air going into the system (J/ (kg * °C))
T_{AC_AO}	Temperature of the air going out of the air conditioner (J/ (kg * °C))
ΔT_{ld_cond}	logarithmic temperature differences of condenser (°C)
ΔT_{ld_evap}	logarithmic temperature differences of evaporator (°C)
C_{p_water}	The specific heat of water (J/(kg * °C))
C_{p_brine}	The specific heat of brine/ water (J/(kg * °C)) ($C_{p_water} \approx C_{p_brine}$)
C_{p_air}	The specific heat of Air (J/ (kg * °C))
K_{cond}	Total heat transfer coefficient of condenser (kW/ (m ² * °C))
K_{evap}	Total heat transfer coefficient of evaporator (kW/ (m ² * °C))
A_{cond}	Total heat transfer area of condenser (m ²)
A_{evap}	Total heat transfer area of evaporator (m ²)
F_{cond}	Correction coefficient of condenser (=1)
F_{evap}	Correction coefficient of evaporator (=1)

1 Introduction

The motivation for this task stems from the close attention from the authorities and the general public to environmental problems and climate change. There is an increasing interest in refrigerants possessing low global warming potential (GWP) which describes the greenhouse gas effect of a specific compound compared to a metric ton of CO₂ over a specific time-frame. Because of this, there is an ongoing search for suitable natural refrigerant alternatives to the environmentally damaging refrigerants used today.

Hydrocarbons, as natural refrigerants, have shown good material compatibility, desirable thermodynamic and transport properties, which can also reduce the direct effects of climate change, by eliminating the presence of greenhouse gasses that could escape heat pump systems. Propane is a hydrocarbon that is regarded as a good refrigerant alternative, because the physical properties of propane are close to those of chlorodifluoromethane (R-22) which has been the market standard for many years. Heat pump and air conditioning equipment has a direct effect on the anthropogenic global warming due to atmospheric emissions.

However, by replacing the refrigerant to a low GWP substance, we are able to sharply reduce the GWP and put the ozone depletion potential (ODP) to zero. To give some context, the GWP over 100 years is lower than 3 for propane, while it is equal to 2100 for R-410A and around 1400 for R-134a, which are sold as environmentally friendly alternatives in the phaseout of R-22 [1](ScienceDirect, 2021) and [2](Del Col, Bortolato, and Bortolin, 2014).

1.1 Contracting entity

Rambøll is a multidisciplinary consulting group with a focus on engineering. Founded in Denmark in 1945 it has increased its presence to 35 countries and encompasses 16500 people globally. Norwegian subdivision of Rambøll has 15 offices and 1500 employees.

The majority shareholder of the Rambøll group is The Rambøll Foundation. The remainder are owned by the employees of Rambøll. Among the services delivered by Rambøll there are solutions related to buildings, transport, water, environment and energy.

1.2 Problem description

The contracting entity wants to know the advantages and disadvantages of propane and CO₂ based systems compared to the more traditional heat pump systems that use refrigerant like R-22 or other environmentally hazardous refrigerants.

We have looked at the setup of a new system(s) and existing systems, and the changes that the new refrigerants introduce. Are we able to make a system comparably stable, fast and as efficient as the current ones utilizing environmentally damaging refrigerants? Additionally, at what cost of performance of the system. We as automation engineers will try to compensate the drawbacks of the new refrigerants by using system and control engineering, in the pursuit of making a sustainable system.

2 Specification of requirements

We have a fairly open assignment, with just a few guidelines/ requirements. We are going to immerse ourselves in how propane will operate in a heat pump system. What changes are going to occur that a traditional system would not have and what are the drawbacks? We are mainly going to focus on the output of the heat pump and how we with control system engineering and general system design can make an optimal system given the new conditions.

As per the contracting entity's request, we are going to focus on heat pump systems around the 100 kW range. Such systems are used in medium sized public institutions or shared housing units. With the guidance of our contracting entity we have conducted field work to establish a reference point for our simulation work. It was performed at an existing physical facility in western Bergen where a propane based heat pump system is utilized to heat a school and its complimentary buildings. We have access to detailed plans of this system, as well as the system includes main points of our interest – floor, radiator and tap water loads; control hardware and several control safety mechanisms. Therefore, it is going to constitute the basis for our requirements of system design values used in simulation and control analysis work.

3 The Fundamental Theory

This chapter gives a concise theoretical foundation, needed to understand the functional principles of a heat pump. It also briefly discusses the subject of refrigerants, specifically - how does R-290 characteristics compare to other working fluids. Finally, the chapter describes existing control hardware solutions.

For the sake of simplicity, we are referring to the “cold side” as to the side which is extracting heat from a thermal reservoir and transferring it to the “warm side” which corresponds to the heating system of a building.

In simpler terms – we want to transfer the energy from the outside into the building. From now on we are referring to inside of the building as the “warm side”.

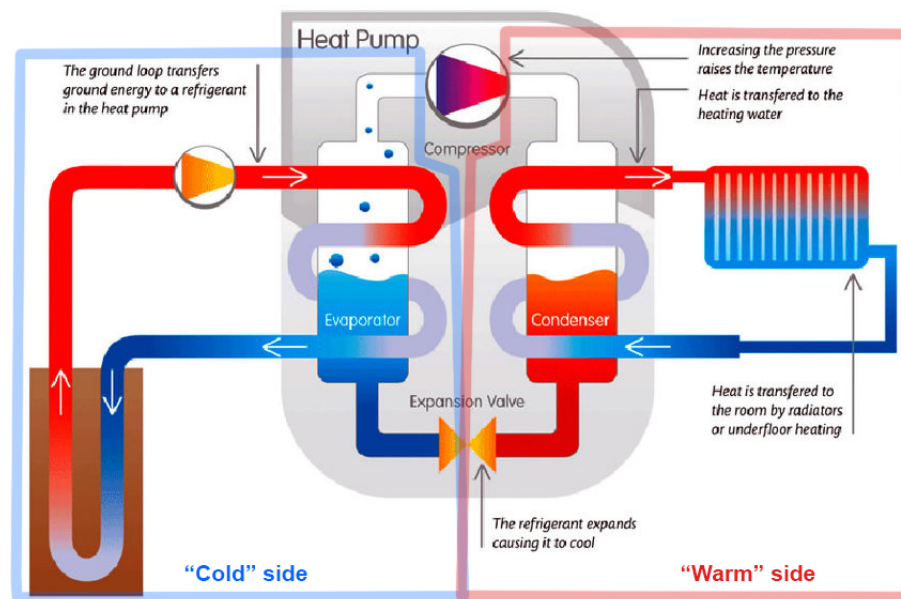


Figure 1: Schematic presentation - working principle of a heat pump system.

The whole system is composed of 3 parts, the heat pump itself, the cold side (left), the heat pump and the warm side (right). [3](Zheng, 2018)

The heat pump is the driving force that pulls the energy from the “cold side” liquid and adds it to the “warm side”. Additionally, most heat pumps can be used also in cooling mode, reversing the cold and warm side. However, in this project we are only focusing on heat pump being used for heating. It has to be noted that our proposed changes might reduce the efficiency of the reversed cooling cycle. However our changes to the system - control system design and valve setting adjustment - can be easily reversed. Ergo, the coolant cycle will not be affected by the proposed changes to the heating cycle.

3.1 Cold side

The choice of the system type and the sizing of the aforementioned system has a significant effect on the heat pump. If the specific characteristics of the heat pump are not achieved, it will have a degraded operational performance, which results in increased power usage and faster breakdown of the compressor.

Compared to the warm side, the cold side of a system has fewer components, and less complex control systems. The requirements of the heat pump on the cold side can be generally achieved by increasing the cold side flow rate, which is controlled by the well pumps.

To achieve a deeper understanding, it is necessary to look at the system as a whole, including the cold side even though the main focus of this project is the warm side and the heat pump itself. However, dimensioning and choosing the proper cold side system is a multidisciplinary field,

thermodynamics being among them, thus it is out of the scope of this project. Therefore, seen from the automation and control engineering perspective, the warm side of the system and the active controls related to it are the main points of our interest.

There are several ways to classify heat pump thermal sources. We are presenting a few of the most common sources:

3.1.1 Air / atmosphere

Air source heat pumps are the most common. The relative ease of install and decreasing unit cost have made them popular for heating and cooling needs of individual residences, both for new developments, as well as a retrofit option for existing housing. In 2012, 24% of Norwegian households were using air-to-air heat pumps. [4](SSB, 2014) However, the biggest disadvantage is the high fluctuations of the possible outdoor temperature depending on the time of year and geographical location. In addition, a change of outside temperature which is faster than the heat pumps ability to compensate (because of the slow nature of heat pump systems) can result in system reaching saturation which suffocates the system since it does not transport enough energy. On a cold winter night or a warm summer day the temperature differential might have a significant impact on the temperature output. While the decreased efficiency can be relatively easily compensated in small scale installations by additional power use, this poses a more significant problem in larger installations, where the fluctuation in power use would result in a much higher degree of expenses. Therefore, air-based heat pumps are mostly used in less complex and smaller units.

3.1.2 Seawater

Seawater is characterized by a marginal change in temperature. Seawater source heat pump installations typically are from several hundred kW and can be easily scaled up to MW range. Typical use cases are large communal districts or industrial areas.

Downsides stem from the geographical availability and the lack of small-scale solutions. It requires relatively high investment and significant running costs because of the damaging properties of salt water and foreign particles from the sea. This results in high maintenance regarding the filtering parts and possible corrosion issues.

3.1.3 Groundwater

There are several categories of ground source heat pump systems. Among them are ground water sources both from the surface and aquifers as well as vertical boreholes. This necessitates a more complex and expensive installation, for example the costs associated with borehole drilling. Such systems usually are closed loop, with fluid only as a heat transfer medium. It will not experience pipe scaling and mineral buildup. Therefore it has lower running costs in relation to other systems.

Ground thermal sources offer the highest temperature stability which in turn will decrease the heating expenses, offsetting the initial cost of the system. This however is dependent on the system being properly dimensioned as well as it having a control system that ensures proper working conditions for the warm side of the heat pump.

From the perspective of control engineering this is the closest to a system with ideal characteristics. It is a common solution for medium to large installations around the world.

3.1.4 Gray-water

On an individual installation level, gray-water (waste water from sinks and similar appliances) can be a source for partial heat recovery. However, to obtain the full needs/requirements of a heat pump, the system needs to have a much higher volume, for example, a sort of communal gray-water collection.

Such availability is dependent on local infrastructure and the local laws and existing installations like heat cables. This can be viable in other countries with suitable conditions where the gray-water would maintain more stable temperatures year around.

3.2 Heat pump

In this section we take a closer look at the physical components of a heat pump - heat exchangers, compressor and the relevant refrigerants.

The heat pump itself must be seen as an independent control loop in the bigger control system. Its function is to transfer energy from one side of the system to the other. The travel direction of the compressed refrigerant depends on the way the energy (heat) has to be transferred.

Compressing the refrigerant will increase the temperature. This is the ideal gas law, Figure 2 is a textbook representation of Boyle's, Charles's and Avogadro's law.

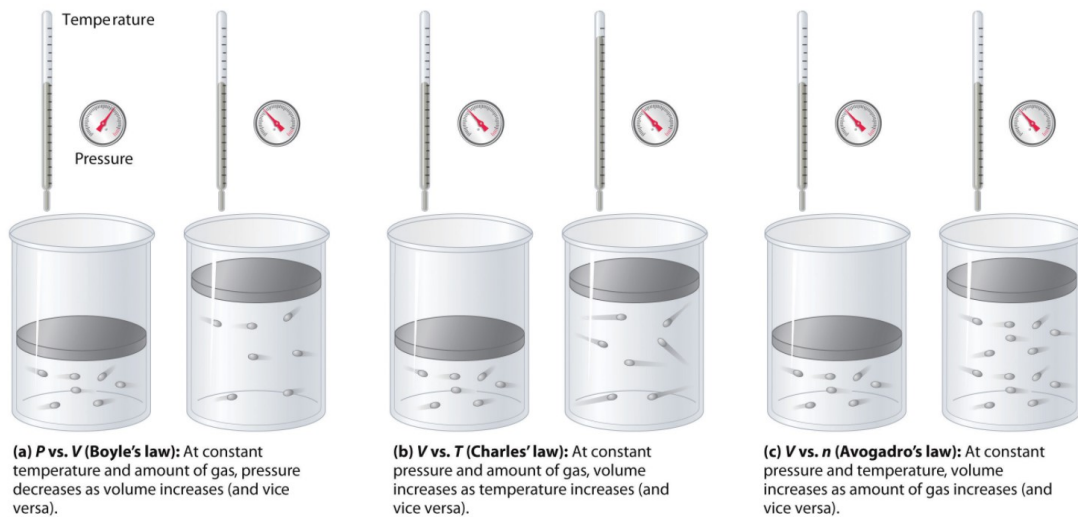


Figure 2: Boyle's, Charles's and Avogadro's gas laws [5](chem.libretexts, 2017)

But to encapsulate the essence, we have a constant volume and are compressing gas into it, this creates pressure. With a constant volume and with increasing amount and pressure we are going to get an increase in temperature, described by the ideal gas law $pV = nRT$.

When the hot side (Figure 1) has reached a certain pressure-point, a release/ expansion valve will allow some of the gas to escape to the cool side where the gas gets to decompress, and the gaseous temperature will significantly drop. Continuing the circle, but before it goes back into the compressor, it takes up heat from the cold side liquid with the help of a "heat exchanger".

3.2.1 Heat exchanger

Heat exchangers are the bridges that connect the heat pump to the other systems. There are many specific types of heat exchangers, for many specific tasks. But the principle of all of them is quite simple, there are two physical media that by means of physical contact exchange/give heat to the other.

The difficulty comes when there is a required target temperature on one of the medias and it is needed to control the other loop.

3.2.2 Tube heat exchanger

Similar to the other types of heat exchangers, a tube in tube heat exchanger is comprised of two tubes, one for each fluid. However, the tubes are coiled together to form an outside and inside pattern.

The application for a tube in tube design can get fairly creative. Since the tubes are coiled together, most designs for this type are compact. Applications for a tube in tube heat exchanger center around high temperature and high pressure. Since it runs at a higher output, a tube in tube heat exchanger tends to have greater efficiency.

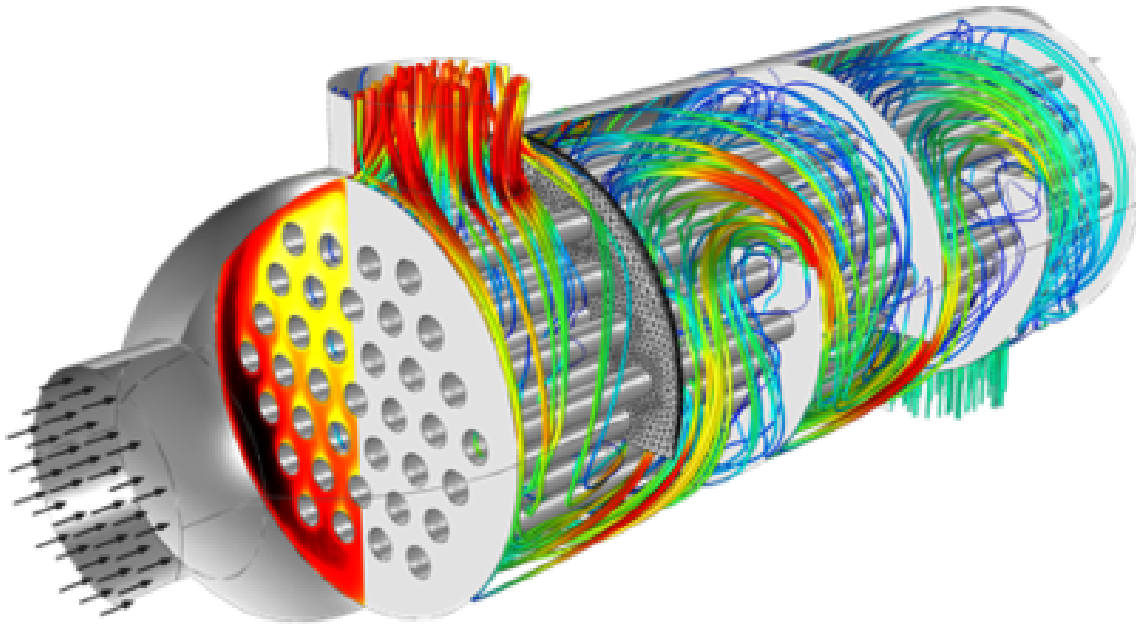


Figure 3: Representation of how a tube heat exchanger works. [6](Comsol, 2021)

3.2.3 Plate heat exchanger

Most types of heat exchangers have a similar design, the plate heat exchanger is the exception. Metal plates are used to transfer heat between two fluids. The plate is a metal shell, with spaces inside each plate that act as hallways for fluids to travel through.

With a plate heat exchanger, there is a greater surface area in contact with the fluids, which gives it better heat transfer compared to all other types. Although plate heat exchangers can be more expensive, the efficiency gained by the design is a huge plus. This type of heat exchanger has great durability and long lifetime. Because of its large form factor it is most commonly used in larger industrial applications.

3.2.4 Microchannel heat exchangers

Microchannel heat exchangers are a form of plate heat exchangers, where all the fins have cooling liquid flowing through it. In this setup the cooling medium has increased travel path which results in longer exposure time to the substance that inflicts heat. Giving us a system very close to an ideal system. The disadvantage of the microchannel is that there is a higher risk of spillage

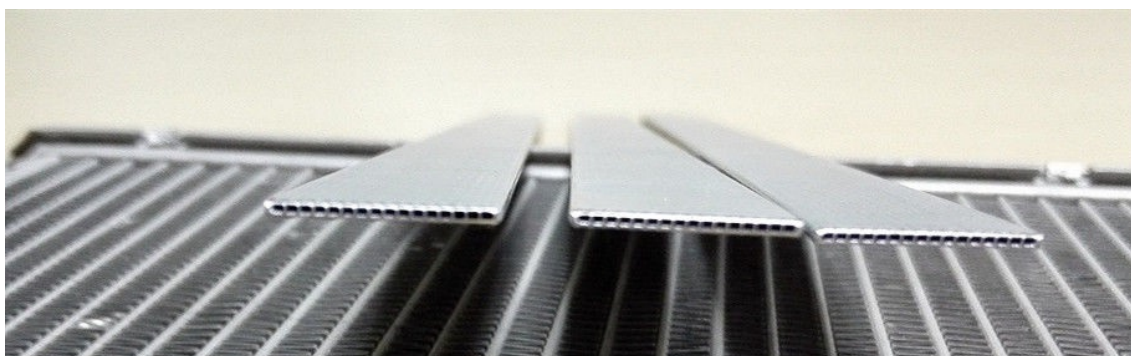


Figure 4: How the inside of a microchannel heat exchanger looks [7](Trumony, 2021)

between the two substances. Additionally, they are mostly used with “air to air” systems today. But there is an increased focus on this technology, and we are most likely going to see it in more applications over the next few years.

3.2.5 Propane

To conclude propane suitability as a replacement for R-22 it is necessary to quantify its properties and their relevance to propane application as a refrigerant. In Figure 5, we can see a phase diagram that is the heart of those questions.

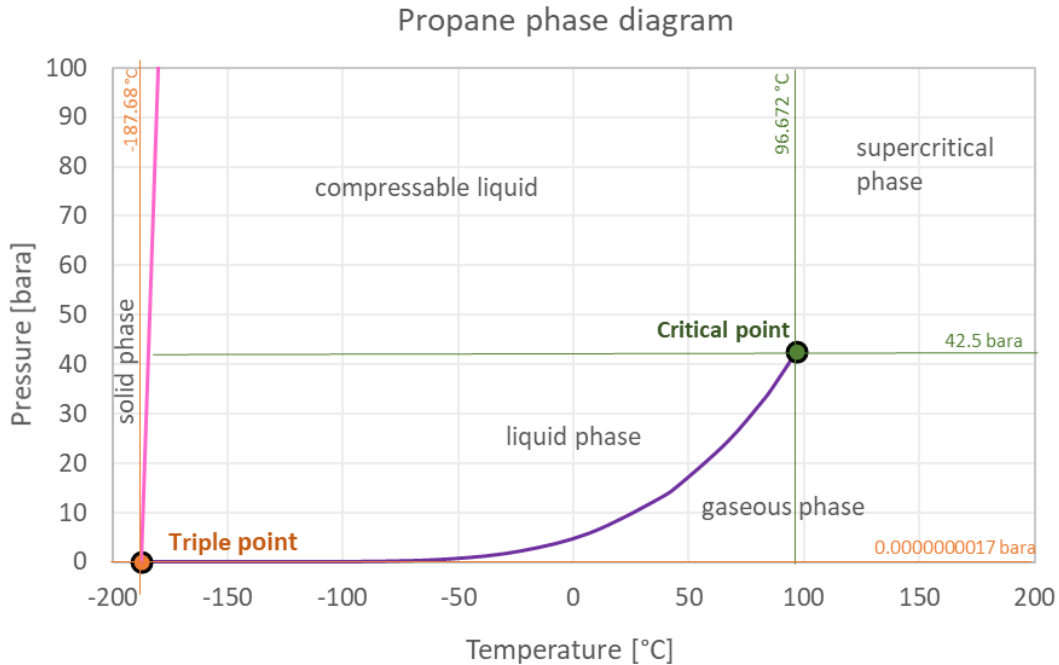


Figure 5: Propane phase diagram [8](EngineeringToolbox, 2021)

The normal working pressure of a heat pump is between 0 and 50 bar. And we can see from the graph that we have some major points we need to look at. The phase change of the gas is one of them. If the system compresses propane to 10 bar and the temperature of the gas goes from 40°C to 20°C, the “gaseous” propane will go to its liquid phase.

Additionally, if the critical point of propane is in this scope of 0 to 50 bar, then the “critical face marker” signals the point where the gas phase and the liquid phase coexists, and all the gas in this mixture gets the same density as the liquid propane has, giving the mixture new properties. As seen in figure 6, the supercritical phase is reached at the critical point. The supercritical phase can conduct more energy at the temperature around the 200°C area.

This is very interesting in systems where it is necessary to super-cool/ heat something, even for a 100 kW building, having supercritical phase will create a lot of complications when it comes to equations and representation of the systems and therefore it complicates the simulation of this process. But it is an essential part of these systems and we need to accept this state to reach higher energy levels. So, operation below the critical point of 42.5 bars are not practically feasible for a system that needs to deliver 100 kW energy at a high rate, and additionally at 60°C.

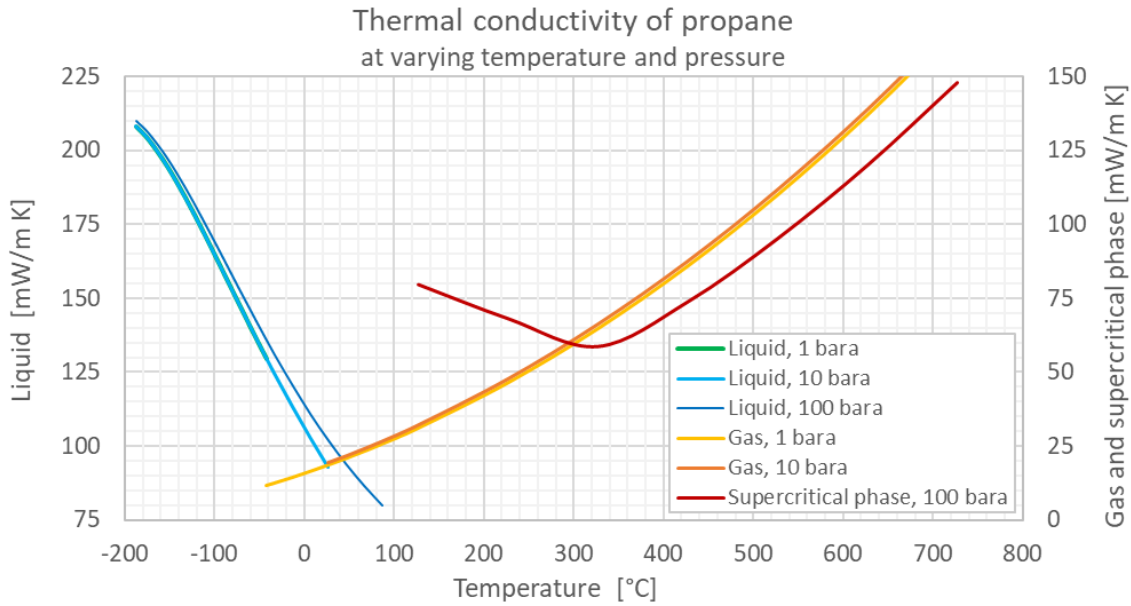
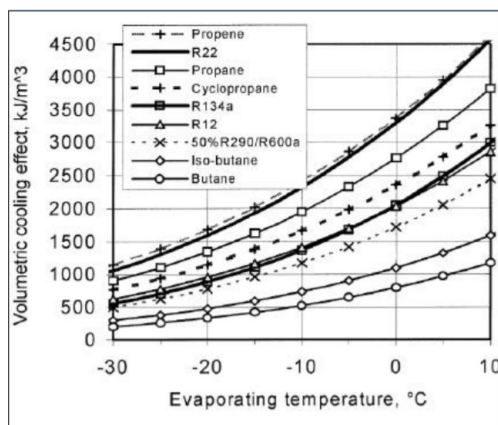


Figure 6: Thermal conductivity of propane in different phases and temperatures [9](EngineeringToolbox, 2021)

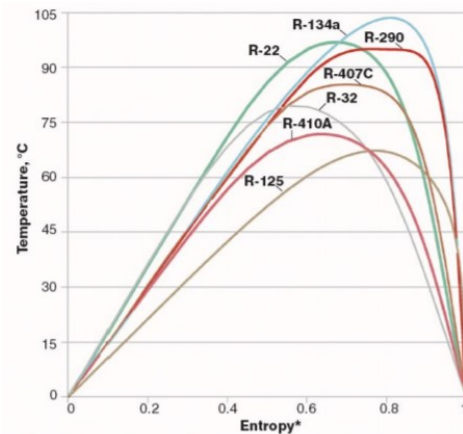
As seen in the graph 7a, it is easy to take the conclusion that propane is not only similar to R-22, but also slightly outperforms it. Giving it great energy conductivity potential.

However, natural propane does not have odor and the propane that is used for heating and combustion has additives that do not mix well with lubricants or other substances that are needed for the moving parts and the longevity of the systems. Therefore, the only variety from the different producers of R-290 (propane refrigerant) is different lubrication, if premixed. R-290 is the most readily available propane refrigerant for purchase.

Additionally, we are interested to see if it is possible to increase the Δ energy transfer and/or COP (coefficient of performance) by using a tailored refrigerant based on propane or other natural refrigerants.



(a) The evaporating temperature and cooling effect of propane and other refrigerants - [10](BEL, 2005)



(b) The difference in thermal entropy in different refrigerants - R290 thermal properties are on par with other refrigerants. [11](Calm and Domaski, 2004)

Figure 7: Properties of different refrigerants

3.2.6 R-290

R-290 has two components - pure propane C_3H_8 and a lubricant. We will look at its properties and compare it to R-22 since it is a very common reference when comparing refrigerants. Our source of data [12](Park and Jung, 2008) also compares it to R-1270 even though it is not our main focus.

As seen in figure 8, R-290 and R-22 have close performance in regards to COP and energy transfer. A point of interest from the article is that change in COP and general properties of R-290 slightly outperform R-22. This makes sense because R-22 was initially developed as a non-flammable replacement for natural refrigerants used in early phases of refrigeration system development. However, as we see in figure 7b, R-290's biggest difference from R-22 is the entropy curve which describes the optimal working area.

We can also see from 8 that the R-290 is a capable refrigerant for temperature control and heat pump systems. This poses a question why propane has seen limited use until now. The main reason is high flammability of propane. To ensure safe use, it is necessary to use gas leak sensors as well as other safety precautions.

It is just the recent years where the general increased heat pump popularity and increasingly stricter regulations [13](Miljødirektoratet, 2019) have made the demand for natural refrigerants to have a second try.

With many heat pumps located in dedicated technical rooms there is a low probability of failure or injury, therefore there is a higher initiative to utilize R-290.

Refrigerants can have additives based on their intended use case. Cases can be tailored towards specific work areas or properties. It might require doping of the base refrigerant. By doping or other means of changing the standard propane it could give a more favorable substance for the heat pump systems, but it can also increase the GWP. Therefore it is not so common practice to have additives, other than lubrication, in the refrigerant. This makes R-290 the de facto standard.

The balance of pros and cons is the main drive for finding more suitable refrigerants for all applications. In 2008 Jung and Park did a comprehensive study about R-290 and propylene (R-1270).

Comparing them to R-22 in all kind of conditions that a heat pump would be used for commercial heating. We refer [12](Park and Jung, 2008) as further reading about the confirmed capabilities of R-290.

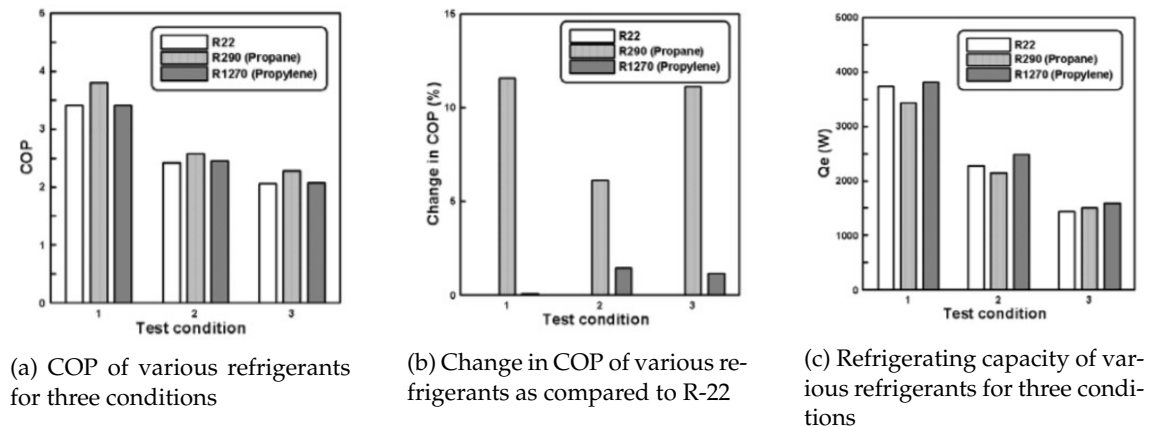


Figure 8: Reference: [12](Park and Jung, 2008)

In the figure 8 from the article of Jung and Park there are three tests with three very different conditions (test 1: $7^\circ C / 45^\circ C$, test 2: $-7^\circ C / 41^\circ C$, test 3: $-21^\circ C / 28^\circ C$). It shows the coefficient of performance of propane, compared to the R-22 and R-1270 refrigerant.

An important note is that propane requires far less pressurization, which causes far less of a load on the system, and the potential danger/ damage. All 3 of the refrigerants in the figures show capability of use in heat pump systems.

R-1270 is a good refrigerant as well, and a good alternative. But in Jung and Park research, it shows that the R-1270 has even closer similarities to R-22 and not only in COP, but also in pressurization/ power need, making R-1270 a good replacement for R-22. It also shows that R-

R290 performs 6.1-11.5% better than R-22 with lower power draw / consumption from the power-grid. We conclude from these results that R-290 is a good refrigerant for new installations and for conversion of existing installations R-1270 might be better choice.

3.2.7 CO₂

Using CO₂ as a working fluid requires high pressure and in turn more expensive setup. But there is an upside to using CO₂ if we work around these challenges. The biggest advantage to CO₂ is its heat capacity. The largest disadvantage of using CO₂ is that the critical temperature is at 31.1°C, which means that sub-critical operation yields a low output temperature. Thus limiting its usefulness for heating larger installations. Fortunately, the design pressure of CO₂ can get a lot higher than other refrigerants. CO₂ systems typically have 150 bar at its maximum pressure.

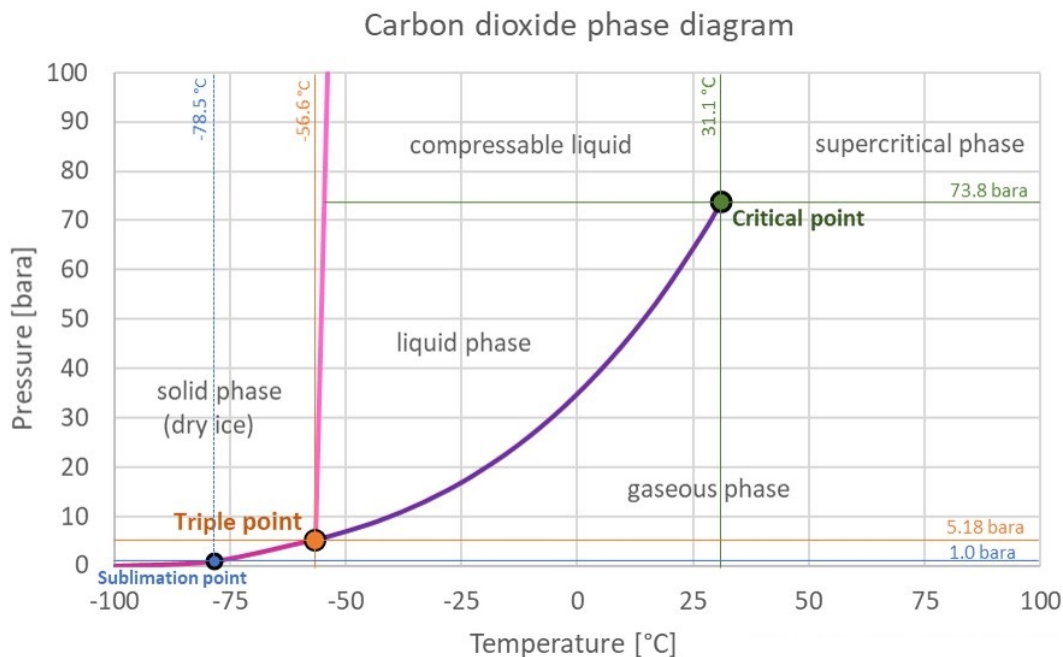


Figure 9: Carbon dioxide phase diagram - Reference: [14](EngineeringToolbox, 2021)

But as discussed in previous chapters, pushing the working fluid over the critical point means that the system must endure a substance in a super critical phase, giving the gas and the liquid the same density and new properties.

This means that the heat rejection is not done any more by condensation, but by cooling, by the gas cooler. The heat cooler has similar designs to the condensers but uses other thermodynamic principles in the builds because of the new properties of the super critical phase. The principles of the heat exchangers we have discussed stands regardless if it has to be tweaked, for the designer to have maximum efficiency.

The advantage of using a heat pump design that is capable of the high pressure and the challenges of working over the critical temperature is that it results in an impressive energy efficiency, thermal conductivity and a low value of its dynamic viscosity that is characterized by the CO₂.

Compared with other refrigerants, thermodynamic properties of CO₂ seem to be rather favorable in terms of heat transfer, given we want to take on the more complex installation which could give both higher costs in set-up and in servicing over the years.

3.2.8 Compressor

Heat pumps mainly utilize volumetric reciprocating (commonly referred to just as piston compressors) and rotary compressors. Reciprocating compressor consists of one or several cylinders with pistons. Their rotational speed varies depending on the type of compressor. Large reciprocating compressors with a compression ratio of 2 can have around 100 RPM, but smaller, faster compressors with a ratio of 10 can achieve several thousand RPM. [15](Grassi, 2018)

The most typical control of reciprocating compressors is a simple on-off scheme. The main drawback is the increased number the startup phases, which increase premature wear of compressor and motor components. The next design iteration is step control in multi-component compressor. As an example, in a reciprocating compressor with multiple cylinders, some of them can be shut off.

A recommended solution which is the most relevant to this thesis is modulation of the driving motor itself with the help of an inverter. This allows the largest COP gain. It however comes at an increased cost and complexity with regards to control strategies.

There are several other types of compressors, screw compressors being one of them. It can be described as two counter-rotating screws that compress the space between their threads. Full seal between them is achieved by an oil film, this oil film must be the same as the lubrication in our case, since we will not be filtering/ recovering the lubrication by an oil separator. Instead, this must be taken in the consideration when choosing lubrication.

Their main advantage is a smoother operation compared to reciprocating compressors. This results in less noise and vibrations. They also have better properties for control and modulation. It can be done by a sliding valve which affects the suction volume. While they are available in most power ranges, their best efficiency is in systems above 50 kW.

Vane and scroll compressors are typically used in small applications, for example in heat pumps in electric cars. Vane compressor consists of an eccentrically placed rotor that moves in a spiral motion while compressing the fluid. In regards to larger systems, vane and scroll compressors are typically outperformed power and cost wise by reciprocating compressors.

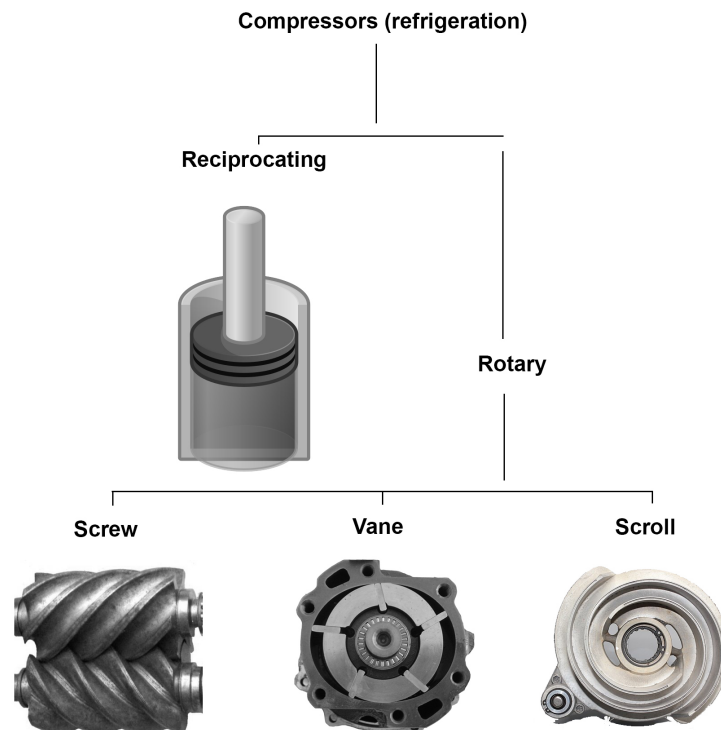


Figure 10: Various types of compressors used in refrigeration
 [16](TilmannR, 2019) [17](Motorhead, 2006) [18](Xlory, 2006) [19](MailManG40, 2021)

3.3 Warm side

The warm side generally is a closed water loop with various load alternatives depending on the application. They can be used alone or in a combination as a bigger system.

3.3.1 Ventilation

As the warm side of the system in our case uses liquid as the heat transfer medium, an additional water to air heat exchanger is necessary in the case of heated ventilation. The system can be made more efficient by utilizing a partial heat recovery from the returning air in its circulation cycle.

3.3.2 Underfloor/ Outside under-road heating

Underfloor heating is well suited in heat pump based applications. Compared to a conventional radiator system, it requires lower temperatures, which often will correspond to the most efficient operating temperatures of various heat pumps. It can as well be utilized in colder climates to heat up a residential driveway or in specific industrial applications that require a snow free surface. [20](Sarbu and Sebarchievici, 2016)

3.3.3 Radiators

While not as commonly paired with heat pumps as underfloor heating, radiators do entail several advantages. For example, such system will be much quicker to respond to the occupant's requirements when warming up or cooling down. However, as the heat pumps have rather narrow and specific optimal temperature ranges, the sizing of the radiators must be taken in account when designing the system. [20](Sarbu and Sebarchievici, 2016)

3.3.4 Electric boiler used in the temperature control

An essential part of the existing system that we are going to evaluate, is the electric heating unit that regulates the temperature in the hot side system. In the existing physical system that we are basing our thesis on, the heat pump is always going at 100 % efficiency, and it is the electric heater's task to regulate the temperature depending on the load. The heater has electric heating elements that normally can reach 120°C temperature and can be controlled based on a set value and a current value via PID (a feedback regulation technique utilising proportional, integral and derivative components). Or it can be controlled by a 0 – 10 V or 4-20 mA control signal from for example a PLC (Programmable Logic Controller).

This specific unit uses a NTC (negative temperature coefficient) thermistor sensor unit to measure the water's temperature as the water departs the heating chamber. The internal heater's control will then choose an appropriate step, based on the 15 heating steps it can choose from, where it turns on or off several of its many heating elements, where each element will hold a stable 90°C. This setup is harder to control, especially since it does not have a particularly linear graph. In this case we can control the electric heater via PLC if we override default internal controls of the electric heater. Use of step control instead of feedback control will also will introduce irregularities in the system temperature. (Discussed in chapter 7)



Figure 11: Electric heater [21](Armatec, 2015)

3.3.5 Thermal and electric boiler

The heating system for the tap water is heated in 2 portions. One part by thermal heat transfer from the heat pumps warm side system. The other part by electric heating which rises and holds 70°C for long enough time to make sure that no harmful bacteria like legionella bacteria can survive. It thrives in temperature between 20°C and 50°C, but cannot establish or grow in temperatures over 60°C. It is also important to hold a high temperature for a long enough time to kill off any other already established bacteria that comes from the water source. So, the temperature and the amount that is used from the warm water tank is monitored to make sure that all the water that is used is safe for consumption.

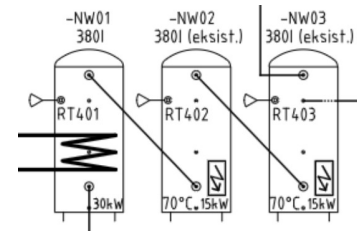


Figure 12: System to pasteurize and heat the warm water.

3.4 Control hardware

3.4.1 PLC

There are several ways the heat pump and the warm side can be monitored and controlled. Small systems typically use embedded microcontrollers in the heat pump itself. While this is the cheapest and space wise smallest option, such system offers little degree of flexibility. Thus, medium and large sized installations will rely on programmable logic controllers (PLC).

Multiple configurations of control exist. The heat pump might have its own integrated controller for its low-level tasks and PLC might be used to control the warm side parameters and to turn the heat pump on/off when necessary. There is also a possibility of delegating all the heat pump internal control to a PLC. The specific configuration will depend on the model of heat pump and the needs of the system.

The main advantage of a PLC system is the inherent flexibility. If needed, the system can have a very simple logic that will ensure the function of the system. However, it also offers the possibility of more advanced computation and wide array of sensing options, which enables a dynamic and adaptive control. Such solution will try to keep the heat pump within its best operational characteristics which in turn will lower the energy usage.



Figure 13: Cabinet housing PLC units for a propane based heat pump system. Picture taken from the field work at our reference system location.

3.4.2 Sensor system

Sensors monitor the physical aspects of a system. The system can have basic functioning with relatively few sensors, however there is an inherent advantage of having a multitude of them – mainly because it is possible to gain a more complete understanding of physical processes at various locations, at different timepoints. This data can be used to investigate potential inefficiencies and to optimize the running parameters of a system.

This process can be done manually by a person at the time of the initial setup of the system. However, the main benefit stems from automation implementation that utilizes sensors continuously. Depending on the type of installation the sensors are connected either physically or wirelessly to a PLC. According to the type of sensor, it most often will have an analog value, that gets sampled and digitized. This process is usually done by the PLC itself.

The warm side of the circuit, typically will implement a combination of temperature, pressure and flow sensors on the outgoing and returning lines. The heat pump itself will also have its own sensors. In addition to the sensors mentioned before, it might have sensors to ensure safe operation in case of a refrigerant leak – a propane-based heat pump often is mandated to have a

gas sensor within its enclosure. Variable speed compressors might utilize additional sensors to monitor the compressor performance.

3.4.3 Valve control

A control valve is an active tool used to change specific process variables, for example flow rate and pressure of a fluid. It can function as a shutoff (stopcheck) device or it can be regulated to achieve a specific set point. The latter implies use of control logic and sensors before/after the valve.

3.4.4 Propane safety

There is no way around the fact that considerations have to be taken when using a flammable substance like propane. There must be systems monitoring leakage from the heat pump. Leakage is the problem heat pumps are most prone to have, but the amounts/ accumulation of refrigerant is usually low and dangerous failure is unlikely. Furthermore, they are less likely in a system with lower pressurization.

Propane also has no natural odor. Unlike fuel propane, common for cooking and other uses, propane used as a refrigerant in heat pumps has no artificially added odor. R-290 is flammable, so it must be monitored continuously for potential leakages. In the worst-case scenario, a propane-based system can explode. However, with basic sensor monitoring and the right stop sequences these scenarios are highly unlikely.

To ensure safe operation, multiple redundant solutions are implemented. The heat pump is enclosed in an EX-area with negative air pressure. Negative pressure is attained by a continuously running an exhaust fan. Thus, an eventual leak would be expelled to atmosphere where it would safely dissipate. An uninterrupted power supply is utilized in the case of a power loss. Regardless, there are also multiple propane sensors situated around the room if the exhaust venting would fail. Since propane is heavier than air, these sensors are located near floor level.

While several parts of this can be controlled from the PLC, it is important to have a manual system in addition to PLC-based control that would not be affected by a power failure. For additional safety, those controls can be even made available outside of the room itself.



(a) EX enclosure with an exhaust line in upper right corner



(b) Propane sensor head unit, UPS and a battery bank



(c) Manual controls situated outside of the room

Figure 14: Safety related components. Pictures taken at the location of our reference system.

4 Potential solutions

In this chapter we are going to draft different aspects that we can change, and the different outcomes they might have on the system. This discussion is going to primarily focus on power savings and the cost and trade-offs, these changes have. These hypotheses we draft in this chapter, will later be tested using a simulation tool.

4.1 Description of possible solutions

Now that we have seen the intrinsic issues and how the warm side system and a heat pump works, we are able to strategize and make some clear decisions on how we are going to tackle the problem of Δ energy (W) in the warm side system.

Today, most of large scale installations of heat pumps are regulated by PLCs that control valves and pumps based on sensor data. We plan to use the same approach, just with adjustments in the flow map (component and process layout) and more complex sub-loops.

It is clear that the colder the returning water is, the slower the system is going to be. But the more efficient the use of energy is. We can use a water reservoir or electric hot water storage tank that supplements / adds water with a higher thermal effect, which can rise the main water pipe temperature via a 3-way valve from 55°C to 65°C if we need the extra energy. However, using a bigger electric water reservoir will cost both space and money. Keeping the size close to what is common practice in this profession, might be advisable. Additionally, it will supplement energy needs that the fast changes will demand.

4.2 The approach for a physical solution

R-290 has a high Δ energy uptake, however the optimal temperature range where R-290 has the highest COP is at a lower temperature than for example R-134a, which is one of the most common refrigerants used in new installations. This means that even if R-290 has higher energy uptake than R-134a, it needs to go outside its optimal COP area. In turn the compressor uses more energy to achieve higher refrigerant pressure.

We can use minichannel condensers or other forms of adding or releasing more energy from the refrigerant, which will give more room for energy to be taken up by the source. Alternatively, expose the hot-side water longer to give off as much energy as possible. Giving room for a higher energy take up on the cold side.

Or we can scale up the system, using more refrigerant, but this is a step in the wrong direction as the maximum R-290 refrigerant volume is often limited to 5 kg due to safety regulations. It might be more cost efficient to scale up the heat pump itself, instead of scaling up the hot side transfer systems, to give a longer exponent time.

Instead of using bigger or better heat exchangers we could use two or more heat pumps in series (to bypass 5 kg limit). This is common practice if we want 60+°C water in all the warm side circuits. This would double or triple the compressor energy usage for higher temperature level on the warm side.



Figure 15: Two ENRAD Chiller VKA. Picture courtesy of [22](Galaasen, 2019)

By using two ENRAD VKA heat pump units (same heat pump as in our reference facility) we can hold "60+" °C water and get different kW outputs depending on the heat pump model put in series. The heat pumps in example picture [15] deliver 283 kW at 100% performance.

The approach of getting a better heat exchanger relies on the principle of getting more energy from the cold side to the warm side, but this strategy of using two or more units goes on the principle of getting even more from the cold side at a higher temperature potential. This must be factored in the well-equation which represents the cold side system. It is described more in-depth in Chapter 5.

Using two heat pumps is a viable option at the cost of increased energy usage. However, it allows us to bypass the electric heater (which has significant energy consumption) completely and control/ regulate the warm side loop with only the heat-pumps that are in "Master- Slave" mode and can regulate the compressors by both PWM (Pulse Width Modulation) and voltage.

The reason that the electric heater is favored, is because it has a fast response time and can compensate big and fast load changes, and the electric heater does not have a temperature level that is too low, compared to the heat pump that has a breakpoint in performance at low temperatures.

So, by bypassing the electric heater instead of getting rid of it, we can still use the electric heater for startup phases and, as a back-up solution, if one of the heat pumps breaks or an abnormally large workload would occur.

4.3 Proposed changes of the system

The current system design is highly optimized. With our current knowledge level it is difficult task to find significant improvement areas. Our goal is to approach this problem with our abilities in search for potentially unconventional solutions and to question the standard design practice.

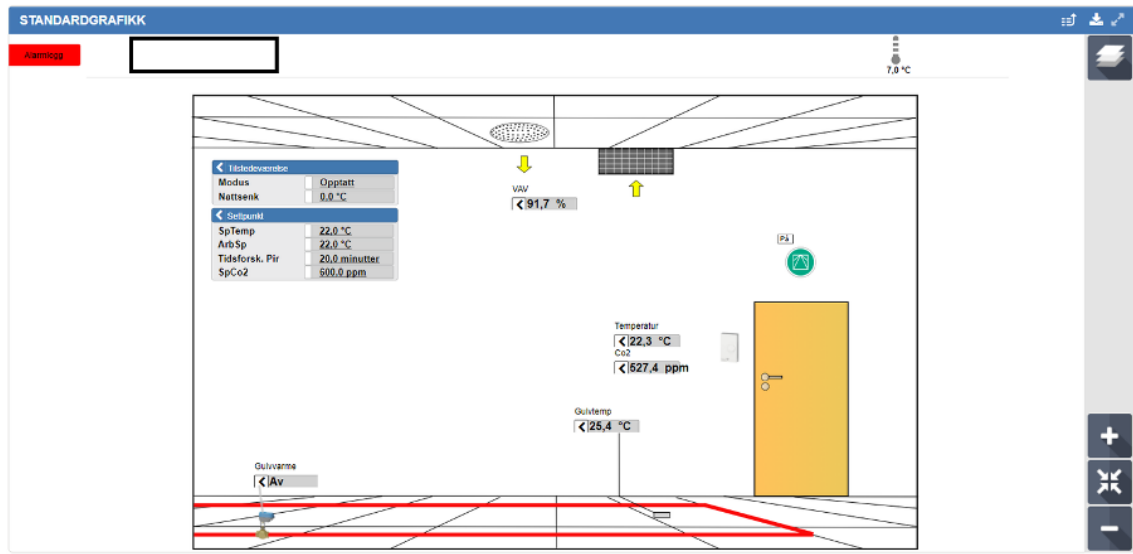


Figure 16: Graphical user interface for manual observation and control of individual rooms in our reference facility.

One thing to note is that we try to hit a specific heating effect in watts, in every individual room which is calculated by the room's current temperature and an individualized set-point temperature. Additionally, we are focused on keeping the return fluid at a specific value, to keep the heat pump at a good COP as previously explained. Meaning that we want to change the input heat or the speed of the pump.

Traditionally this is done by "on-off" control. As of today, the system pumps 60°C water into the pipes, then overshoots the room target temperature by a small amount. Then it waits for it to drop to a lower value before continuing the cycle. This "on-off" control system uses only a small amount of energy in the pump usage, since it rarely pumps water to more than a few rooms at a time.

4.3.1 First proposed modification: recycling water loop

One of the changes that we propose is that we connect the independent warming loops of the radiator cycle to the floor heating and replace the low pump energy scheme for a low water heating scheme. Meaning that we regulate the temperature with water that has already gone through one cycle with the new 60°C water, in a more constant stream of flow in the rooms, or with a higher rate of "on-off" cycles (like PWM) because of the lower heat transfer rate of the new inlet value.

The advantages of this proposal

- Less electric heating.
- Less room temperature change, ergo a more stable temperature.

The disadvantages about this proposal

- Increased mechanical stress for the pump because of more startups.
- The power consumption of the pump.
- A lower Δ heat difference between the warm water and the concrete floor. Which decreases the heat transfer rate and creates a slower system. (Lower temperature gradient).

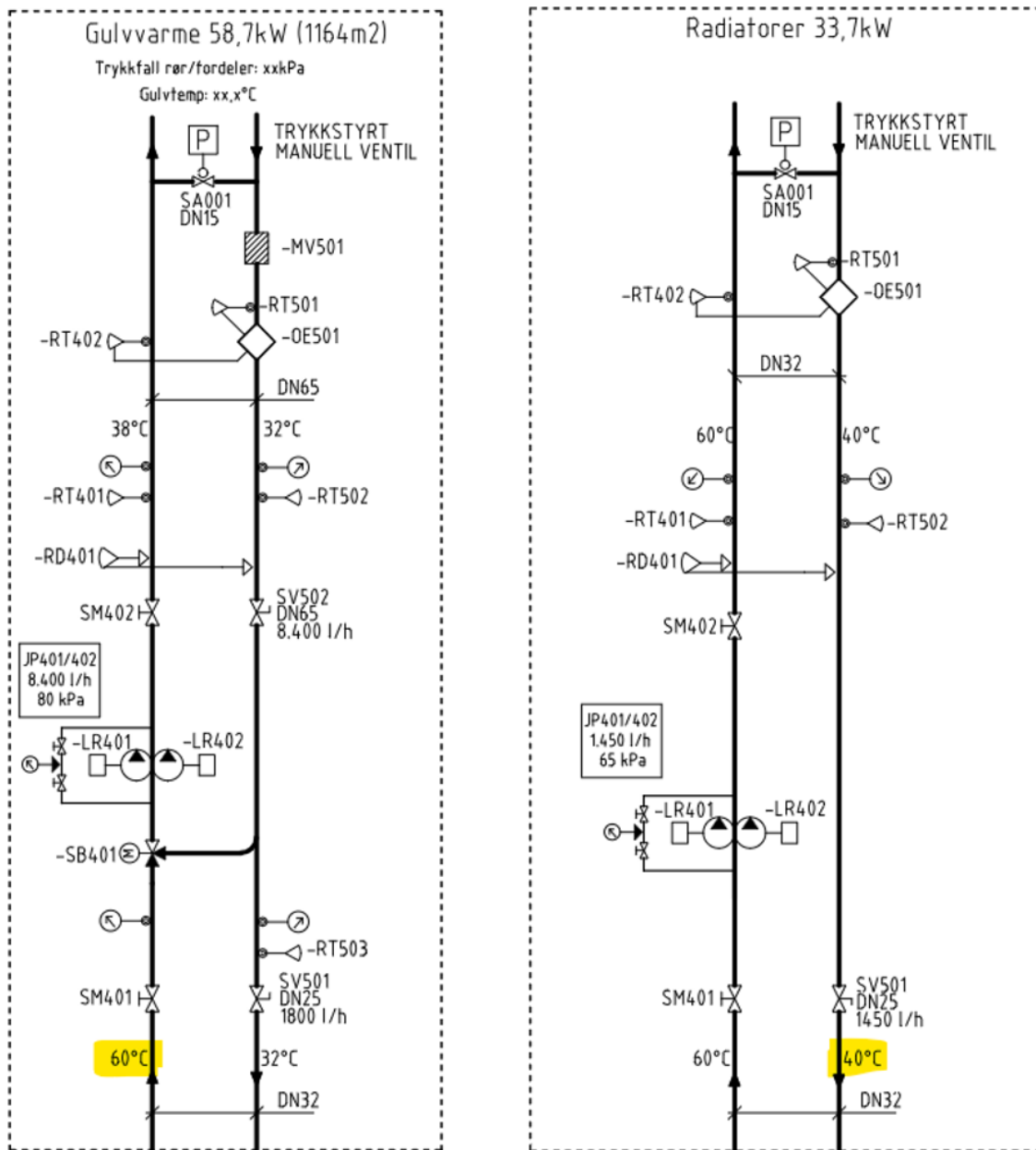


Figure 17: Part of the piping and instrumentation diagram (P&ID) that showcases the floor and radiator heating where the marked temperatures illustrate where we propose the series connection.

The concrete acts as a capacitor and smooths out the change in temperature, meaning that the original system's jump in overshoot is not that noticeable for the users of the building. If we do not take in account the more stable temperature, because of the effect of concrete floor - Then the only advantage of our proposed modification is the reduced energy usage (W) for the electric heater. If this change increases the mechanical stress on the pumps so much that it decreases the lifetime of the pumps, this might not be an economical approach to this problem, even though we might have increased power savings.

4.3.2 Second proposed modification: continuous flow for the recycling water loop

Continuous flow is preferred because it allows stability. There are other advantages of having a continuous flow. For example, the start sequence for the pump leads to the biggest drain of power. Meaning that these types of pumps usually are built to mitigate the stress and power spikes of the quick starts of the pumps, and not for power savings in normal operation, where the water already is in circulation. Meaning that the kinetic power intake only needs to be the equivalent of the "friction" / resistance on the water by bends on the pipes, turbulence, viscosity or an increase of potential energy by increased height of a pipe.

However, by having pumps created to mitigate power consumption, and for continuous flow - the question is whether the existing setup is more economical. We know that the temperature will be more stable and controllable, because of the elimination of the fluctuation. But if it is not cost efficient then these aspects of the system are not great enough gain to change the setup.

Even though this would give this thesis a great foothold and a more interesting system to regulate, since we could use target value regulation with PID and Robust regulation. There is an increased cost with a more complex system. For example - the controlling unit and/ or the new pumps. The initial purchase might not be worth it if the system is not large enough to have sufficient savings to cover the investments over a short enough period. Furthermore, there might be need for a break-even analysis if the system does not critically need the accuracy that is offered by the regulated continuous flow system.

4.3.3 Third proposed modification: Parallel piping

Not all applications require 60°C water, as long as it achieves its target energy transfer and achieves the set-point room temperature. Heating all the water is not necessary for proper operation, but the heat dependent applications like the water heater tanks and the air conditioner still need to be supplied with 60°C water. Even though the floor heating and radiators do not need 60°C water to transfer sufficient energy. The loss in energy transfer rate might be avoided with a setup that we will propose in the regulation part later.

So, by letting the radiators and floor heating having purposefully slower heat transfer rate operation, we are saving energy on the normal operation. This would be perfect fit in facilities that do not have "downtime" like on hospitals or hotels.

However, on private housing where the resident might move away from the residence for longer periods of time, this might not be the approach. But depending on the needs of the users, there could be piping from the 60°C water to the radiators and floor heating in addition, so it can supplement the "heat up" phase to faster heat up the building from a cold state.

A possible solution would be to have a smart house application to the user's phone, that lets the user to turn on the heating of the house in a slower and more cost-efficient way before the arrival. Effectively avoiding the problem of a cold home to get home to.

The reference system's heat pump can deliver 55°C. Our proposal is to split the main line off the heat pump in two parallel lines. One is further heated by electric heater to 60°C and the other one stays at 55°C and is used for floor and radiator heating.

This "parallel" piping idea where the heating unit that raised the temperature to only a portion of the applications/ heating circuits might not be a good fit for this existing facility like the school building because of the cost of the new modifications and the fact that this building has frequent "downtimes" at night, weekends and school breaks, giving us more startups and a smaller gain to cover the new cost of installation.

4.4 Problem analysis conclusion

We have now discussed and formed our possible solutions. A point of particular interest is the warm side changes that will reduce high energy outputs that require the electric heater to subsidize with assistance. Which is not economically wise use of electric power, since the point of heat pumps is to have a COP that is bigger than 1 and uses energy from another source rather than electric.

To test our hypothesis, we need to study the underlying mathematics, so we can make a proper choice when it comes to simulation tool. The tool needs to be able to handle all the thermodynamic and fluid-dynamic aspects of the problem, while being open enough to implement custom code, changes and structure.

5 Defining The System

To define the system, we first need to make some assumptions, and form a theory of how the system works. Initially our goal was to use real-life data. Unfortunately we did not get satisfactory measurements from our reference system. We had no access to change or adjust the system and we only got log data from a very short time-frame. Therefore we need to rely on mathematical and physical theory to design a simulation program.

5.1 Mathematical models

The whole system we are going to recreate in this chapter is going to be a construct of several smaller systems. And to make it clear and easy to read through, will we try to systematically build up to the complete model.

We will explain what each of these smaller systems does and contain, and our recommendation is to have the list of nomenclatures by your side while reading through this chapter. The first part that must be explained before starting to build the model, is the physical model of a heating/cooling system.

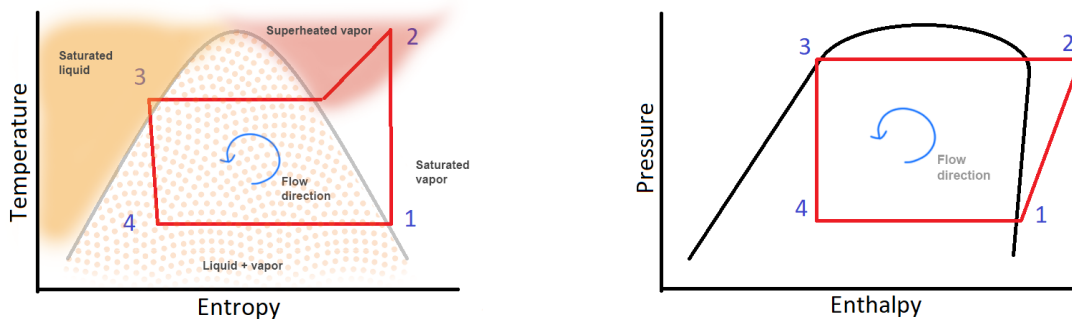


Figure 18: Basic graphs for concept understanding of entropy and enthalpy in the heating/cooling cycle.

As you can see in figure 18, we have pointed out 4 stages in the heating/cooling cycle. The main points of interest in these graphs are:

1. Between the evaporator and the compressor.
2. The point that the refrigerant escapes the compressor and have built up pressure.
3. After losing thermal energy in the condenser, right before entering the expansion valve.
4. Right after the expansion valve, where it has gone back to the same pressure as in point 1. And the refrigerant is on its way into the evaporator to gain thermal energy.

There is some point to be made in figure 18. To the left of the dome shaped curves in both the figures, the refrigerant is going to be saturated liquid. And under the curves there are going to be a mixture between liquid and gas.

And on the right side of this curve there are going to be super-heated vapor. Where the red lines touch the curves the refrigerant is going to be saturated, as a liquid on the left side and as a vapor on the right side.

Point 1: At the graph we can see that we have both low pressure and temperature, and that the vapor is saturated. The refrigerant in this stage is going into the compressor, the compressor is going to build up a pressure (kPa). One note is that the pressure at point 2 is the amount of efficient pressure that the compressor can give, subtracted the suction pressure from point 1.

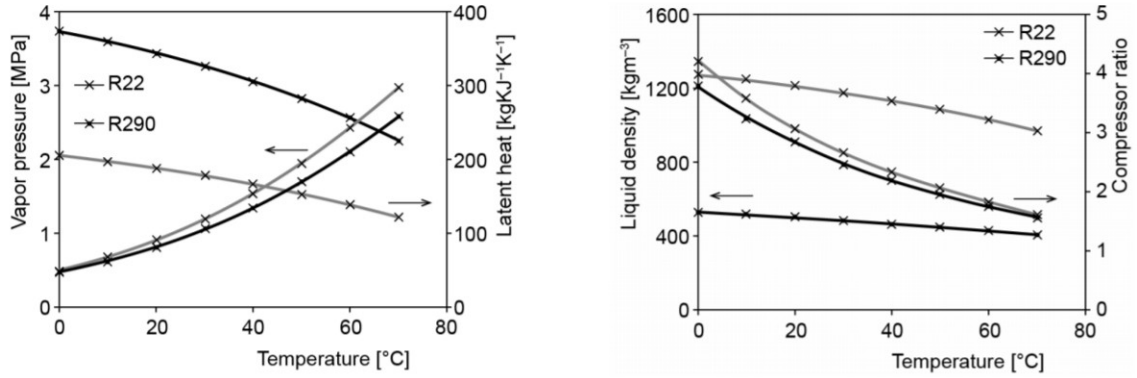


Figure 19: (Left) Vapor pressure and latent heat, (Right) Liquid density and compression ratio [23](Paradeshi, Mokkal, and Jayaraj, 2017)

As previously mentioned, there is a big difference in necessary pressure (to reach optimal COP) between R-22 and propane. The big difference between them is in the temperature range 0°C to 25°C where R-290 have 70% higher latent heat. Giving it good potential for solar heating applications. Which is not that useful for us since we are interested in giving heat in the 50°C – 60°C range. But it gives us a good image of how the substances differ and lag in their properties.

Propane has the ability to get to 96,7°C at 4.25 MPa vapor pressure given sufficient uptake from the cooled side. This gives us the context of a working area between 1.4 – 4 MPa vapor pressure. Which defines the compressors requirements, in kg/s and kPa. We can take a lookup in the saturated vapor refrigerant table, for example - [24](Climalife, 2020).

To refresh our memory 1 bar = 100 kPa = 0.1 MPa, and atmospheric pressure is around 1.01325 bar or 101.325 kPa. In addition the low-pressure side of a heat pump system is around 3 bar to 3.5 bar. So, therefore we use 3.453 bar for this analysis since that is a good value in the range and it is given to us in the saturated vapor refrigerant table. R-290 has a liquid entropy of 0.909 kJ/ (kg *K).

Work done by the compressor has a formula:

$$Netpower(kW) : \dot{W} = \dot{m}(H_2 - H_1) \quad (1)$$

Where H_1 is the enthalpy in point 1 and H_2 is the enthalpy in point 2 (see in saturated vapor refrigerant table for values). And \dot{m} is the mass flow of the refrigerant.

$$Example : \dot{m} = 3kg/s, H_1 = 250kJ/kg, H_2 = 280kJ/kg$$

$$3(kg/s)(280 - 250)(kJ/kg) = 90kJ/s = 90kW = \dot{W}$$

So, if we want to have a system that can deliver 100 kW, then we can either compress the refrigerant more to change the H_2 or have to use more refrigerant to increase the mass.

Point 2: The refrigerant has been compressed and gained a higher heat potential and is now in super-heated state where the vapor/ liquid has the mass equal to the liquid alone if it all was liquid. If we say that the compressor is isotopic, then there is no change or hysteresis to the values you will find in the saturated vapor refrigerant table. And we now know that the entropy in the super-heated point 2 is equal to point 1 (0.909 kJ/ (kg *K)). This is because super-heated refrigerant has the properties of the liquid state.

The refrigerant is now traveling down the condenser to give up its heat potential to the warm side, this will in real life also reduce the pressure as the temperature goes down, making the system a bit more complex since the transfer of energy is not uniform over the whole travel length. Even though we are going to simplify the system and assume that the input and the output of this system are the only factors, and the transfer of energy is the temperature difference for $T_2 - T_3$ in stabilized operation. But in reality, step functions and high fluctuation will have a small slowness effect on this system before the output reacts accordingly (condenser output slowness hysteresis).

Point 3: In point 3 we are still having a high pressure, but the temperature has dropped due to giving off energy in condenser, and ideally we are in a liquid phase, and the refrigerant gets new enthalpy and entropy values based on the drop in temperature and the final settling temperature. (Note: In a constant pressure the temperature change equals change in enthalpy). These are not that important for us even though it is easy to find them for a saturated liquid in the the saturated vapor refrigerant table since the liquid in point 3 is on its way into the expansion valve.

Point 4: After the expansion valve the refrigerant is mostly gas, but also partly liquid. We want to have saturated gas in point 1 so from point 4 to point 1 the refrigerant is going to go through an evaporator that takes up energy from the cold side system until all the working fluid has become gas. Since we are setting up equilibrium equations such that between points 2 and 3 there is constant pressurisation, then as a result point 4 to 1 must have constant pressurisation as well. To ensure that we only have gas at point 1, we can put it through a "S" neck pipe with a thermocouple that measures the temperature of the refrigerant. This is necessary because not all compressors are designed for liquid intake, like most screw type compressors. But some compressors are capable of the strain it will have on it. In that case the safety mechanism is not needed.

Parts models: There are as we discussed a lot of individual parts we must consider in this model. And we are now able to phase out and model all the different components in this build. The compressor efficiency is composed of the loss of energy in mechanical strain and electrical use, meaning that compressor efficiency (unitless) is a product of electrical energy efficiency of the compressor and the mechanical efficiency of the compressor. Explanation of the variables and their units is available in the nomenclatures section.

$$\text{Compressor efficiency} = \eta_{elec} * \eta_{mech} \quad (2)$$

Net compressor power (kW) is a product of mass flowrate of the refrigerant (kg/s) and the refrigerant gain from the compressor. Put together with the compressor effect formula, we get:

$$\text{Net compressor power(kW)} : \dot{W}_{net} = \dot{m}_r(H_{comp} - H_r) \quad (3)$$

Approximation to H_{comp} by using isentropic values/process of system.

$$H_{comp} = H_r + \frac{H_{comp-isen} - H_r}{\eta_{comp-isen}} \quad (4)$$

Power consumed by compressor in the process:

$$\dot{W}_{comp} = \frac{\dot{W}_{net}}{\eta_{elec} * \eta_{mech}} \quad (5)$$

The heat rejection rate in the condenser is calculated by the next 3 equations: The heat transfer (kW) from refrigerant in the system.

$$\dot{Q}_r = \dot{m}_r \frac{H_{comp}}{H_{cond}} \quad (6)$$

The condenser energy capacity (kW) is determined by the return fluid from the warm side system.

The specific heat value of water (Cp_{water}) can found in table 1. Note how little (Cp_{water}) changes over the scope from 0°C to 150°C in figure 20.

$$\dot{Q}_{cond} = \dot{m}_{water} C_{P_water} (T_{water_out} - T_{water_inn}) \quad (7)$$

In the system we would have to include the energy used by pumps in the case of a an incline in height. This value is not going to significantly change after installation of the system. Therefore we will not take it in account in our system at all.

Temperature - T (°C)	Isochoric Specific heat of H_2O C_v (kJ/(kg * °C))	Isobaric Specific heat of H_2O C_p (kJ/(kg * °C))
20	4.1570	4.1844
40	4.0737	4.1796
55	4.0048	4.1824
60	3.9767	4.1851
80	3.8726	4.1969
100	3.7682	4.2157

Table 1: The Specific heat values of water (H_2O)
[25](EngineeringToolbox, 2021)

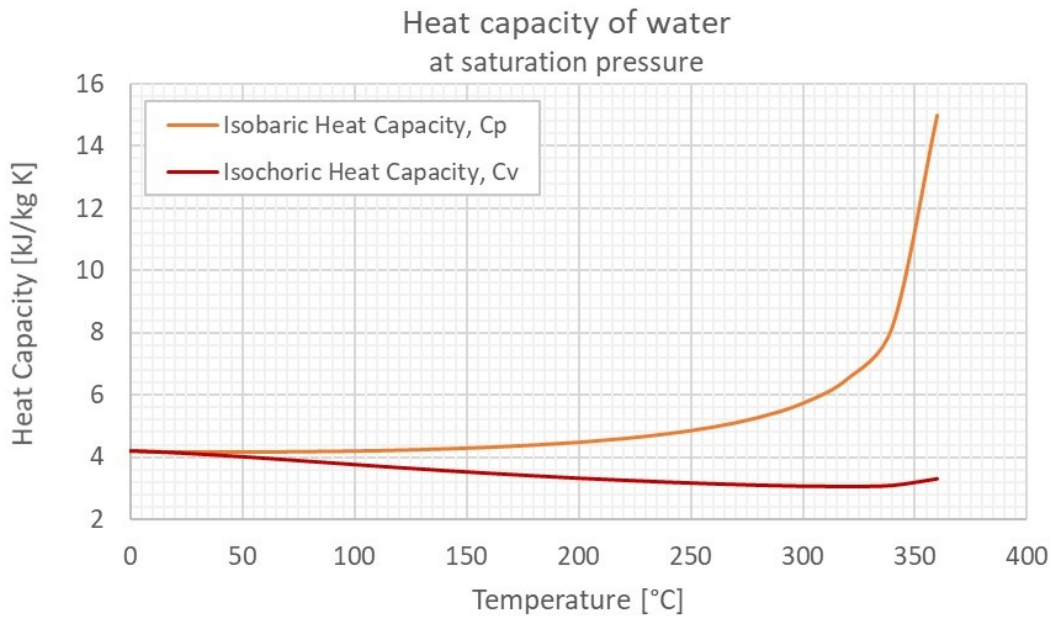


Figure 20: The heat capacity of water
[25](EngineeringToolbox, 2021)

$$\dot{Q}_r = \dot{Q}_{cond} \quad (8)$$

To simplify the approximation of the heat transfer from refrigerant to the warm side water we are going to say that $\dot{Q}_r = \dot{Q}_{cond}$. And since we can say that all the energy is stored in the system and that the heat transferred from the refrigerant to the warm side water is the same as the condenser capacity, we are greatly decreasing the complexity of the mathematical model. This is not far from the truth for a modern heat transfer system with a high thermal retention, but even a modern well isolated system will let energy escape into the surrounding environment. But the escaping energy in this system is so small that the statement $\dot{Q}_r = \dot{Q}_{cond}$ is accurate enough and can be negligible rounding error. That means that \dot{Q}_{cond} is only calculated by its physical elements as the total heat coefficient and transfer area, the correction coefficient and logarithmic temperature differences of the condenser.

$$\dot{Q}_{cond} = K_{cond} A_{cond} F_{cond} \Delta T_{ld_cond} \quad (9)$$

ΔT_{ld_cond} is the auxiliary equations for the logarithmic temperature difference in the temperature of the condenser. It is used to the temperature driving force in the flow system.

$$\Delta T_{ld_cond} = \frac{(T_{cond_inn} - T_{water_out}) - (T_{cond_out} - T_{water_inn})}{\ln \frac{T_{cond_inn} - T_{water_out}}{T_{cond_out} - T_{water_inn}}} \quad (10)$$

The same equation can be written for the evaporator. These equations will be part of the equation when we make the general mathematical model of the system, but we are going to focus on making a steady state system of the heat pump to warm side system later. Which are mostly derived from the condenser and not the evaporator which we require and have already checked that it has the the ability to deliver enough energy. The heat transfer rate of the evaporator is:

$$\dot{Q}_{evap} = \dot{m}_r \frac{H_r}{H_{evap}} \quad (11)$$

$$\dot{Q}_{evap} = \dot{m}_{brine} C_{p_brine} (T_{brine_inn} - T_{brine_out}) \quad (12)$$

$$\dot{Q}_{evap} = K_{evap} A_{evap} F_{evap} \Delta T_{ld_evap} \quad (13)$$

$$\Delta T_{ld_evap} = \frac{(T_{brine_inn} - T_{evap_out}) - (T_{brine_out} - T_{evap_inn})}{\ln \frac{T_{brine_inn} - T_{evap_out}}{T_{brine_out} - T_{evap_inn}}} \quad (14)$$

The expanded refrigerant after the expansion valve has the same enthalpy as the refrigerant before the expansion valve:

$$H_{cond} = H_{expa}$$

The first law of thermodynamics dictates conservation of energy and we can derive this equation for the heat pump system:

$$\dot{W}_{comp} + \dot{Q}_{evap} = \dot{Q}_{cond}$$

COP or coefficient of performance (unitless) describes the ratio of the useful heat transfer from the heatsink/ condenser, based on the total energy input, which in our case is the electric power (kW) from the compressor.

$$COP = \frac{\dot{Q}_{cond}}{\dot{W}_{comp}} \quad (15)$$

5.2 Mathematical model ground source heat pump system

Now we can finally derive a general mathematical model for the whole system. The following equations are equilibrium equations that detail the energy balance at specific points at the heat pump system. Collectively it describes the complete system and we can derive the state variables from them.

$$\frac{\dot{m}_r (H_{comp} - H_r)}{\eta_{elec} * \eta_{mech}} + \dot{m}_{brine} C_{p_brine} (T_{brine_inn} - t_{brine_out}) - \dot{m}_r (H_{comp} - H_{cond}) = 0 \quad (16)$$

$$\frac{\dot{Q}_{cond}}{H_{comp} - H_{cond}} (H_r - H_{expa}) - K_{evap} A_{evap} F_{evap} \Delta T_{ld_evap} = 0 \quad (17)$$

$$K_{evap} A_{evap} F_{evap} \frac{(T_{brine_inn} - T_{evap_out}) - (T_{brine_out} - T_{evap_inn})}{\ln \frac{T_{brine_inn} - T_{evap_out}}{T_{brine_out} - T_{evap_inn}}} + \frac{\dot{m}_r (H_{comp} - H_r)}{\eta_{elec}} - K_{cond} A_{cond} F_{cond} \frac{(T_{cond_inn} - T_{water_out}) - (T_{cond_out} - T_{water_inn})}{\ln \frac{T_{cond_inn} - T_{water_out}}{T_{cond_out} - T_{water_inn}}} = 0 \quad (18)$$

The problem with this solution is that, the energy balance equation we have derived does not have the true complexity of the condenser system. Since we have a substance that is super-heated, and is in the process of giving out energy, and will eventually change state back to a liquid. That means that we do not have one system but a minimum of 2 systems depending how deep one wants to go to explain and to simulate a system. Luckily for us we have a myriad of books, articles and theses that explain how to derive an equation closer to the truth. We discuss this model further in Chapter 7.

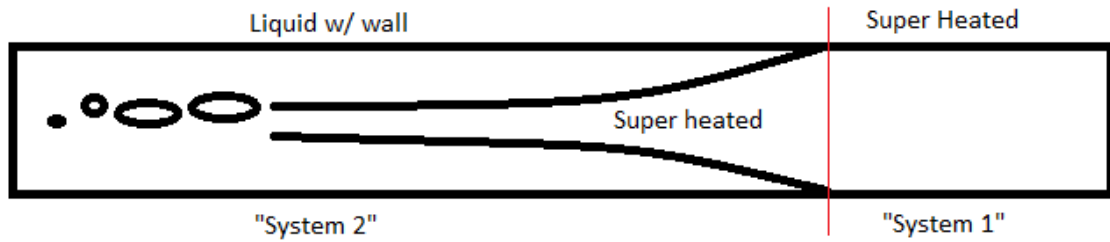


Figure 21: The super heated (system 1) and the phase shifting system (system 2)

5.3 Mathematical model air-conditioner/ radiators

Radiators/ heater panel/ condenser and a lot of other heat transfer units can be described using and formulated very similarly to equation 7 at 5.1. These systems only have different efficiency numbers of heat transfer. Some systems are slower and need mere exposure or a higher flow/ temperature. But from a regulation perspective these systems do have different gain values.

Temperature - T (in °Celsius)	Isochoric Specific heat of air C_v (kJ/(kg * °C))	Isobaric Specific heat of air C_p (kJ/(kg * °C))
-13.2	0.7168	1.006
0.0	0.7171	1.006
6.9	0.7173	1.006
15.6	0.7175	1.006
26.9	0.7180	1.006
46.9	0.7192	1.007

Table 2: The specific heat values of air [26](EngineeringToolbox, 2021)

The condenser energy capacity (kW) is derived by the return fluid from the warm side system.

$$\dot{Q}_{AC} = \dot{m}_{air} C_{P,air} (T_{AC,AO} - T_{AC,AR}) \quad (19)$$

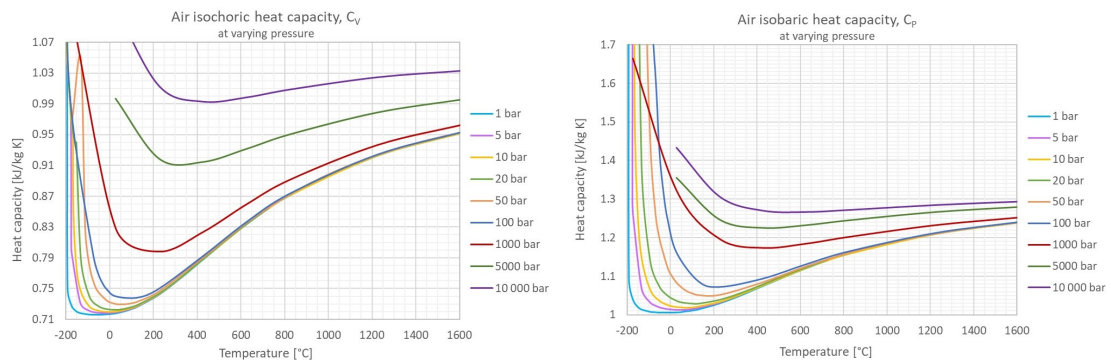


Figure 22: Air's isochoric and isobaric heat capacity [26](EngineeringToolbox, 2021)

Since the only pressure that air is going to receive is the atmospheric pressure, and as shown in table 2 and in figure 22 these values are not going to drift that much that we are going to be taking account of it.

6 Testing and Simulation

Since we did not have a possibility to change system parameters for the real life system, we have not acquired the necessary measurements. So to test our hypotheses developed in earlier chapters, we have to rely on theoretical models. As the system consists of multiple components and involves complicated thermodynamics, we are going to explore the possibility of simulating the system (or specific parts of it) with software intended for this purpose.

Our search has revealed that there is a multitude of such software. We are referring to a report from Sintef about software for modelling and simulation of ground source heating and cooling systems [27](Sintef, 2016). While the report mostly focuses on borehole modelling, there is an overlap with heat pumps. However, it is still necessary to evaluate how well any software would fit the purposes our project. Our main criteria were:

- The availability of the software – The software should be possible for us to obtain. While there are some open-source alternatives, most of the software has significant licensing fees.
- Ability to implement control engineering solutions - Many of the currently available programs are entirely focused on the HVAC aspects, with rather limited choices with regards to control engineering.
- Ease of use – Modelling of a system often involves taking care of low-level aspects and underlying thermodynamics. Therefore, the program(s) should have a degree of abstraction that makes it viable to use for an automation engineer.

After comparing the software mentioned in the Sintef article and accounting for our criteria, we decided to focus on a simulation software named TRNSYS. The main reason being its widespread use in academia and the fact that we were able to obtain licenses from our university. Initial research suggested that also our additional criteria could be met.

6.1 Simulation - TRNSYS

TRNSYS is a collection of software indented for the simulation of transient systems. While it can be utilized to model various dynamic systems, its main focus is the field of thermodynamics. Originally released in 1975 as tool to solve various algebraic and differential equations in solar energy systems, TRNSYS has seen 18 version iterations. It has a widespread use in academia. TRNSYS keyword search in scientific article search engines returns several thousand results.

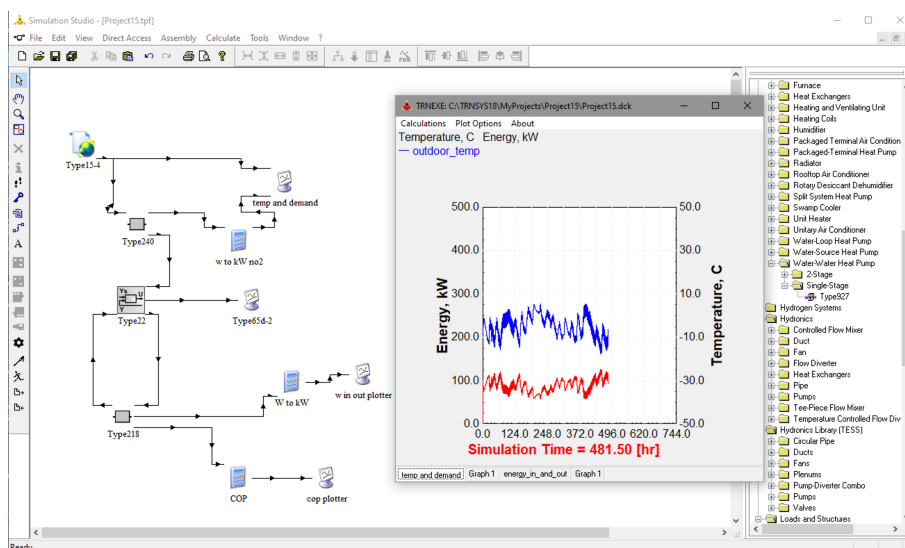


Figure 23: TRNSYS interface

6.1.1 Modularity and cost

One of the main selling points of TRNSYS is the modularity of the software. TRNSYS has a multitude of premade components that can be used to model various systems. Presented here is selection of some components:

HVAC – boilers, cooling coils, furnaces, heat pumps;

Power sources – batteries, photovoltaic panels, wind turbines;

Hydronics – fans, pipes, pumps;

and several other groups, as building loads, storage tanks, international weather data and even hydrogen systems. Each component has several parameters that can be adjusted to scale the components to the needs of the user.

The total cost of the software depends on the aforementioned modules. As per the time of writing this thesis, the base version single-user commercial license costs \$5060. It includes a significant amount of components. Library components developed by TRNSYS partners can be bought at an additional cost.

6.1.2 Learning

Due to the in-depth nature of the software, there are various proficiency levels one can achieve. It is reasonable to expect for an end-user to be able to conduct simple simulations with the in-built components after a relatively short introduction course.

However, from our personal experience, developing own components is a significant undertaking as one has to fully cover the concerned field. Besides the mathematical representation of the component, it is necessary to understand the underlying working principles of the TRNSYS simulation.

The learning resources we have used in the scope of this project consist mostly of the TRNSYS user manual and a limited selection of online materials. This has made the learning process rather challenging at some points. It has to be noted, that TRNSYS and its associated partners offer online and on-site training for the program, which would be a preferable option before using the software for commercial purposes.

6.1.3 Interface

Besides many advantages due to the modularity of TRNSYS, there are inherent drawbacks resulting from its fragmentation. Debugging and troubleshooting can be difficult, because of lacking debugging tools compared other common programming compilers. Due to the way of the underlying software, the components have to be compiled beforehand. However unexpected issues can arise at runtime, thus the types have to be compiled again.

Unfortunately, unsanitized variables often were the cause of unexpected errors and undocumented issues. Additionally, we encountered several instances of software not responding or crashing. This in combination with the lack of automatic save feature or integrated version control can pose a risk of losing important portions of a project.

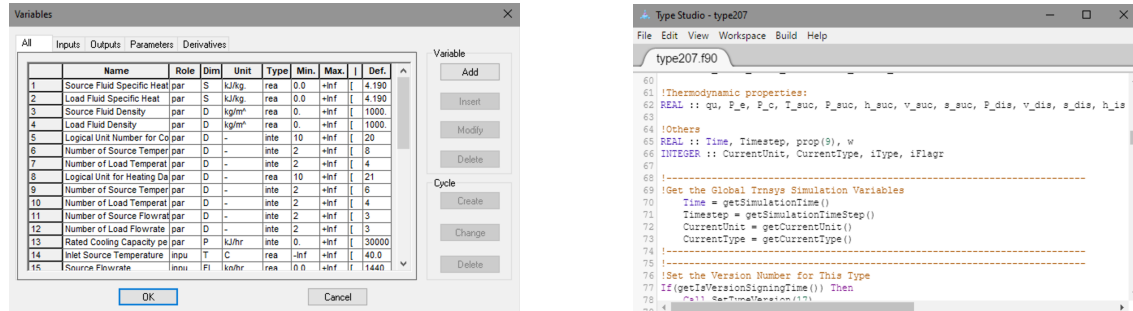
The software does not take advantage of modern graphical user interface design principles of the last decade, thus the workflow often can be cumbersome and inefficient compared to design and simulation software of other engineering fields.

An important point however can be made - whether such criticisms are valid knowing the limited userbase and the niche use of the software? The definite conclusion is essentially dependent on the specific use cases. In our situation, the lack of any another alternative software nullified the user-friendliness concerns of TRNSYS.

It also has to be noted that while the version used in this thesis (18.00.0017 / 6.0.0.7) is the latest major version; we have not tested whether any following minor updates might have alleviated some of the crashing issues mentioned before. Potential issues can be discussed on a free TRNSYS mailing list [28](onebuilding.org, 2021).

6.1.4 Custom components

User has the ability to develop and define new components and to edit the source code of existing components. Purchase of TRNSYS also includes the source code of its kernel as well. TRNSYS allows programming of the components in programming languages that support .dll structure, namely Fortran and C++. However, in practice nearly all publicly available TRNSYS learning resources only use Fortran. TRNSYS is compiled with Intel Visual Fortran. However, user customized component code can be compiled with gFortran. [29](TRNSYS, 2017)



(a) Multiple properties of an example component

(b) Fortran90 environment used for programming of custom components

Figure 24: Component creation in TRNSYS

Therefore, when considering the use of TRNSYS in a commercial setting, one should account for the user willing to learn basic use of Fortran programming language. Fortran stands for “Formula translation” and it has been historically used in engineering and numerical computation settings. TRNSYS uses Fortran 90 standard from 1991, which despite its age can be considered as “modern Fortran”. It is a free-form language, and it does not have the strict character positioning rules of Fortran 77 that were necessary to ensure a compatibility with punch cards (a common criticism of older Fortran versions)

We find it important to mention that development of additional components most often consists just of translating mathematical formulas and conditional statements. It is relatively easy to formulate such tasks in the Fortran language, especially if one has general background knowledge of any other of programming languages.

The inherent difficulty however arises when interfacing the component with the TRNSYS itself and other components. This entails an understanding of various functions and subroutines employed by TRNSYS kernel, because they are often called from the user supplied components. TRNSYS Programmer’s Guide – Volume 7 offers valuable insight in this.

6.1.5 Simulation of variable speed heat pumps (VSHP)

TRNSYS and TESS (a private TRNSYS developer entity) component libraries offer a wide array of air-air; air-water and water-water heat pumps. However, all of them are available only in single stage or two stage configurations. Single stage configuration refers to a heat pump with a compressor that has fixed rotational speed. Therefore, the heat pump will always run at a full capacity, even if the heating demand does not require it. Two stage compressor partly alleviates the issue by offering two levels of operation.

However, recent industry developments have made variable speed compressors more widely available and affordable. A variable speed compressor can adjust its speed to fulfil the exact heating demand. This benefit comes at an increased complexity of the heat pump, which in turn increases the cost of installation. However various research suggests that the benefits and energy savings of VSHP offsets the higher installation cost. [30](EPRI, 2014).

To effectively utilize the benefits of VSHP, it necessitates the use of a suitable control strategy, which is a significant point of interest in this thesis. Our initial research confirmed TRNSYS as a well-fitting software for this project, except for the lack of the said VSHP.

After an extensive search, we concluded that there is no publicly available VSHP component. Nevertheless, VSHP models have been utilized in several research papers, both in the context of TRNSYS and other software like Matlab. None of them however present source code or guidelines

for replication, besides their basic concepts and running principles. Developing such a component from scratch is by no means impossible, however it is beyond the scope of this thesis as it requires extensive thermodynamics modelling knowledge and a larger time-frame.

In our search we came across a master thesis from HVL [31](Vindenes, 2018) which examines the design of heat pump compressors in TRNSYS. Even though we had access to the source code of the said component, the implementation was not effortless and took a significant time. Part of the issues can be explained by our limited knowledge of TRNSYS; however, we encountered several inconsistencies in the behavior of the simulation. We can stipulate that this might be due to differing software versions, though this is not a definite conclusion. We did not research it further because we alleviated the issues by adapting the components by a different approach. Eventually we decided against using this model, mainly because it has its main focus on the compressor performance, however our interest lies in the temperature characteristics of a heat pump.

6.1.6 Control options in TRNSYS

As this thesis is control engineering oriented, we have also taken a look at the control components supplied with the software. As noted in TRNSYS mathematical reference files - there might be an issue regarding time steps and the PID component. The typical time step in a TRNSYS simulation ranges between minutes and hours. Its main use case are simulations over a longer time period, for example - seasons of a year. However, the sampling time of a typical commercial digital PID controller is much smaller, often down below to 200 milliseconds. It appears that TRNSYS is not optimized for such small frames, as evident by the long simulation times. Depending on the dominant time constant of the process, it might still be possible to have a larger timestep (and in turn larger sampling time). However, it has to be noted that the controller parameters and settings in this case would not apply to a real-life system and the controller would have to be re-tuned.

6.2 Simulation process

To evaluate various control scenarios, we have decided to simulate simplified configurations using TRNSYS. The following chapter explains commonly used components and reasons for our choice of various parameter values.

6.2.1 Heat pump

The heat pump used in those simulations is a Type927 component from TESS library. This is a single stage pump. Our reasoning for not using a VSHP is described in chapter 6.1.5. In addition - the data from our visit to the physical location indicates that for a typical winter day VSHP will run at full speed. Thus, this allows to simplify the simulation by using an existing component.

In regards to the cold side of the heat pump, TRNSYS has readily available components to simulate boreholes of various properties. However, as this task is automation oriented, the accurate simulation of boreholes would have no influence on data we are interested in. An intrinsic property of the boreholes is their stable temperature. While they do experience eventual temperature variation over several years, our simulation timeframes are much shorter, thus we have deemed it unnecessary to simulate them. This in turn shortens the computation time. The input temperature of the cold side is set to constant 7 degrees. Source flowrate is set to constant 18000 liters per hour.

The output temperature of the cold side is dictated by the inner workings of the heat pump, namely the compressor, the pressure and refrigerant. Figure 25a shows our simulated output of the cold side fluctuates around 4°C area, which corresponds to the design temperatures of a typical propane heat pump. The fluctuation magnitude is defined by the varying warm side inlet and they are acceptable for our design.

Heat pump component Type927 is based on empirical heating performance data of a heat pump utilizing R-22 working fluid. As discussed earlier in chapter 3.2.6, R-22 working properties are of a great similarity to R-290. We have contacted several companies asking if it would be possible to acquire heating performance tables of a R-290 heat pump. However, at the time of writing this thesis we haven't succeeded at receiving this data. A speculative reasoning for this might be the wish of the producers to limit publicity of this information, given the fact that it is relatively new technology. Therefore, all our further simulations are carried out with a R-22 refrigerant.

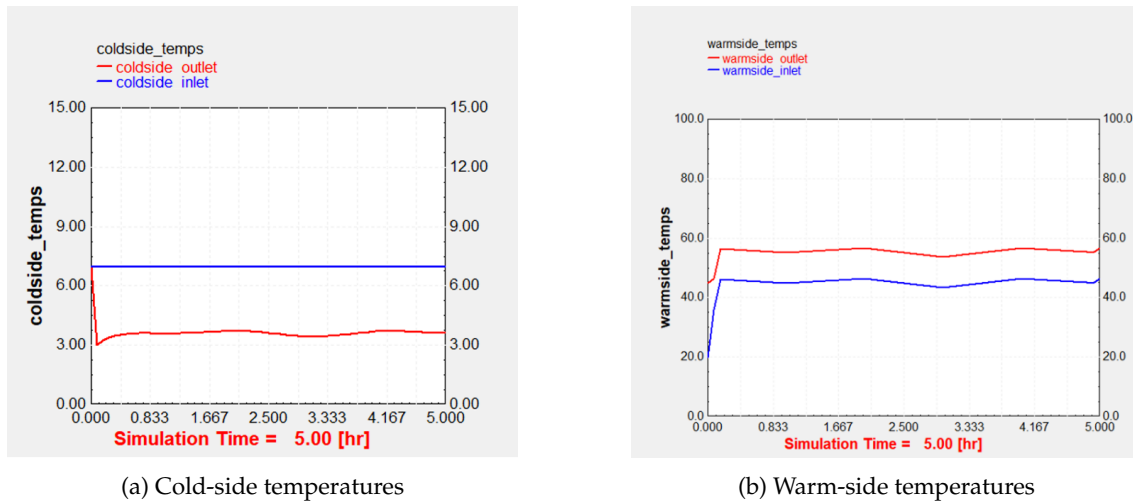
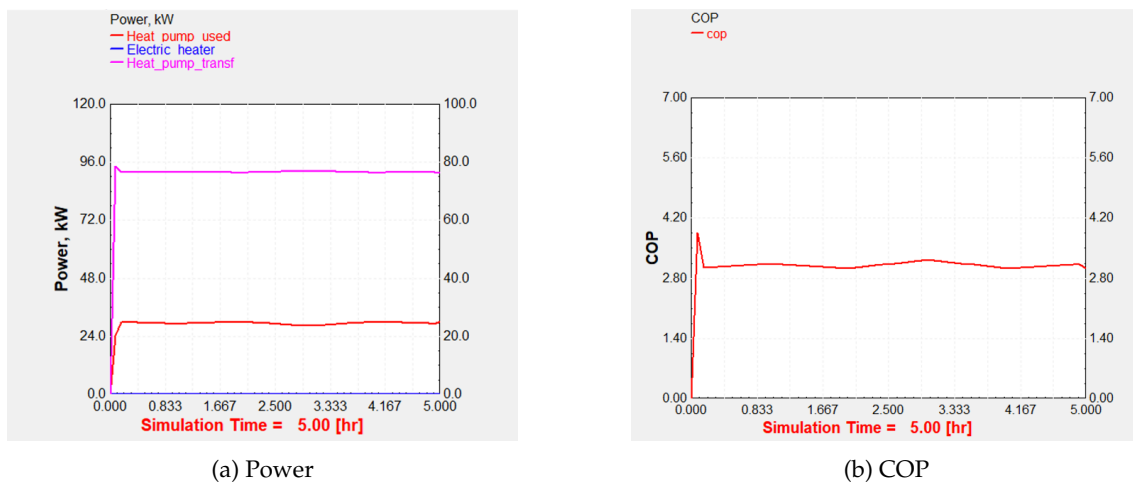


Figure 25: Heat pump temperatures

The output of the warm side is similarly dictated by the refrigerant and working pressures. However, it also has increased fluctuations depending on the load. In those simulations the inlet temperature is roughly 45°C and outlet is 55°C (figure 25b). This corresponds to the exact temperatures from the datasheet of an existing real life propane heat pump [32](ENRAD, 2018). While this is not an absolute indicator of the overall correctness of the simulation model, it confirms the basic expectations.

Finally – it is necessary to scale the heat pump. As we are trying to make the simulation as close as possible to the existing facility, we have experimentally concluded that the Type 927 heat pump component has to be scaled 12 times. Scaling in this context means running several heat pumps in a parallel combination. This is entirely for the simulation reasons, since this allows us to acquire larger heating output while utilizing heating performance data from a much smaller unit. In a real-life scenario, a single properly dimensioned heat pump would be used. (Additional note – it is also common to utilize parallel heat pumps in real-life whenever there is a necessity for heating needs bigger than what a single pump can supply. This however does not apply in our case).

Figure 26a shows that heat pump in full operation consumes around 30 kW. At the same time heat pump transfers heat energy that can also be expressed in kilowatts, namely – around 92 kW. This equates to the COP shown in figure 26b. A COP of 3 indicates that for every unit of energy consumed, 3 units of energy is transferred. It is not uncommon for modern heat pumps to achieve even higher COP, however the heating performance data used in the simulation stems from a heat pump model from early 2000s. [33](TESS, 2012) If data from a more recent heat pump were used, it would be reasonable to expect higher COP. Regardless, we are going to compare this model in different configurations against a model of the existing reference system. Therefore, this model satisfies our needs in this case.



(a) Power (b) COP

Figure 26: Heat pump power usage characteristics

6.2.2 Additional components

We have included additional components deemed necessary for this simulation:

Figure 27a: Cylindrical storage tank(s) utilized to simulate additional fluid in the system. While it is possible to simulate also individual pipes and their properties, for the simplicity of the simulation we have decided to use tanks with appropriate fluid amount. The tank simulates heat loss from its outer surfaces. The main purpose of the tank in our simulation is to simulate the buffer effect of water volume.

Figure 27b: Fluid pump used for the circulation of the warm side fluid. Flowrate is set to 12000 liters per hour.

Figure 27c: Auxiliary fluid heater used to increase the warm side outlet temperature. Since the maximum outlet temperature of this pump is 55°C, it might be necessary to heat it up more for various purposes - for example to heat the tap water which requires a higher temperature to avoid bacteria growth. In this simulation the setpoint is 60°C.

Figure 27d: Flow diverters of various arrangements that affect the flow direction and flow rate.

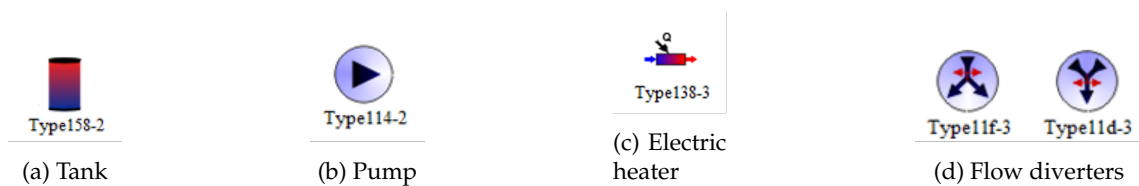


Figure 27: Additional components

6.2.3 Characteristics of a typical system

The real-life system has a significant number of individual radiators and subfloor heating courses. Simulation on component level would use significant computation resources as well as would take a significant setup time. Therefore, all the individual units are simulated by a single Type1231 (radiator based heating load) component. We are differentiating between floor heating and radiators with various parameters. The main differences are the design capacity and the delta T exponent with 1.0 being a characteristic of floor heating and 1.3 for wall mounted radiators.

As figure 28 indicates, there is a multitude of data which needs to be assigned to parameters, inputs and outputs of every component, as well as it needs to be interconnected with other components in the system. Even though the graphical programming TRNSYS simplifies the setup, a significant effort has to be taken to properly configure the custom elements of the system. This data is not always readily available because of company secrecy. In our case we have utilized data from existing installations where available, otherwise we have been required to do the dimensioning ourselves to recreate the existing physical facility.

Parameter	Input	Output	Comment
(series,tpf) radiators			
Name	Value	Unit	More Macro
1 Design Capacity	162000	l/h	More: <input checked="" type="checkbox"/>
2 Design Surface Temperature	45	C	More: <input checked="" type="checkbox"/>
3 Design Air Temperature	19	C	More: <input checked="" type="checkbox"/>
4 Design Delta-T Exponent	1.3	-	More: <input checked="" type="checkbox"/>
5 Number of Pipes	20	-	More: <input checked="" type="checkbox"/>
6 Pipe Inside Diameter	0.015	m	More: <input checked="" type="checkbox"/>
7 Air Pressure Exponent	0.5	-	More: <input checked="" type="checkbox"/>

Figure 28: Properties of the radiator component

To introduce disturbances in the system TRNSYS has the ability to use weather data (dry bulb temperature, humidity and others), however its main benefit is most evident when simulating over a timeframe of days to months. Thus, we are using a manual function generator that adjusts the temperature of the room in a more controlled fashion.

To compare the different hypotheses against something meaningful, we first had to make a system close enough to the real-world system. This system uses the same temperature set-points and delivers the same effect as the actual system. Additionally, we have made the 2 loads at appropriate values, where we have taken equivalent components such as two radiators at 35 kW and 55 kW which equal 90 kW, and other similar load simplifications.

Figure 29a shows that heatpump outlet reaches 55°C with return flow (warmside inlet) around the target of 45°C. It also shows that the auxiliary electric water heater helps heating up the return water, speeding up that start up phase, but eventually sees decreased power use due to reaching the setpoint. This shows that our simulated system reaches the ideal operational values (outlet and inlet temperatures of heat pump and power use) of the existing physical facility.

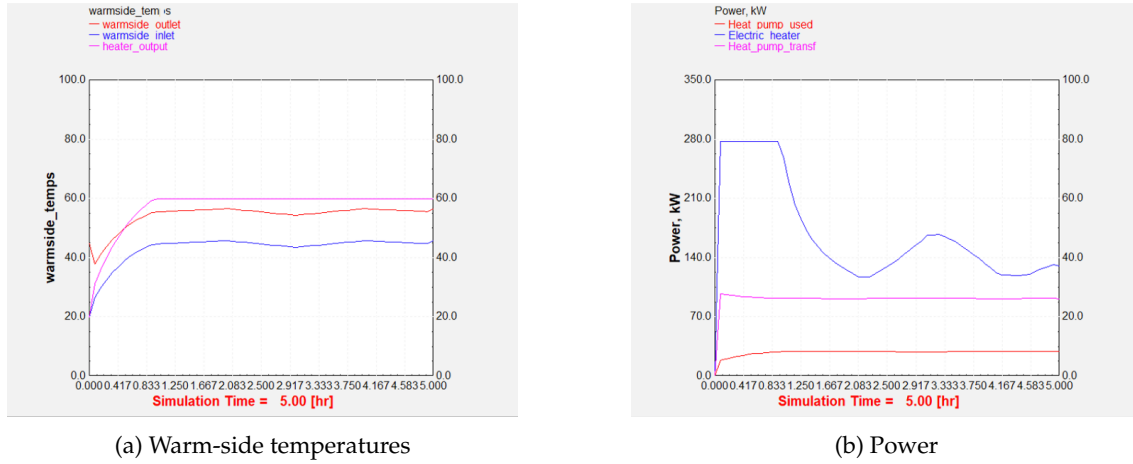


Figure 29: Heat pump characteristics

Figure 29b shows a notable power increase at 3.3 hours of the auxiliary electric heater. This is due to heat pump reaching its maximum possible performance, but still not satisfying the required temperature of 60°C. To alleviate this issue, an auxiliary electric heater can be programmed to compensate the peak loads. Due to the fast-changing nature of peak loads, an electric heater has much faster response than the heat pump.

There is notable visual short term amplitude difference between the simulated and real-life graph in figure 30. There are several factors which contribute to this behavior. Random noise is introduced in measurements. As well as the system experiences seemingly random fluctuations due to the way people use the facilities. For example, increased tap water usage during breaks. Additionally, our simulation uses an electric heater that can be fully modulated within its working amplitude. The electric heater used in the physical facility has distinct step-based heating levels. Therefore, it introduces delays in the system. Finally, our simulation has a lot larger timestep than the sampling time of real-life sensors, thus the fluctuations are smoothed out.

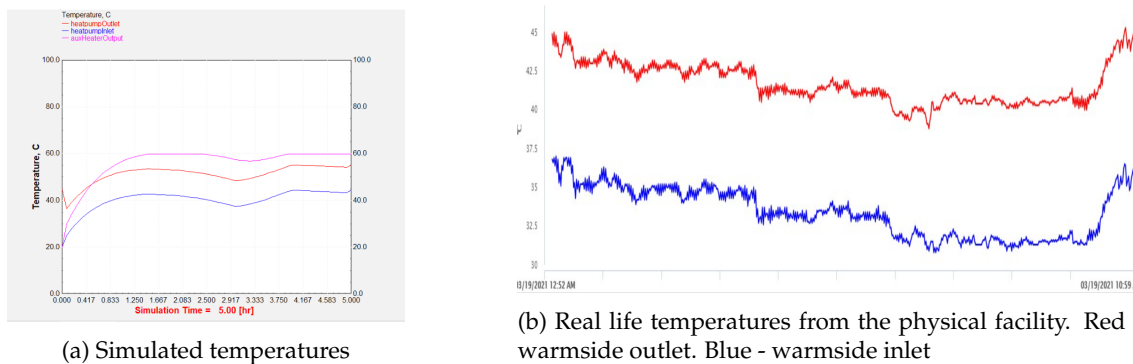


Figure 30: Temperature comparison

6.3 Simulation of proposed solutions

We are testing four distinct scenarios based on the discussion in Chapter 4.3. They have the following water distribution proportions:

"Standard system" - Tap water, radiator and floor heating is connected in parallel. (33% tap water, 33% radiators, 33% floor)

"Series system" - Tap water and radiators are in parallel, radiators and floor are in series. (33% tap water, 66% radiators and floor)

"Series with bypass" - Same as series system, but there is a bypass (33% tap water, 33% radiators and floor, 33% bypass)

"Parallel with heated tap water" - Tap water, floor and radiators are parallel, however only tap water is connected to the external heater (33% tap water (additional electrical heating), 33% radiators, 33% floor)

Note: The distribution values in those simulations do not necessarily reflect the values of a real-world system in the sense that the resultant values do not always match, but are within an acceptable error range. They have been arbitrarily chosen to see the limits and edge cases of the systems.

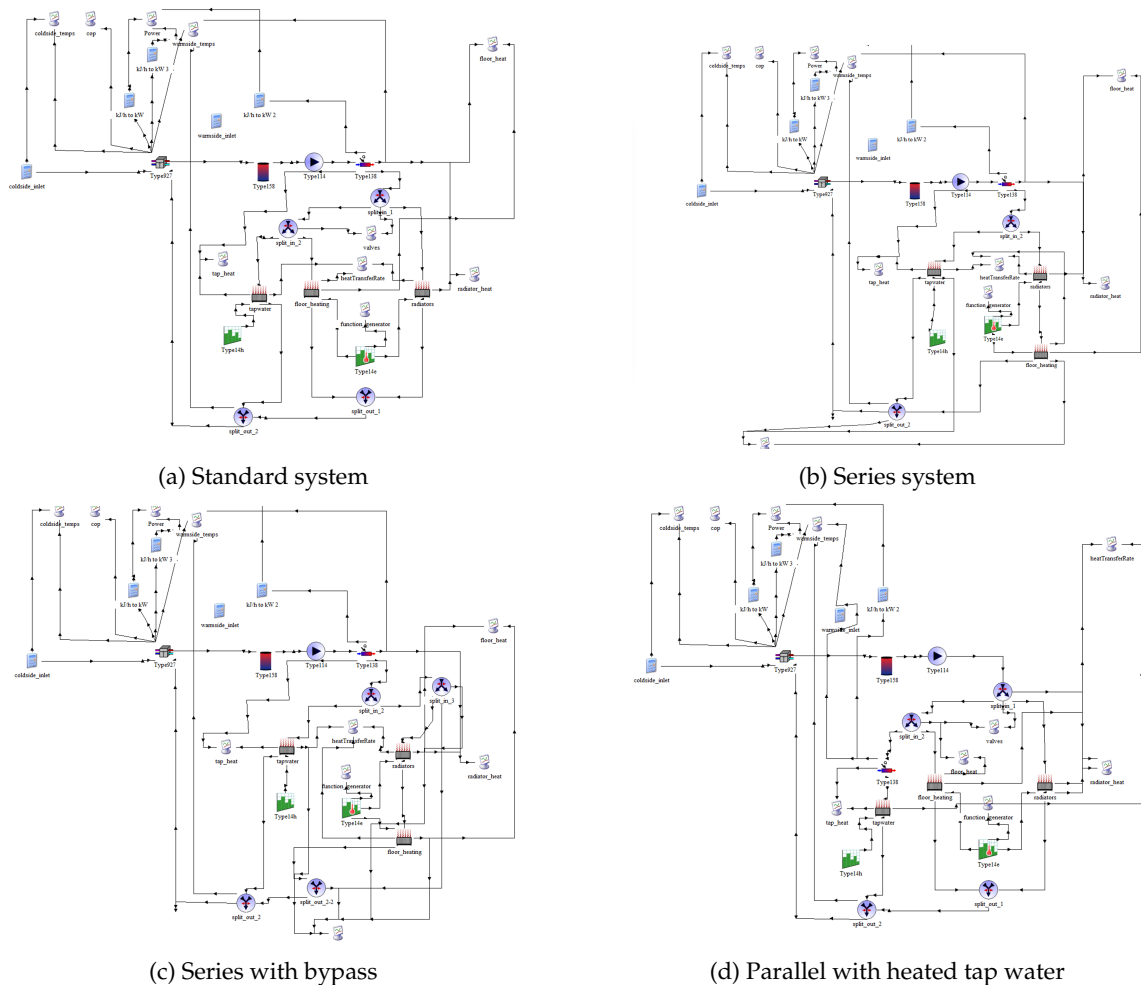


Figure 31: TRNSYS component layout for the proposed changes
Full size images and source files are available in Appendix B.

6.3.1 Results

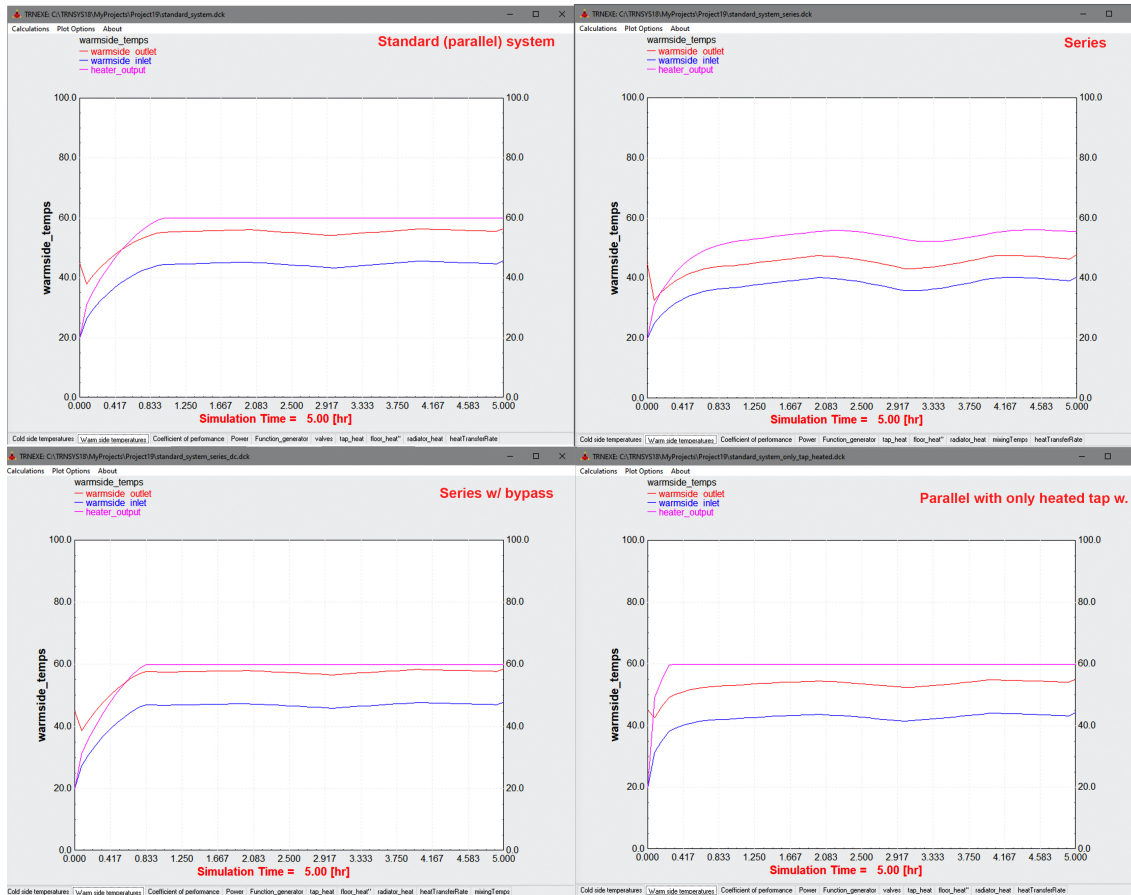


Figure 32: Heat pump inlet, outlet and electric heater outlet temperatures

The physical facility has the design temperatures of 55°C at outlet and 45°C at inlet. For our simulations it is important to get the same temperature range or exceed it. The reason for this being that if the return (inlet) temperature is too low, the electric heater will need to compensate this loss, in turn increasing the power consumption.

As indicated in figure 32, three of the systems can uphold the 60°C goal with the help of the electric heater, however, one of them ("Series system") drops below whenever there is additional load.

The "Series system" scenario is an artificial system which we have made to illustrate how detrimental too low return water temperature can be for the system. As we see in the "Series system" graph, the return temperature is below 40°C and as a result the delta energy that the heat pump can deliver (the area between red and blue lines) is much smaller compared to other systems, because it is outside of the optimal range for the specific heat pump. Furthermore, because of this, the system struggles to keep up with the demand.

If we compare the "Standard system" with "Series with bypass" system, we can see that the outlet energy is close to the purple line (electric heater) which means that the electric heater will compensate less and will reduce the energy usage significantly. We discuss the cost of this modification in the next section.

In a "Parallel with heated tap water" system there is a notable difference between the inlet/outlet and the setpoint 60°C line. This can be a misleading interpretation, because it is only 1/3 of the system water that gets heated from the outlet to the purple setpoint line.

6.3.2 Power

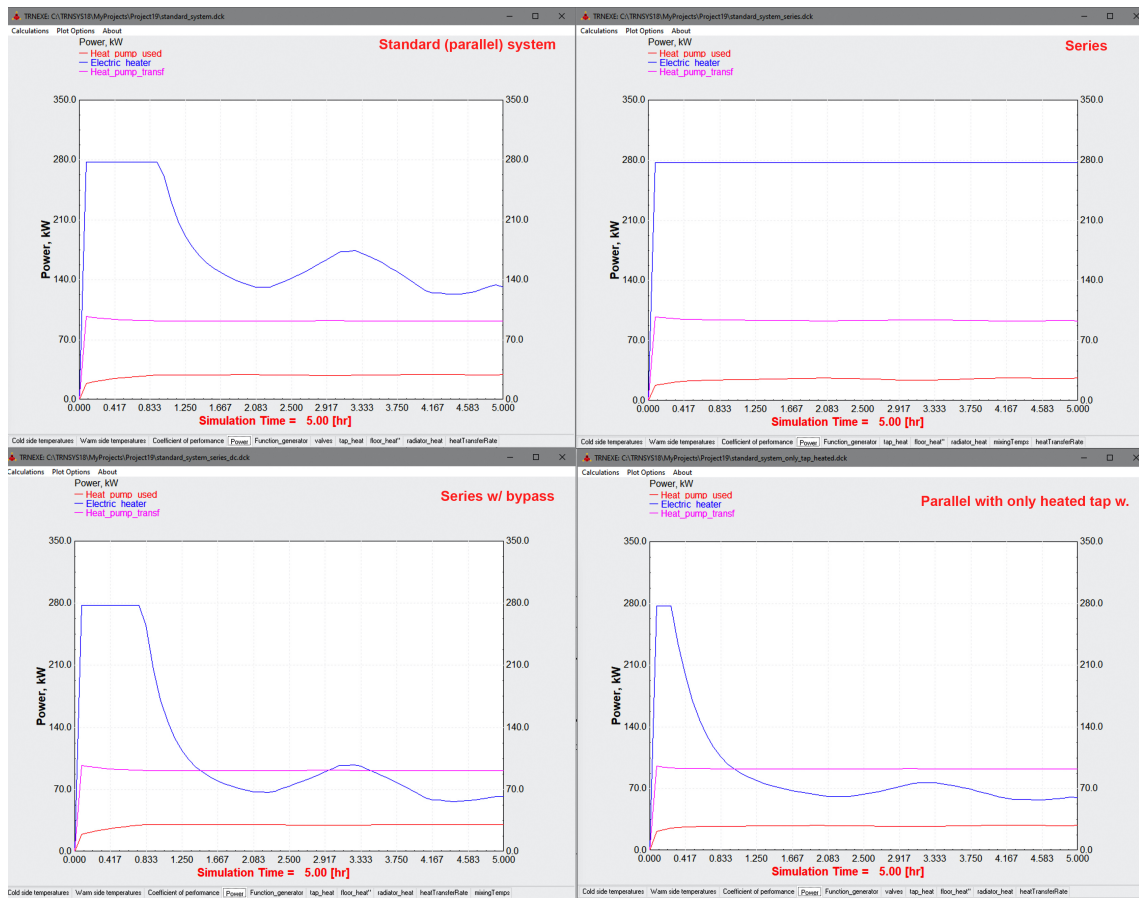


Figure 33: Power consumed and delivered by the heat pump and the power consumed by the electric heater

Even if the previous temperature graph indicated that the *"Parallel with heated tap water"* system had temperature shortcomings, in the figure 33 we can see that it actually might be the most cost-efficient system due to its decreased power use. It is also important to note that this reaches the setpoints quicker than other systems, because the heat pump inlet temperature reaches its optimal range much quicker. This way the heat pump maintains its most efficient temperature area, which in turn helps to deliver more heat to the system. This makes a more stable system that rebounds easier, however this does not imply a quicker heat up of the facility.

We can see from the figure 33 that we have significantly reduced energy use in auxiliary heater. It has to be noted that we cannot compare the simulation reference system directly with the real life system using precise quantities, since this might be wrong in an academical sense. However we can modify the simulation system and look at the resulting proportional changes and use that as guideline whether our hypotheses can be confirmed. In this case we can see that we have at its best close to 50% energy reduction in auxiliary heater. When taken in account with the rest of the system, this could give up to 20% energy use reduction for the whole system and this could be a worthwhile modification.

When comparing the *"Standard system"* with *"Series with bypass"* system, we can see a significant improvement in the latter. This system saves more energy than the *"Standard system"*, however it is still comparably slow regarding the speed or stability.

In all the scenarios the heat pump is running at its maximum output and an additional electric heater is necessary. This suggests that it might be of benefit to research systems with multiple heat pumps. In the artificial *"Series system"* system, both electric heater and heat pump quickly reach their maximum potential because of the unproportionable load it handles. It is not a practical scenario, however it helps with the comparison of other graphs, as it illustrates the potential negative attributes in an exaggerated manner.

6.3.3 COP

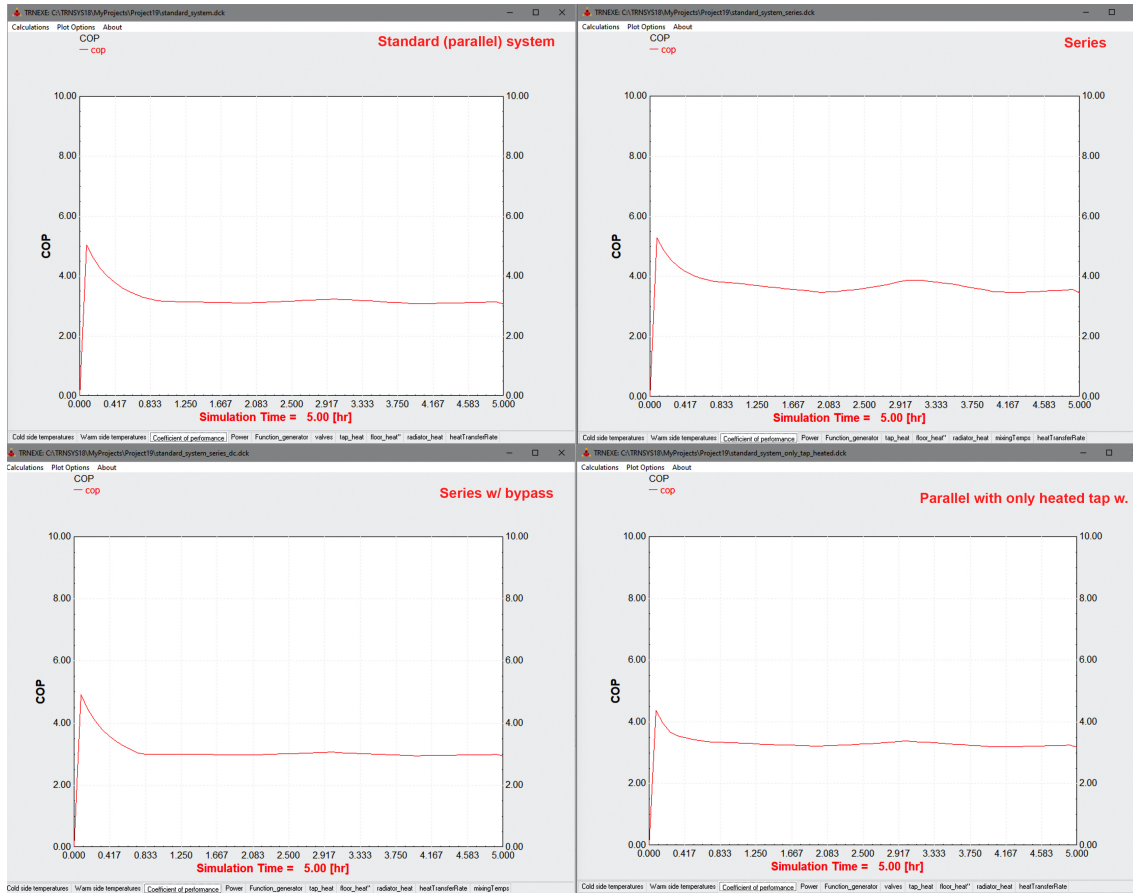


Figure 34: Coefficient of performance

The coefficient of performance measures how much energy we gain compared to the used energy and this obviously means that a higher value is better. The COP is rather similar in all scenarios mostly because the heat pumps are at max output and perform quite similar. An important factor on COP is the temperature of the cold side inlet, however, in this case we have few variations because of the stable borehole temperature (as compared to an air-water heat pump that would see significant COP changes as the outdoor air temperature changes). In addition, the linear cold side inlet temperature utilized in this simulation exaggerates this effect even more. Thus, the most notable COP changes stem from load variations.

6.3.4 Heat transfer rate

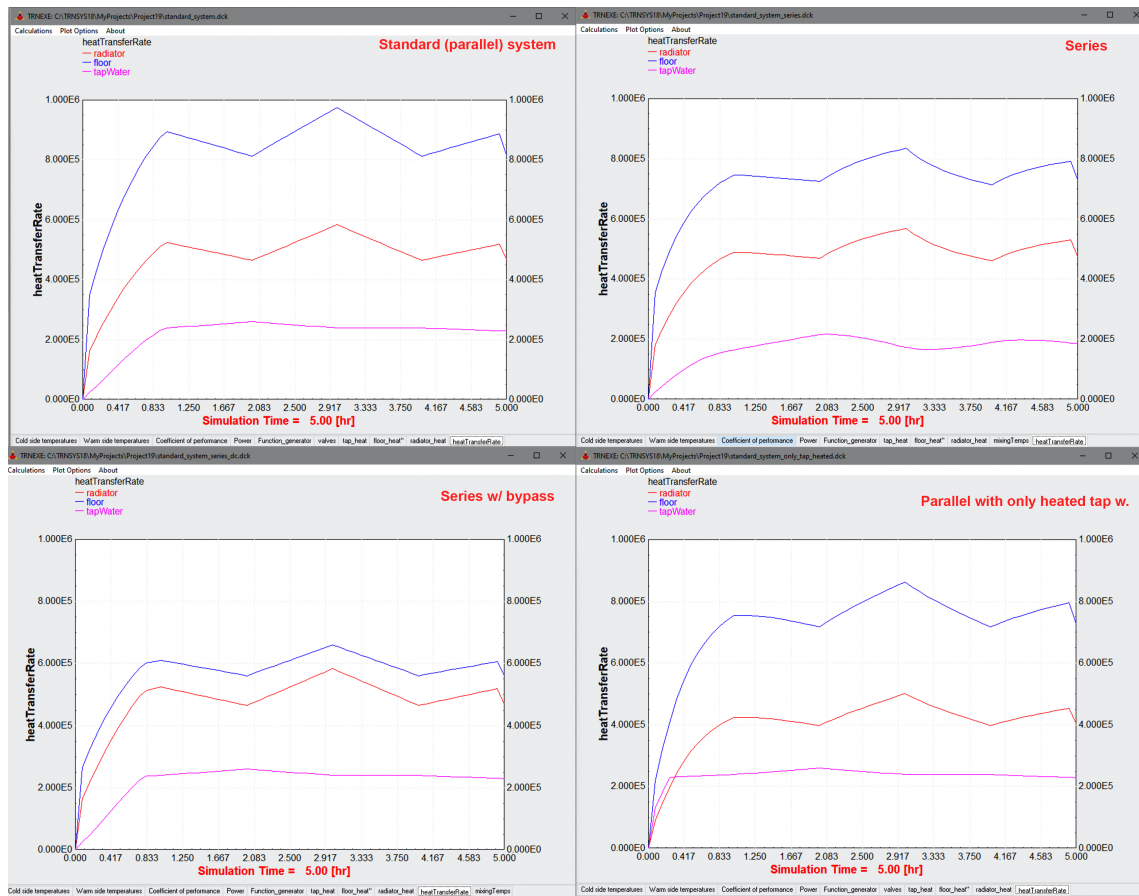


Figure 35: Heat transfer rates

While in the previous graphs the "Standard system" did not have the most favorable attributes, it is very notable in these graphs that "Standard system" is the "golden standard" in practical applications and is the default system of choice when designing and setting up a new system. This is due to its heat transfer abilities as evident in this graph. This setup is a proven solution that can be applied to almost any heating setting.

From our proposed scenarios the "Parallel with heated tap water" system comes as a close alternative. However, it has a pronounced difference in the heat transfer via radiators. So the energy saving benefits of such a system would be best utilized depending on the function of the building, namely the ratio of floor and radiator heating. For example, if the floor heating is more important, this system might be a better fit.

The "Series system" is quick to reach setpoint values, especially regarding the heat pump operational values, however even though we have good values, we see the indications from the curved tips of the graphs, that the system might be fast to respond, but heat transfer dynamics might be impaired. While it is barely noticeable here, it has implications for more severe effects in regards to failures or abnormal operational temperatures.

6.3.5 Tap-water temperature

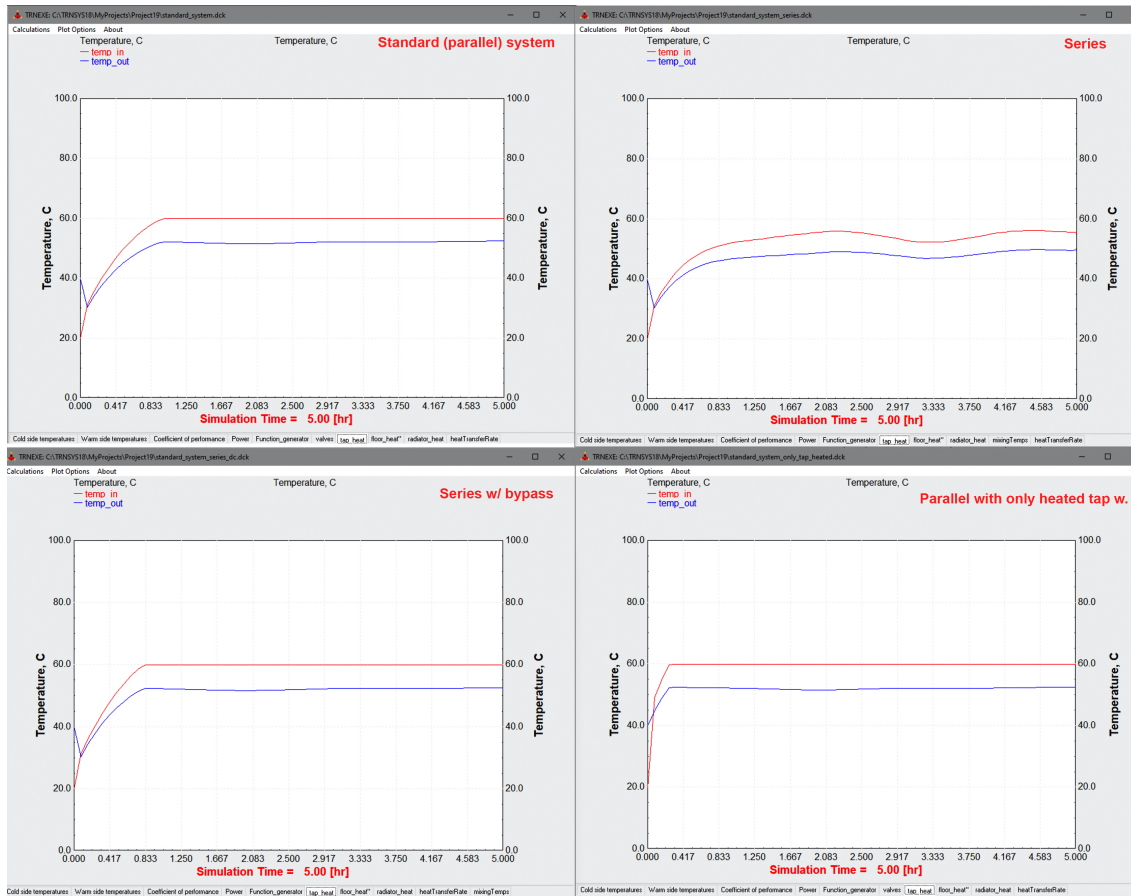


Figure 36: Temperature lost heating tap water

The working-fluid that heats the tap water is in all our tests at a minimum of 60°C. That is because both the tap water and the working fluid needs to be pasteurized, so no bacteria (especially legionella) and no other organisms can live in the water. Additionally, air-condition units need to be at a certain temperature to work as intended, and need to be in the 60°C branch of the system.

6.3.6 Floor temperature

Figure 37: The area between red (inlet) and blue (outlet) temperatures of the floor heating denote the energy supplied to the surroundings. The “standard system” has a clear advantage as it delivers a stable temperature because of the electric heater assist. The output temperature in “Standard system” and “Series system” systems has a higher temperature than our proposed systems, due to our strategy of delivering energy in a lower temperature range while still having sufficient heat transfer.

6.3.7 Radiator temperature

Figure 38: The same effect can be seen here, but in this setting, we see that “Series with bypass” is more responsive than “Parallel with heated tap water” system. Even though the area is not as big as in the “Standard system”, in a real-life setting, these systems will be driven at a lesser capacity. Even though the “Standard system” is able to give more at once, we can mitigate this in a real system by either having a higher rate of water exchange iterations or continuously modulating the flow.

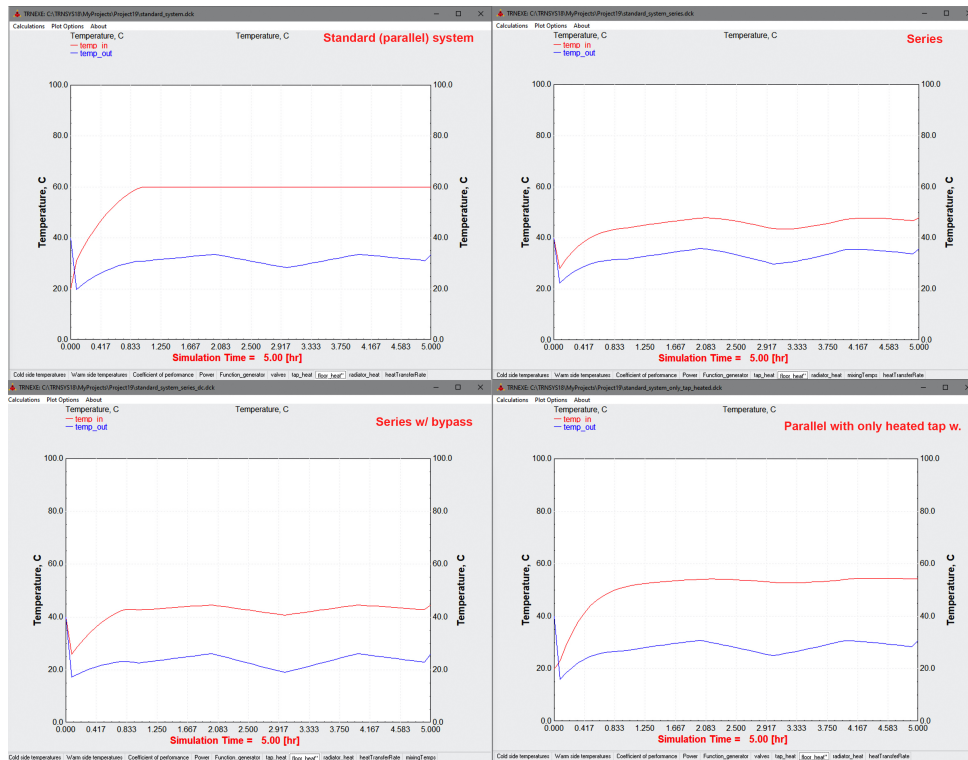


Figure 37: Temperature lost heating floors

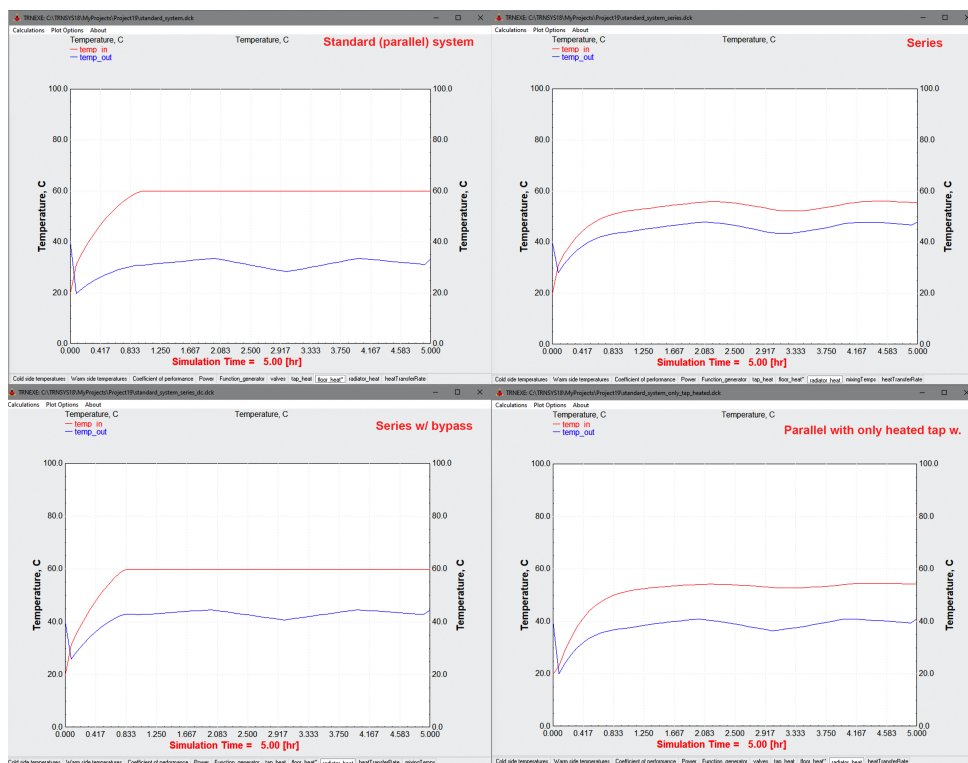


Figure 38: Temperature lost heating radiators

7 Control system engineering

Our initial plan was to solve the problem in Matlab/Python. Upon learning the significant workload and complexity of modelling thermodynamics and fluid mechanics, we chose to rather shift our attention to an existing simulation software. Nevertheless, we have chosen to illustrate some of the control problems we encountered and had to work around to do a simulation in TRN-SYS. This is meant to showcase the control nuances to Rambøll and for eventual expansion of the current project, since the task has a potential for additional research within MIMO/non-linear systems and/or production of a simulation program with the help of Python. We would advise also the use of a Simulink program to achieve fast and reliable feedback response when adjusting the system.

7.1 Discussion about the mathematical system

As discussed in chapter 5.2, the system needs to be divided in sub-systems, in different states. Our suggestion was to divide it in at least two systems. The super heated system and a system after phase shift (figure 21). This would mean that we either would have two steady state systems or a collective system made up of several sub-systems. So we have looked for articles which have already solved this problem and found [34](Wang et al., 2018) which explains how a similar system can be set up and regulated with robust control. Unfortunately their approach is beyond the scope of our bachelor thesis. We also found [35](Yao, Wang, and Huang, 2015), which have solved this problem in a similar fashion to our system. We can also see from [36](Yao, Huang, and Chen, 2013) that they have similar prerequisites and similar approach with a far stronger background in thermodynamics. It is our proposal to work with this system if further comparisons with the simulation results are desired.

7.2 Regulation of the system

To control the system, we must first find the potential issues in the system. This system is derived from an article by [35](Yao, Wang, and Huang, 2015). A notable problem in this system is that we have positive poles in several transfer functions. This poses difficulties for potential control.

Nowadays it is common to let the heat pump run at 100% power and linearize the system around an optimal working point. This simplifies the control since the system will not have notable drift and zero-pole cancellation, imaginary pole transformation, PID and robust control work well on the system. Even though pole-cancellation does not work well in the real world where values are changing and the system is outside of optimal conditions where cancellation does no longer apply, since system equation is not stationary in a dynamic system.

Therefore we see it as a difficult task to control all parts of heat pump system. There is an established practice to compensate with a secondary system, for example an auxiliary heater. Since as seen in COP graphs, the system fails (a noticeable avalanche effect occurs where the system cannot keep up). This happens when the system leaves its ideal conditions, even though it has a good control design. This has to do with the dynamics of the system itself and this is why there is an increased focus on regulation within this domain, with the goal of achieving high COP and maintaining the ideal working area of the heat pump.

7.3 The Matlab script

The program is designed to sequentially illustrate how the system is set up and shows the problem areas of this system. Program delivers the information in smaller "pieces" in the command window. The sequence can be advanced by pressing "Enter". Whenever there is a conditional statement, "Y" and "N" can be used too. The program is designed in a way that it is possible to skip any irrelevant information (a specific matrix or similar) for the potential user with the help of the said conditionals.

We hope the program will be a good way of showing the weaknesses of a system with the help of relevant values and graphs for persons who might not have full background of control engineering. Source code and the .m file is available in Appendix D. There is also a showcase of the content of the MATLAB script in Appendix C, the showcase can be a good indication if the script is a useful tool for an potential expansion work.

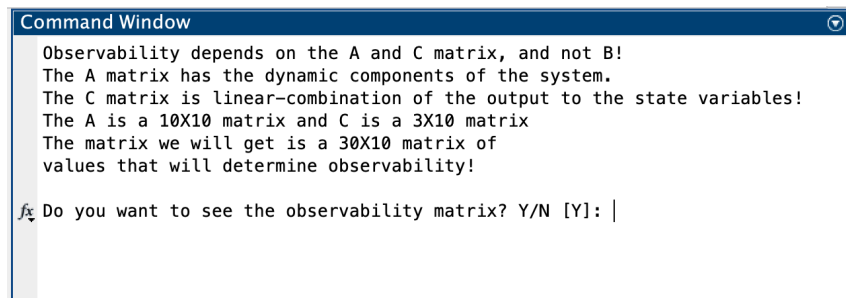


Figure 39: Illustration of the conditional interface

7.4 Proposal of potential thesis “Compressor speed control with continuous feedback regulation.”

As you probably have gotten the gist from this chapter, in the case of an opportunity to continue this research it would be beneficial to have a multi discipline thesis with more contributors with different academic backgrounds, for example - in energy systems and lab work. As well as cooperate with an another university or institution that has access to an already existing lab for testing the system and its model, or build a testing rig from scratch at HVL.

We would like to focus more and would suggest a potential additional work on the compressor speed control with continuous feedback regulation as the main topic. We have done the grunt part of the work, and following work would focus on further indepth parts.

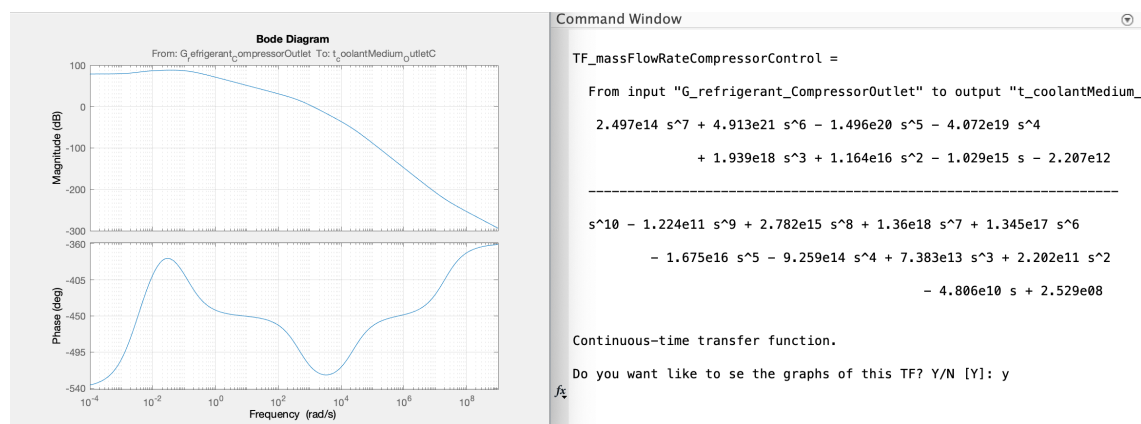


Figure 40: The magnitude and phase of the transfer function of the mass flow rate from compressor to the output temperature from compressor

The main motivation for this expansion is that there is a significant demand for engineers that can work with energy systems and regulation, in both the classical industry, home automation/ climate control and in the oil field. We feel that this area would benefit from more focus relating to control engineering.

8 Discussion

This thesis has been mainly a theoretical task and a significant part of it has been collection of already available information and applying it to our case. As evident from our visit to the physical facility, there is no doubt that such a system can successfully function. Therefore, the main reference points have been the indicators of how energy economic and practical the system is.

A topic of discussion is the values used in our simulations. Initially we hoped to compare simulation data with real-world data. However, as it proved to be not possible to attain the physical data, we shifted our philosophy regarding the simulation. We still kept the original heat pump dimensioning, however we simplified load side so that we could obtain more general characteristics of the system, as for example its breakpoints. This means that the simulated system can be a good indication of performance, but cannot be directly applied to a real-life scenario.

During the process of writing this thesis, the topic has presented several alleys of expansion. As the time is limited, we have had to consider which of these expansions were most relevant to the original goal. Likewise, there have been areas where we decided to not pursue the topic any further. For example, the development of VSHP component which proved to be too complex task, even though we had already devoted significant time to it.

9 Conclusion

There always is room for improvement in any produced work and this thesis is no exception. There are points about this work that we would wish were different. The fact that we did not get a startup ramp / step response from the physical facility has been a hindrance. As well as the fact that we did not have a physical test platform to try out our theories.

The limited access to academic help and physical presence limitations due to Covid-19 have also hindered several aspects of what we realistically could do in this timeframe in regards to thesis. For a bachelor level assignment, we must confess that this system's overall complexity has resulted in an overwhelmingly large workload, given that we underestimated the task. We digress, the work has been rewarding and exciting every step of the way.

The effects of having limited data on the system has been detrimental to the mathematical aspect, which would be used to test the theory against the reality. This in turn has directed our focus towards simulation tools to back up our claims, instead of the numerical approach that we originally had a planned for when we got the assignment.

It is our profound belief that this thesis has a great expansion potential as a new additional thesis if given the opportunity to work with a physical test rig. We hope that our MATLAB script and chapter 7 will help potential continuation of this study. The purpose of this MATLAB script is to make it simpler and faster for people to familiarise them with this system and its problem areas. Together with a test rig this could narrow down the complexity and loosen up the approach for making a generalized system explanation, that gives an overview of all propane systems.

The approach of utilizing a simulation is something that is more standard to do in our field today. We have been internally referring to TRNSYS as "an absolute monster" of a simulation tool, which can account for all counts of inputs and has significantly reduced the level of academic depth that we would need in the field of thermodynamics. The positive aspect of having a simulation tool that can increase the accuracy of our predictions and calculations far outweighs the one-time cost of the program, and the fact that it is convenient and modular for changes is only an additional advantage.

Overall, we are satisfied with the outcome of this thesis. Not only by the result of all our work, but also by the fact that we have been doing real engineering work that involves reading up on new and complex systems and the improvement of them using the knowledge from our background fields.

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- [36] Ye Yao, Mengwei Huang, and Jing Chen. "State-space model for dynamic behavior of vapor compression liquid chiller". In: *International Journal of Refrigeration* 36.8 (Dec. 2013), pp. 2128–2147. ISSN: 01407007. DOI: 10.1016/j.ijrefrig.2013.05.006. URL: <https://linkinghub.elsevier.com/retrieve/pii/S0140700713001187> (cit. on pp. 39, 50).

Appendix A - Tables & Diagrams

Propane - natural/ pure	(C ₃ H ₈)
Molecular Weight	44.097
Specific Gravity of gas (where air = 1)	1.52
Specific Volume	0.552 (m ³ /kg)
Density of liquid at atmospheric pressure	580 (kg/m ³)
Vapor pressure at 25°C	135.7 (psia), 0.936 (MN/m ²)
Absolute Viscosity	0.0074 (centipoises)
Sound velocity in gas	253 (m/s)
Specific Heat - cp	0.39 (cal/g°C), 1630 (J/kgK)
Specific Heat Ratio - cp/cv	1.13
Gas constant - R	188 (J/kg°C)
Thermal Conductivity	0.017 (W/m°C)
Boiling Point - saturation pressure 14.7 psia and 760 mm Hg	-42.2 (°C)
Latent Heat of Evaporation at boiling point	428000 (J/kg)
Freezing or Melting Point at 1 atm	-189.9 (°C)
Latent Heat of Fusion	44400 (J/kg)
Critical Temperature	96 (°C)
Critical Pressure	618 (psia), 4.26 (MN/m ²)
Critical Volume	0.0045 (m ³ /kg)
Heat of combustion	50340 (kJ/kg)

Table 3: The properties of Propane (C₃H₈)

Appendix B - TRNSYS simulation layouts

All files are available for download from: <https://github.com/boe28/boe28>

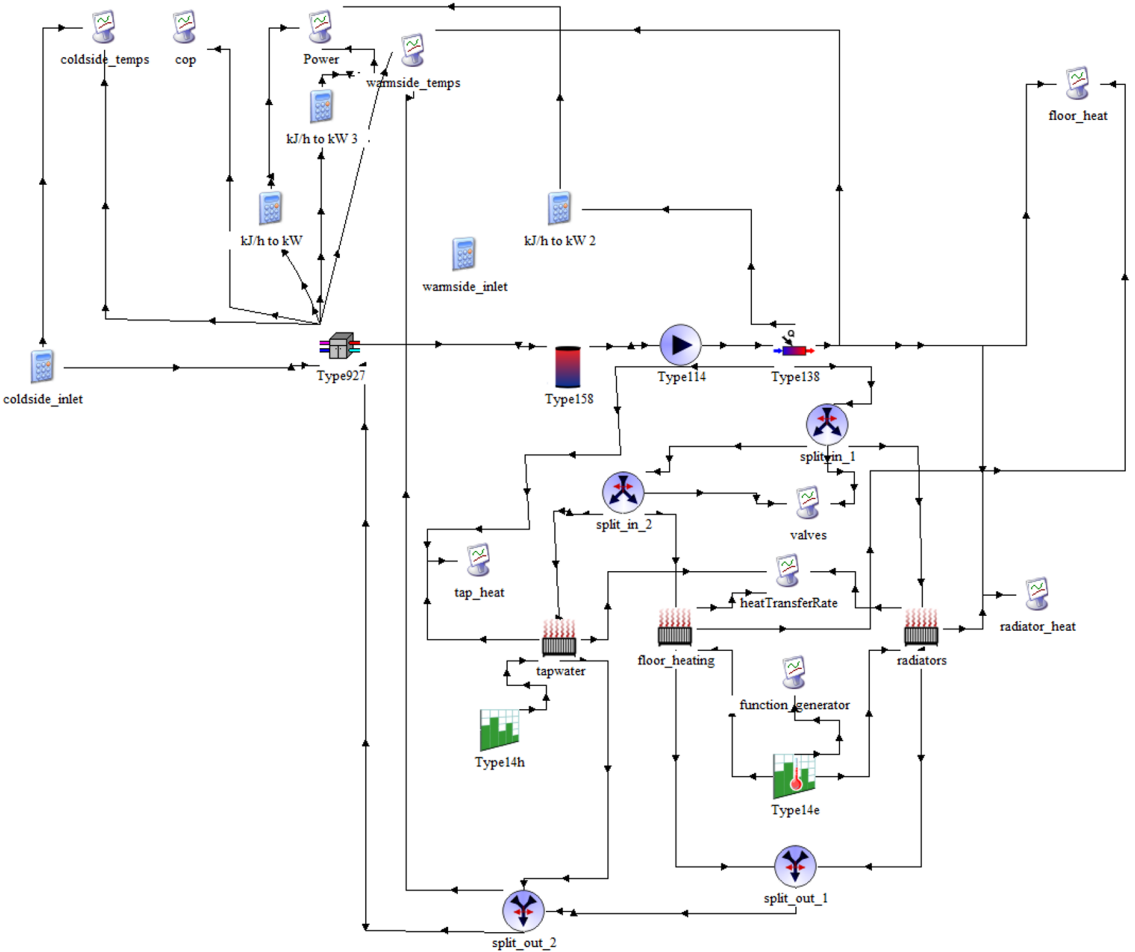


Figure 41: Standard system

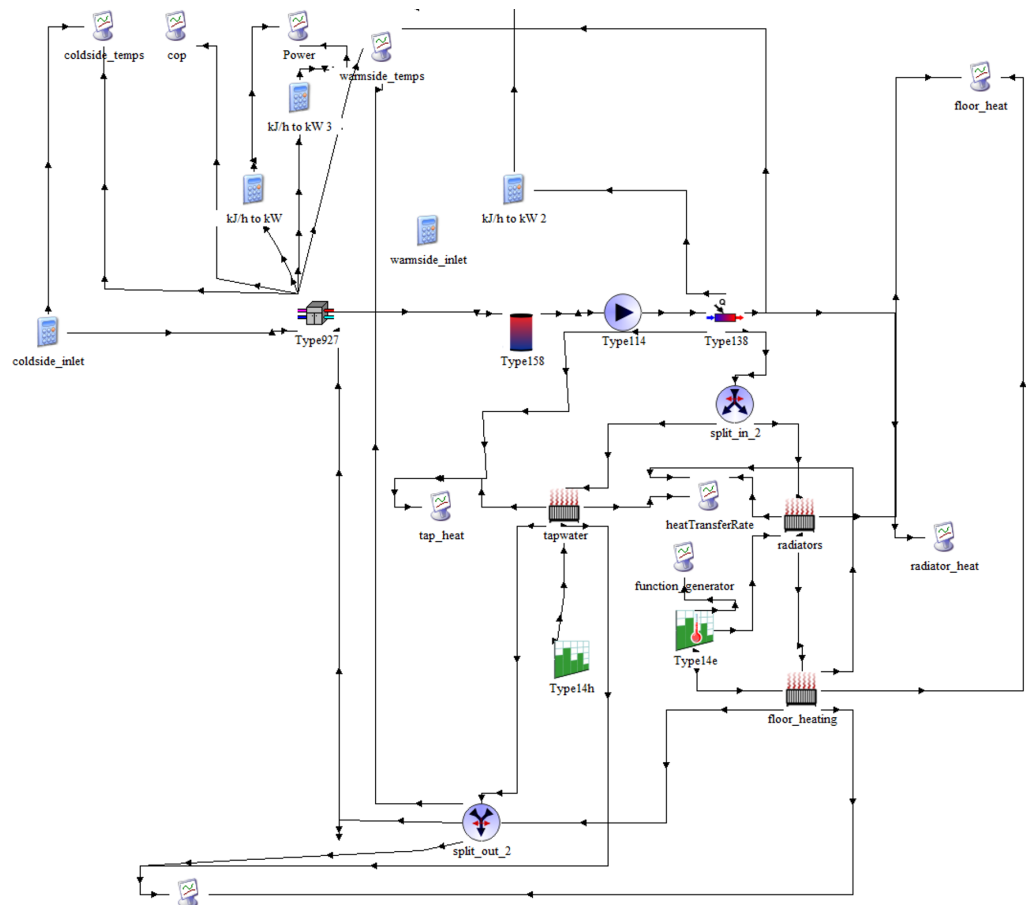


Figure 42: Series system

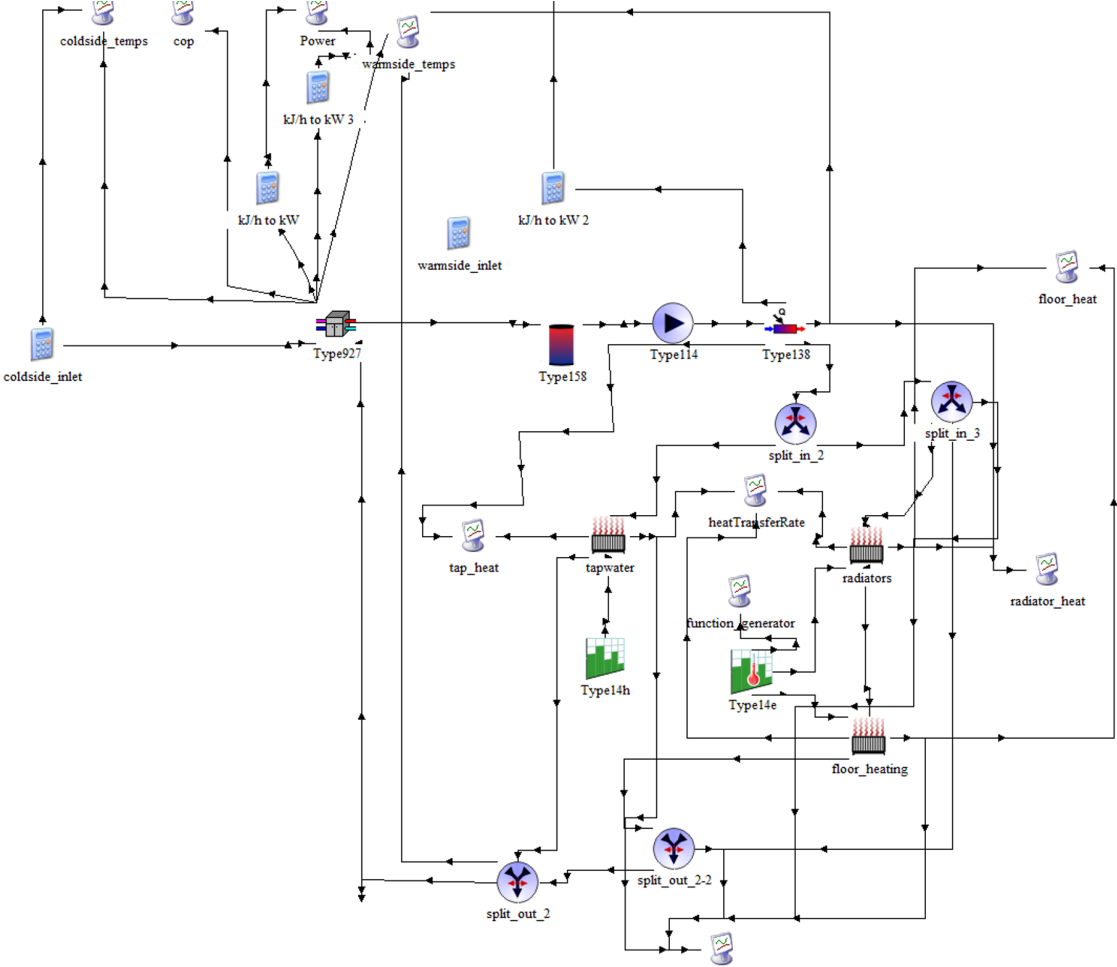


Figure 43: Series with bypass

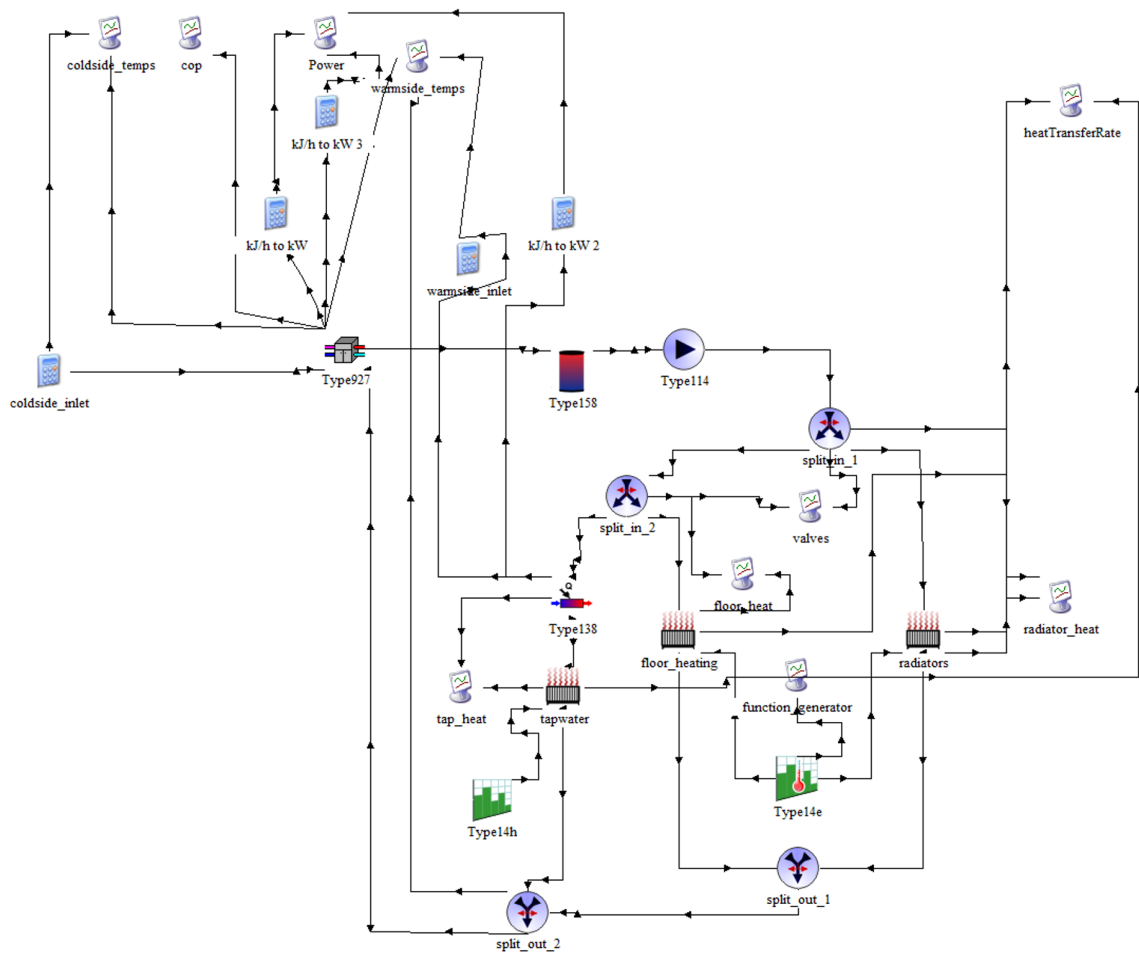


Figure 44: Parallel with heated tap water

Appendix C - Chapter 7 supplement - MATLAB showcase

The .m script can be downloaded from <https://github.com/boe28/boe28/blob/main/boe28.m>

State-Space

- The A Matrix (Figure 45) is the "state (or system) matrix" that describes the system's dynamic components. In our case this translates to a 10X10 matrix that contains so much that we would rather show it to you. All the elements of the matrices displayed are shown on the sides of them, so it is easy to identify the elements.

A =

	L_Overheated	t_kondensasj	L_2phaseRegi	t_condenserO	t_heatExchan
L_Overheated	1.668e+06	-6.753e+05	0	0	1.351e+06
t_kondensasj	-3.023e+11	1.223e+11	2.371e+07	0	-2.447e+11
L_2phaseRegi	0	2.291e+04	2.291e+04	0	0
t_condenserO	3.023e+11	-1.223e+11	-2.371e+07	-0.01825	2.447e+11
t_heatExchan	0	0.01337	0	0	-0.0144
t_heatExchan	0	0.02675	0	0	0
t_heatExchan	0	0.01337	0	0.01337	0
t_coolantMed	-0.003008	0	-0.003008	0	0
t_coolantMed	-0.003008	0	-1.047	0	0
t_coolantMed	-0.2392	0	1.047	0	0.005263

	t_heatExchan	t_heatExchan	t_coolantMed	t_coolantMed	t_coolantMed
L_Overheated	0	0	0	0	0
t_kondensasj	1.186e+07	0	0	0	0
L_2phaseRegi	-2.291e+04	0	0	0	0
t_condenserO	-1.186e+07	0.01711	0	0	0
t_heatExchan	0	0	0	0.02058	0.02058
t_heatExchan	0.0144	0	0.01337	0.01337	0
t_heatExchan	0	0.0679	0.02058	0	0
t_coolantMed	0	0.0006579	-0.00192	0	0
t_coolantMed	0.005263	-0.0006579	0.04731	-0.1296	0
t_coolantMed	-0.005263	0.0006579	-0.04731	0.1682	-0.04381

Figure 45: A-matrix

- The B Matrix (Figure 46) is the "input matrix" that shows all the inputs that affect the systems dynamics.

B =

	G_refrigeran	G_refrigeran	t_condenserO	t_coolantMed	G_coolantMed
L_Overheated	6.784e+05	0	-6.753e+05	0	0
t_kondensasj	1.197e+06	0	2.447e+11	0	0
L_2phaseRegi	0	-3084	0	0	0
t_condenserO	-1.197e+06	0.02526	-2.447e+11	0	0
t_heatExchan	0	0	0.01337	0	0
t_heatExchan	0	0	0	0	0
t_heatExchan	0	0	0	0.02058	0
t_coolantMed	0	0	0	0.001262	-0.002364
t_coolantMed	0	0	0	-0.001262	-0.1176
t_coolantMed	0	0	0	0.001262	0.05881

Figure 46: B-matrix

- The C Matrix (Figure 47) is the “output matrix” which shows what our selected outputs are.

```

C =
          L_Overheated  t_kondensasj  L_2phaseRegi  t_condenserO  t_heatExchan
t_kondensasj          0             1             0             0             0
t_condenserO          0             0             0             1             0
t_coolantMed          0             0             0             0             0

          t_heatExchan  t_heatExchan  t_coolantMed  t_coolantMed  t_coolantMed
t_kondensasj          0             0             0             0             0
t_condenserO          0             0             0             0             0
t_coolantMed          0             0             0             1             0

```

Figure 47: C-matrix

- The D Matrix (Figure 48) is the "feedthrough (or feedforward) matrix" (in cases where the system model does not have a direct feedthrough/ linear non-affected components, the D matrix is the zero matrix). This is where our model might have a design fault, though we are fairly certain that the system is correct. It is assumed that there are no linear components in this system, as described as in [36](Yao, Huang, and Chen, 2013), but we would have preferred a physical testing done to verify all the matrices. Especially the D matrix since there is an assumption of a zero matrix.

```

D =
          G_refrigeran  G_refrigeran  t_condenserO  t_coolantMed  G_coolantMed
t_kondensasj          0             0             0             0             0
t_condenserO          0             0             0             0             0
t_coolantMed          0             0             0             0             0

```

Figure 48: D-matrix

Transfer function

We have multiple input values, 5 state variables and 3 outputs, giving us a total of 15 transfer functions. Each describing a relation between a state variable and an output.

If you want to take a closer look at them all, we recommend opening the MATLAB file and following the script. It will show you all of them. Regardless, we will use and showcase an interesting transfer function. Namely the function that describes the relation of the compressor flow rate and the output in temperature of the work fluid. This is something we have read multiple thesis on, because of its ability to give continuous feedback control, instead of “on of” control with or without mutable % steps. But we have never encountered a thesis that has realized this control method. The information pulled by the transfer function will show why this is troublesome.

```

TF_massFlowRateCompressorControl =
From input "G_refrigerant_CompressorOutlet" to output "t_coolantMedium_OutletC":
-2041 s^8 + 2.497e14 s^7 + 4.913e21 s^6 - 1.496e20 s^5 - 4.072e19 s^4 + 1.939e18 s^3 + 1.164e16 s^2
- 1.029e15 s - 2.207e12
-----
s^10 - 1.224e11 s^9 + 2.782e15 s^8 + 1.36e18 s^7 + 1.345e17 s^6 - 1.675e16 s^5 - 9.259e14 s^4
+ 7.383e13 s^3 + 2.202e11 s^2 - 4.806e10 s + 2.529e08

```

Figure 49: Transfer function

Zeros and poles

As we can see from the transfer function's zero/pole representation form, there is a huge problem for this systems stability. It has multiple poles in the right (positive) field.

```
TF_massFlowRateCompressorControl_poleform =

From input "G_refrigerant_CompressorOutlet" to output "t_coolantMedium_OutletC":

2.4966e14 (s+1.968e07) (s+0.09509) (s-0.07067) (s+0.02087) (s+0.002113) (s^2 - 0.07786s + 0.001517)

-----

(s-1.224e11) (s-2.322e04) (s+478.5) (s+0.1304) (s+0.09329) (s-0.07023) (s-0.04505)

(s-0.03116) (s+0.02735) (s-0.005668)

Continuous-time zero/pole/gain model.
```

Figure 50: TF zero/pole representation form. The values of "s" in the brackets in the numerator equal the zero points. The "s" values that gives zeroing in the denominator are the poles of the system.

Observability

As seen in figure 51, there is a significant amount of data presented in MATLAB, therefore we suggest to explore it in the MATLAB file. Our finding is that the system is observable, reachable and controllable even though it is not stable.

```
observability =

0.0000e+000 1.0000e+000 0.0000e+000 0.0000e+000 0.0000e+000 0.0000e+000 0.0000e+000 0.0000e+000 0.0000e+000 0.0000e+000 0.0000e+000
0.0000e+000 0.0000e+000 0.0000e+000 1.0000e+000 0.0000e+000 0.0000e+000 0.0000e+000 0.0000e+000 0.0000e+000 0.0000e+000 0.0000e+000
0.0000e+000 0.0000e+000 0.0000e+000 0.0000e+000 0.0000e+000 0.0000e+000 0.0000e+000 0.0000e+000 0.0000e+000 1.0000e+000 0.0000e+000
-302.3053e+009 122.3498e+009 23.7146e+006 0.0000e+000 -244.7233e+009 11.8573e+006 0.0000e+000 0.0000e+000 0.0000e+000 0.0000e+000 0.0000e+000
302.3053e+009 -122.3498e+009 -23.7146e+006 -18.2536e-003 244.7233e+009 -11.8573e+006 17.1053e-003 0.0000e+000 0.0000e+000 0.0000e+000 0.0000e+000
-3.0083e-003 0.0000e+000 -1.0470e+000 0.0000e+000 0.0000e+000 5.2632e-003 -657.8947e-006 47.3146e-003 -129.6053e-003 0.0000e+000 0.0000e+000
-36.9875e+021 14.9697e+021 2.9015e+018 0.0000e+000 -29.9423e+021 1.4507e+018 0.0000e+000 158.5851e+003 -5.0353e+009 -5.0355e+009
36.9875e+021 -14.9697e+021 -2.9015e+018 561.9676e-006 29.9423e+021 -1.4507e+018 849.2361e-006 -158.5851e+003 5.0353e+009 5.0355e+009
-5.0191e+003 -21.9510e+003 -23.9824e+003 -8.7990e-006 -4.0631e+003 23.9825e+003 71.7228e-006 -6.1662e-003 16.8679e-003 0.0000e+000
-4.5255e+033 1.8316e+033 354.9996e+027 0.0000e+000 -3.6635e+033 177.4997e+027 0.0000e+000 19.4028e+015 -616.0764e+018 -616.0958e+018
4.5255e+033 -1.8316e+033 -354.9996e+027 1.1002e-006 3.6635e+033 -177.4997e+027 67.5083e-006 -19.4028e+015 616.0764e+018 616.0958e+018
6.6359e+015 -2.6857e+015 -521.1077e+009 1.1199e-006 5.3719e+015 -259.7298e+009 -18.4345e-006 328.7547e+000 237.1494e+000 -83.6023e+000
-553.6983e+042 224.0943e+042 43.4347e+039 0.0000e+000 -448.2320e+042 21.7173e+039 272.8320e+003 2.3740e+027 -75.3778e+030 -75.3802e+030
553.6983e+042 -224.0943e+042 -43.4347e+039 882.8069e-009 448.2320e+042 -21.7173e+039 -272.8320e+003 -2.3740e+027 75.3778e+030 75.3802e+030
811.9111e+024 -328.5988e+024 -63.6902e+021 -159.9973e-009 657.2613e+024 -31.8451e+021 1.2827e-006 -3.4738e+009 118.5297e+012 118.5332e+012
-67.7458e+054 27.4183e+054 5.3143e+051 3.6490e+003 -54.8418e+054 2.6571e+051 33.3812e+015 298.4583e+036 -9.2226e+042 -9.2229e+042
67.7458e+054 -27.4183e+054 -5.3143e+051 -3.6490e+003 54.8418e+054 -2.6571e+051 -33.3812e+015 -298.4583e+036 9.2226e+042 9.2229e+042
99.3385e+036 -40.2045e+036 -7.7926e+033 20.0765e-009 80.4169e+036 -3.8963e+033 41.4429e-003 -425.9113e+018 13.5235e+024 13.5239e+024
-8.2888e+066 3.3547e+066 650.2125e+060 446.4560e+012 -6.7100e+066 325.1061e+060 4.0838e+027 35.5380e+048 -1.1284e+054 -1.1284e+054
8.2888e+066 -3.3547e+066 -650.2125e+060 -446.4560e+012 6.7100e+066 -325.1061e+060 -4.0838e+027 -35.5380e+048 1.1284e+054 1.1284e+054
12.1542e+048 -4.9191e+048 -953.4339e+042 554.2767e-006 9.8391e+048 -476.7167e+042 -5.9874e+009 -52.1109e+030 1.6546e+036 1.6547e+036
-1.0141e+078 410.4481e+075 79.5545e+072 54.6342e+024 -820.9757e+075 39.7772e+072 499.7897e+036 4.3481e+060 -138.0611e+063 -138.0654e+063
1.0141e+078 -410.4481e+075 -79.5545e+072 -54.6342e+024 820.9757e+075 -39.7772e+072 -499.7897e+036 -4.3481e+060 138.0611e+063 138.0654e+063
1.4871e+060 -601.8573e+057 -116.6540e+054 -80.0780e+006 1.2038e+060 -58.3270e+054 -732.6816e+018 -6.3758e+042 202.4447e+045 202.4511e+045
-124.0823e+087 50.2189e+087 9.7336e+084 6.6844e+036 -100.4476e+087 4.8668e+084 61.1547e+048 531.9997e+069 -16.8920e+075 -16.8925e+075
124.0823e+087 -50.2189e+087 -9.7336e+084 -6.6844e+036 100.4476e+087 -4.8668e+084 -61.1547e+048 -531.9997e+069 16.8920e+075 16.8925e+075
181.9471e+069 -73.6381e+069 -14.2728e+066 -9.7992e+018 147.2905e+069 -7.1364e+066 -89.6467e+030 -780.0934e+051 24.7694e+057 24.7702e+057
-15.1817e+099 6.1444e+099 1.1909e+096 817.9128e+045 -12.2899e+099 595.4600e+093 7.4807e+060 65.0909e+081 -2.0668e+087 -2.0668e+087
15.1817e+099 -6.1444e+099 -1.1909e+096 -817.9128e+045 12.2899e+099 -595.4600e+093 -7.4807e+060 -65.0909e+081 2.0668e+087 2.0668e+087
22.2615e+081 -9.0097e+081 -1.7463e+078 -1.1990e+030 18.0212e+081 -873.1480e+075 -10.9707e+042 -95.4456e+063 3.0306e+069 3.0307e+069

rank =

1.0000e+000
```

Figure 51: The observability matrix

We found reachability and contrallability using the methods presented in the two next sections.

Reachability

The reachability problem is to find the set of all the final states $x(t_1)$ reachable starting from a given initial state $x(t_0)$:

- A state $x(t_1)$ of a dynamic system is reachable from the state $x(t_0)$ in the time interval $[t_0, t_1]$ if it exists an input function $u(\cdot) \in \mathcal{U}$ such that $x(t_1) = \psi(t_0, t_1, x(t_0), u(\cdot))$.
- Let $X^+(t_0, t_1, x(t_0))$ denote the set of all the final states $x(t_1)$ reachable at time t_1 starting from the initial state $x(t_0)$.

$$x(k) = \sum_{j=0}^{k-1} A^{k-j-1} B u(j) = [B \quad AB \quad \dots \quad A^{k-1}B] \begin{bmatrix} u_{k-1} \\ u_{k-2} \\ \vdots \\ u_0 \end{bmatrix}$$

Controllability

A system is said to be controllable if it is possible by means of an unconstrained control signal to transfer the system from any initial state $x(t_0)$ to any other state $x(t_1)$ in finite interval of time. The controllability problem is to find the set of all the initial states $x(t_0)$ controllable to a given final state $x(t_1)$:

- A state $x(t_0)$ of a dynamic system is controllable to state $x(t_1)$ in the time interval $[t_0, t_1]$ if it exists an input function $u(\cdot) \in \mathcal{U}$ such that $x(t_1) = \psi(t_0, t_1, x(t_0), u(\cdot))$.
- Let $X^-(t_0, t_1, x(t_1))$ the set of all the initial states $x(t_0)$ controllable to the final state $x(t_1)$ at time t_1 .

$$0 = x(k) = A^k x_0 + \sum_{j=0}^{k-1} A^{k-j-1} B u(j)$$

Determinants, eigenvalues and trace

The controllability matrix is a 10X50 matrix, which means that it is not an equal sided rectangle/ square, and the determinant is not calculable. The systems eigenvalues from the A matrix are on the other hand easily calculated, but very unsatisfactory as we have already seen on the poles (eigenvalue of A matrix equals poles). The positive poles represent the instability of the system and need to be addressed accordingly to be eliminated or compensated for, for this system to work as intended. To compensate for this instability, one might change the design of the system or change the input values, but we propose to focus on the regulation side of this to achieve better results.

By calculating the sum of the diagonal line of the A matrix, we can calculate the trace of the system, which again will describe stability of the system and whether it converges or diverges. It preferably should be negative and diverge to a value. This confirms again that drastic changes need to be done. And these criteria need to be fulfilled, for a system to be used.

```
System_eig =
    122.3515e+009
     23.2185e+003
   -478.5450e+000
  -130.4486e-003
   -93.2872e-003
   -27.3547e-003
    70.2329e-003
     5.6685e-003
    31.1611e-003
    45.0526e-003
```

Figure 52: Eigenvalues

```
Stability_from_trace =
    122.3515e+009
```

Figure 53: Trace of the system

Appendix D - Matlab code

The file can be accessed here: <https://github.com/boe28/boe28/blob/main/boe28.m>

```

1  %% Condenser - Heat pump to Warm side system - Propane (R290)
2  %%Finding the dynamic system that describe the energy transfer.
3  clc; clear all; close all; format shortEng;
4
5  display('Welcome to an interactive presentation of our system!')
6  display(' ')
7  display('This program is supplemental to our report: Control and system engineering to')
8  display('optimise energy transfer from natural refrigerant heat pump systems.')
9  display('Please see chapter 7 for background and additional information.')
10 display(' ')
11 display('The system presented here is based upon a mathematical representation taken from:')
12 display('State-space model for dynamic behavior of vapor compression liquid chiller')
13 display('https://www.sciencedirect.com/science/article/abs/pii/S0140700713001187#fig2')
14
15 x = input('Press "Enter" to continue!');
16 clc;
17 %% variable
18 %
19 display('There are a lot of variables in this model! Everything from the diameter of the pipes, area of the')
20 display('condenser walls, temperature of the substances in different states, for all the inputs and outputs.')
21 display('There is also enthalpy, mass, specific heat at a specific pressure, void fractions, and mass flowrate.')
22 display('Some are easy to obtain from heat pump datasheets, however some variables like')
23 display('inner physical dimensions are not public, as these often are company secrets.')
24 display('We have proposed the closest estimates to our ability.')
25 display('The variables can be modified according to your needs. New variables will not 'ruin' the system,')
26 display('they will only change the outcome. However, if one of them has the same value (or very close value)')
27 display('as an another variable, there is a risk that it might cancel out one or more of the equations. ')
28 display('Some mathematical knowledge is required.')
29 display(' ')
30 x = input('Press "Enter" to continue!');
31 clc;
32 %diameter of pipe [m]
33 d_evaporator = 0.00137; % Y Table 2
34 d = 0.059; % Y Table 2
35 %density [kg m^3]
36 p2 = 2; % N (2 = compressor outlet/condenser inlet)
37 p_refrigerantVaporC = 1.5; % N
38 p_liquid = 550; % N hva slags "liquid"?
39 p_refrigerantLiquid = 400; % Y
40 p_heatExchangerWallC = 2700; % N
41 p_coolantMedium = 1000; % N hva er forskjellen fra liquid?
42 %enthalpy [1/(j kg)]
43 h1 = 450; % Y ? https://www.researchgate.net/figure/Ideal-vapour-compression-refrigeration-cycle-of-propane-on-the-p-h-
44 diagram_fig3_335869667
45 h2 = 650; % Y ?
46 h_refrigerantVaporC = 95; % N
47 h_latentC = 70; % N
48 %convective heat transfer coefficient [W/(m^2 C)]
49 alfa_refrigerant = 650;
50 alfa_coolantMedium = 1000;
51 %area of heat exchanger [m^2]
52 Ac = 8; % N
53 %length of flow
54 Lc = 4; % N
55 L_OverheatedRegionC = 0.85; % N
56 L_twoPhaseRegionC = 0.4; % N
57 %temperature [C]
58 t_condensation = 87.4; % N
59 t_heatExchangerWall_OverheatedRegion = 91.0; % N
60 t_heatExchangerWall_twoPhaseRegionC = 88.2; % N
61 t_heatExchangerWall_OverCoolingRegionC = 1; % N
62 t_compressorOutlet = 92.5; % N
63 t_condenserOutlet = 83.0; % N
64 t_coolantMedium_InletC = 45.0; % N
65 t_coolantMedium_twoPhaseRegionC = 50.2; % N
66 t_coolantMedium_overheatedRegionC = 55; % N
67 t_coolantMedium_OutletC = 60.0; % N
68 %d/dt
69 dp_dt = 1; %p_refrigerantVaporC derivert med tanke paa t_k
70 % void fraction - 0 til 1 % andel veske til gass"
71 voidFraction = 0.6; % N
72 % specific heat at constant pressure [J / kg C]
73 c_p_Liquid = 1900;
74 c_p_condenserOutlet = 1200;
75 c_p_heatExchangerWallC = 900;
76 c_p_coolantMedium = 3800;
77 % volume [m^3]
78 V_refrigerantC = 0.8; % N
79 V_heatExchangerWallC = 0.08; % N
80 V_coolantMediumC = 0.8; % N
81 % mass flow rate [kg s^-1]
82 G_refrigerant_condenserOutlet = 0.2;
83 G_refrigerant_CompressorOutlet = 0.2;
84 G_coolantMediumC = 3.5;
85
86 %% Inputs and outputs
87 %
88 display('We have a MIMO system, with 10 inputs and 3 outputs')
89 display(' ')
90 display('This means that we have multiple Transfer functions,')
91 display('and aspects of interest in variation of all these inputs.')
```

```

108 X15 = -alfa_refrigerant*L_OverheatedRegionC*(Ac/(2.0*Lc))
109 T2 = (1.0/8.0)*pi*(d_evaporator^2.0)*L_OverheatedRegionC*(dp_dt)
110 X21 = -alfa_refrigerant*(Ac/Lc)*(t_heatExchangerWall_OverheatedRegion - ((t_condensation + t_compressorOutlet) / 2.0))*(p2-
111 p_refrigerantVaporC)/p2*(h2-h_refrigerantVaporC))
112 X22 = alfa_refrigerant*L_OverheatedRegionC*(Ac/(2.0*Lc))*(p2-p_refrigerantVaporC)/p2*(h2-h_refrigerantVaporC) - alfa_refrigerant*(Ac/Lc)*(
113 L_2phaseRegionC/h_latentC)
114 X23 = - alfa_refrigerant*(Ac/Lc)*(1/h_latentC)*(t_condensation - t_heatExchangerWall_twoPhaseRegionC)
115 X24 = -alfa_refrigerant*L_OverheatedRegionC*(Ac/Lc)*(p2-p_refrigerantVaporC)/p2*(h2-h_refrigerantVaporC)
116 X25 = alfa_refrigerant*(Ac/Lc)*(L_2phaseRegionC/h_latentC)
117 X26 = p_refrigerantVaporC/p2
118 X27 = alfa_refrigerant*L_OverheatedRegionC*(Ac/Lc)*(p2-p_refrigerantVaporC)/p2*(h2-h_refrigerantVaporC)
119 T3 = (pi/4)*(d_evaporator^2.0)*p_liquid*h_latentC*(1-voidFraction)
120 X31 = alfa_refrigerant*(Ac/Lc)*(L_2phaseRegionC)
121 X32 = alfa_refrigerant*(Ac/Lc)*(t_condensation - t_heatExchangerWall_twoPhaseRegionC)
122 X33 = -alfa_refrigerant*(Ac/Lc)*(L_2phaseRegionC)
123 X34 = -h_latentC;
124 T4 = p_refrigerantLiquid*c_p_liquid*V_refrigerantC*((Lc-L_OverheatedRegionC-L_2phaseRegionC)/(2.0*Lc))
125 X41 = ((G_refrigerant_condenserOutlet*c_p_condenserOutlet)/(Lc-L_OverheatedRegionC-L_2phaseRegionC))*(t_condensation - t_condenserOutlet)-((T4
126 /T2)*X21)
127 X42 = (G_refrigerant_condenserOutlet*c_p_condenserOutlet)-alfa_refrigerant*(Ac/(2.0*Lc))*(Lc-L_OverheatedRegionC-L_2phaseRegionC)-((T4/T2)*X22
128 /T2)*X23)
129 X43 = ((G_refrigerant_condenserOutlet*c_p_condenserOutlet)/(Lc-L_OverheatedRegionC-L_2phaseRegionC))*(t_condensation - t_condenserOutlet)-((T4
130 /T2)*X23)
131 X44 = -(G_refrigerant_condenserOutlet*c_p_condenserOutlet+alfa_refrigerant*(Ac/Lc)*(Lc-L_OverheatedRegionC-L_2phaseRegionC))
132 X45 = -(T4/T2)*X24
133 X46 = -(T4/T2)*X25
134 X47 = alfa_refrigerant*(Ac/Lc)*(Lc-L_OverheatedRegionC-L_2phaseRegionC)
135 X48 = -(T4/T2)*X26
136 X49 = (t_condensation - t_condenserOutlet)*c_p_condenserOutlet
137 X40 = -(T4/T2)*X27
138 T5 = V_heatExchangerWallC*p_heatExchangerWallC*c_p_heatExchangerWallC*(L_OverheatedRegionC/Lc)
139 X51 = alfa_refrigerant*(Ac*L_OverheatedRegionC/(2.0*Lc))
140 X52 = (alfa_refrigerant*(Ac*L_OverheatedRegionC/Lc)-alfa_coolantMedium*(Ac*L_OverheatedRegionC/Lc))
141 X53 = alfa_coolantMedium*(Ac*L_OverheatedRegionC/(2.0*Lc))
142 X54 = alfa_coolantMedium*(Ac*L_OverheatedRegionC/(2.0*Lc))
143 X55 = alfa_refrigerant*(Ac*L_OverheatedRegionC/(2.0*Lc))
144 T6 = V_heatExchangerWallC*p_heatExchangerWallC*c_p_heatExchangerWallC*L_2phaseRegionC/Lc
145 X61 = alfa_refrigerant*L_2phaseRegionC*(Ac/Lc)
146 X62 = -(alfa_refrigerant*(Ac*L_2phaseRegionC/Lc)-alfa_coolantMedium*(Ac*L_2phaseRegionC/Lc))
147 X63 = alfa_refrigerant*L_2phaseRegionC*(Ac/(2.0*Lc))
148 X64 = alfa_refrigerant*L_2phaseRegionC*(Ac/(2.0*Lc))
149 T7 = V_heatExchangerWallC*p_heatExchangerWallC*c_p_heatExchangerWallC*(Lc-L_OverheatedRegionC-L_2phaseRegionC)/Lc
150 X71 = alfa_refrigerant*(Ac/(2.0*Lc))*(Lc-L_OverheatedRegionC-L_2phaseRegionC)
151 X72 = alfa_refrigerant*(Ac/(2.0*Lc))*(Lc-L_OverheatedRegionC-L_2phaseRegionC)
152 X73 = (alfa_refrigerant*(Ac/(Lc))*(Lc-L_OverheatedRegionC-L_2phaseRegionC)+alfa_coolantMedium*(Ac/(Lc))*(Lc-L_OverheatedRegionC-
153 L_2phaseRegionC))
154 X74 = alfa_coolantMedium*(Ac/(2.0*Lc))*(Lc-L_OverheatedRegionC-L_2phaseRegionC)
155 X75 = alfa_coolantMedium*(Ac/(2.0*Lc))*(Lc-L_OverheatedRegionC-L_2phaseRegionC)
156 T8 = V_coolantMediumC*p_coolantMedium*c_p_coolantMedium*(Lc-L_OverheatedRegionC-L_2phaseRegionC)
157 X81 = G_coolantMediumC*c_p_coolantMedium*(t_coolantMedium_InletC-t_coolantMedium_twoPhaseRegionC)/(Lc-L_OverheatedRegionC-L_2phaseRegionC)
158 X82 = G_coolantMediumC*c_p_coolantMedium*(t_coolantMedium_InletC-t_coolantMedium_twoPhaseRegionC)/(Lc-L_OverheatedRegionC-L_2phaseRegionC)
159 X83 = alfa_coolantMedium*(Ac/Lc)*(Lc-L_OverheatedRegionC-L_2phaseRegionC)
160 X84 = -(G_coolantMediumC*c_p_coolantMedium+alfa_coolantMedium*(Ac/(2.0*Lc))*(Lc-L_OverheatedRegionC-L_2phaseRegionC))
161 X85 = G_coolantMediumC*c_p_coolantMedium-alfa_coolantMedium*(Ac/(2.0*Lc))*(Lc-L_OverheatedRegionC-L_2phaseRegionC)
162 X86 = c_p_coolantMedium*(t_coolantMedium_InletC-t_coolantMedium_twoPhaseRegionC)
163 T9 = V_coolantMediumC*p_coolantMedium*c_p_coolantMedium*L_2phaseRegionC/(2.0*Lc)
164 X91 = -(T9/T8)*X81
165 X92 = (G_coolantMediumC*c_p_coolantMedium/L_2phaseRegionC)*(t_coolantMedium_twoPhaseRegionC-t_coolantMedium_overheatedRegionC) - (T9/T8)*X82
166 X93 = alfa_coolantMedium*(Ac/Lc)*L_2phaseRegionC
167 X94 = -(T9/T8)*X83
168 X95 = (G_coolantMediumC*c_p_coolantMedium-alfa_coolantMedium*(Ac/(2.0*Lc))*L_2phaseRegionC) - (T9/T8)*X84
169 X96 = -(G_coolantMediumC*c_p_coolantMedium+alfa_coolantMedium*(Ac/(2.0*Lc))*L_2phaseRegionC)
170 X97 = -(T9/T8)*X85
171 X98 = c_p_coolantMedium*(t_coolantMedium_twoPhaseRegionC-t_coolantMedium_overheatedRegionC)-(T9/T8)*X86
172 T0 = V_coolantMediumC*p_coolantMedium*c_p_coolantMedium*L_OverheatedRegionC/(2.0*Lc)
173 X01 = G_coolantMedium*(t_coolantMedium/L_OverheatedRegionC)*(t_coolantMedium_overheatedRegionC-t_coolantMedium_OutletC)-(T0/T9)*X91
174 X02 = -(T0/T9)*X92
175 X03 = alfa_coolantMedium*(Ac/Lc)*L_OverheatedRegionC
176 X04 = -(T0/T9)*X93
177 X05 = -(T0/T9)*X94
178 X06 = -(T0/T9)*X95
179 X07 = c_p_coolantMedium*G_coolantMediumC - alfa_coolantMedium*(Ac/(2.0*Lc))*L_OverheatedRegionC -(T0/T9)*X96
180 X08 = -c_p_coolantMedium*G_coolantMediumC - alfa_coolantMedium*(Ac/(2.0*Lc))*L_OverheatedRegionC
181 X09 = -(T0/T9)*X97
182 X00 = c_p_coolantMedium*(t_coolantMedium_overheatedRegionC-t_coolantMedium_OutletC)-(T0/T9)*X98
183
184 x = input('Press "Enter" to continue!');
185 clc;
186 %% A-matrix
187 %
188 display('The A matrix, often known as the transition matrix.')
189 display('It explains the dynamic properties of the system!')
190
191 A = [...
192 X11/T1,X12/T1,0,0,X13/T1,0,0,0,0,0; ...
193 X21/T2,X22/T2,X23/T2,0,X24/T2,X25/T2,0,0,0,0; ...
194 0,X31/T3,X31/T3,0,0,X33/T3,0,0,0,0; ...
195 X41/T4,X42/T4,X43/T4,X44/T4,X45/T4,X46/T4,X47/T4,0,0,0; ...
196 0,X51/T5,0,0,X52/T5,0,0,0,X53/T5,X54/T5; ...
197 0,X61/T6,0,0,0,X62/T6,0,0,X63/T6,X64/T6,0; ...
198 0,X71/T7,0,X72/T7,0,0,X73/T7,X74/T7,0,0,0; ...
199 X81/T8,0,X82/T8,0,0,0,X83/T8,X84/T8,0,0; ...
200 X91/T9,0,X92/T9,0,0,X93/T9,X94/T9,X95/T9,X96/T9,0; ...
201 X01/T0,0,X02/T0,0,X03/T0,X04/T0,X05/T0,X06/T0,X07/T0,X08/T0]
202
203 x = input('Press "Enter" to continue!');
204 clc;
205 %% B-Matrix
206 %
207 display('The B matrix, often known as the routing matrix/ input matrix.')
208 display('It explains the "routing" from input values to the state variables!')
209
210 B = [...
211 X14/T1,0,X15/T1,0,0; ...
212 X26/T2,0,X27/T2,0,0; ...
213 0,X34/T3,0,0,0; ...
214 X48/T4,X49/T4,X40/T4,0,0; ...
215 0,0,X55/T5,0,0; ...
216 0,0,0,0,0; ...
217 0,0,0,X75/T7,0; ...
218 0,0,0,X85/T8,X86/T8; ...
219 0,0,0,X97/T9,X98/T9; ...
220 0,0,0,X09/T0,X00/T0]

```

```

218
219 x = input('Press "Enter" to continue!');
220 clc;
221 %% C-Matrix
222 %
223 display('C matrix - "output matrix."')
224 display('The output linear-combination matrix.')
```

$$C = \begin{bmatrix} \dots & \dots & \dots & \dots & \dots \\ 0,1,0,0,0,0,0,0,0,0; \dots & \dots & \dots & \dots & \dots & \dots & \dots & \dots & \dots & \dots \\ 0,0,0,1,0,0,0,0,0,0; \dots & \dots & \dots & \dots & \dots & \dots & \dots & \dots & \dots & \dots \\ 0,0,0,0,0,0,0,0,1,0 \end{bmatrix}$$

```

229
230 %% D-Matrix
231 %
232 display('D matrix.')
```

$$D = \begin{bmatrix} \dots & \dots & \dots & \dots & \dots \\ 0,0,0,0,0; \dots & \dots & \dots & \dots & \dots \\ 0,0,0,0,0; \dots & \dots & \dots & \dots & \dots \\ 0,0,0,0,0 \end{bmatrix}$$

```

238
239 %D_1 = eye(3,5)
240
241 x = input('Press "Enter" to continue!');
242 clc;
243 %% system eq.
244 %
245 display('State system is based upon A, B, C and D matrix. Since this a large MIMO system')
246 display('with 5 state variables/"control points" and 3 outputs, it will be converted into 15 transfer functions.')
```

$$\text{display('For an easier overview, input and output variables will be displayed on all the matrices.')} \\ \text{display('Afterwards we will display all the 15 transfer functions.')} \\ \text{display('')}$$

```

249
250 x = input('Press "Enter" to continue!');
251 clc;
252
253 delta_X = ['L_OverheatedRegionC' 't_condensation' 'L_2phaseRegionC' 't_condenserOutlet' 't_heatExchangerWall_OverheatedRegion' '
t_heatExchangerWall_twoPhaseRegionC' 't_heatExchangerWall_OverCoolingRegionC' 't_coolantMedium_twoPhaseRegionC' '
t_coolantMedium_overheatedRegionC' 't_coolantMedium_OutletC'];
254 delta_U = ['G_refrigerant_CompressorOutlet' 'G_refrigerant_condenserOutlet' 't_condenserOutlet' 't_coolantMedium_InletC' 'G_coolantMediumC'];
255 delta_Y = ['t_condensation' 't_condenserOutlet' 't_coolantMedium_OutletC'];
256
257 Sys_ss = ss(A,B,C,D,'statename',delta_X,'inputname',delta_U,'outputname',delta_Y)
258
259 x = input('Press "Enter" to continue!');
260 clc;
261
262 Sys_tf = tf(Sys_ss)
263
264 display(' ')
265
266 A_a = deltaX;
267 [deltaX] = gradient(A);
268 B_b = deltaU;
269 [deltaU] = gradient(B);
270 C_c = deltaY;
271 [deltaY] = gradient(C);
272
273 %deltaXdots = deltaX*A_a + deltaU*B_b
274 %deltaYdots = deltaX*C_c
275
276 x = input('Press "Enter" to continue!');
277 clc;
278
279 %% Transfer function's
280 %
281 display('We have 10 inputs, 5 state variables and 3 outputs for our system.')
```

$$\text{display('This means that we have 15 transfer functions for this system.')} \\ \text{display('For an example, here we have the transfer function for')} \\ \text{display('the output temperature from regulation of the mass flow')} \\ \text{display('rate by the the compressor.')} \\ \text{display('')}$$

```

287
288 x = input('Press "Enter" to continue!');
289 clc;
290
291 TF_massFlowRateCompressorControl = Sys_tf(3,1)
292
293 %[num,den] = ss2tf(A,B,C,D,3);
294 %abc_Output1 = tf(num(1,:),den)
295 %abc_Output2 = tf(num(2,:),den)
296 %abc_Output3 = tf(num(3,:),den)
297
298 prompt = 'Do you want like to se the graphs of this TF? Y/N [Y]: ';
299 x = input(prompt,'s');
```

$$\text{if strcmp(x,'y') | strcmp(x,'Y')} \\ \text{figure(1)} \\ \text{bode(TF_massFlowRateCompressorControl)} \\ \text{grid on}$$

```

304
305 pause
306 x = input('Press "Enter" to continue!');
307 clc; close all;
308
309 figure(1)
310 step(TF_massFlowRateCompressorControl)
311 grid on
312
313 pause
314 x = input('Press "Enter" to continue!');
315 clc; close all;
316
317 pole(TF_massFlowRateCompressorControl)
318 figure(1)
319 pzplot(TF_massFlowRateCompressorControl)
320 grid on
321
322 end
323 % prompt = 'Do you want like to se an exemple of regulation? Y/N [Y]: ';
324 % x = input(prompt,'s');
```

$$\text{% if strcmp(x,'y') | strcmp(x,'Y')} \\ \text{% } \\ \text{% G = 0;} \\ \text{% } \\ \text{% zero(G)} \\ \text{% } \\ \text{% }$$

```

330
331 %

```

```

332 %
333 %   poles = [-20-5*i,-20+5*i]
334 %   L = place(A',C',poles)
335 %   desired_FM = 50;
336 %   %bode
337 %   figure('Name', 'Bode of G(s)'); bode(G), grid on;
338 %   %Regulering
339 %   n = 10.0;
340 %   H_arg = 20.0;
341 %   %Oensket kryssfrekvens der vinkel Gp er:
342 %   G_phase_arg = -180.0 + desired_FM - H_arg
343 %   w_onsket = input('Skriv inn oensket kryssfrekvens: ');
344 %   Ti = 2.8 / w_onsket;
345 %   Td = 1.0 / w_onsket;
346 %   G_dB = input('Les av 20lg|Gp(w_onsket)| : ');
347 %   H_dB = 3.5;
348 %   kr_dB = - G_dB - H_dB;
349 %   kr = 10.0^(kr_dB/20);
350 %   H = tf([Td,1],[Ti,0]) * tf([Td,1],[Td/n],1);
351 %
352 %   kr
353 %   H
354 %   Gr = kr * H;
355 %   G_Regulert = Gr * Gp
356 %
357 %   figure('Name', 'Bode of regulert system')
358 %   bode(G)
359 %   grid on
360 %
361 %   M_TilbakekobletSys = feedback(G_Regulert,1)
362 %   figure('Name', 'Step-response of M(s)')
363 %   step(M_TilbakekobletSys)
364 %   grid on
365 %
366 %
367 %   figure(1)
368 %   bode(TF_massFlowRateCompressorControl)
369 %   grid on
370 %
371 %   pause
372 %   x = input('Press "Enter" to continue!');
373 %   clc; close all;
374 %
375 %   figure(1)
376 %   step(TF_massFlowRateCompressorControl)
377 %   grid on
378 %
379 %   pause
380 %   x = input('Press "Enter" to continue!');
381 %   clc; close all;
382 %
383 %   pole(TF_massFlowRateCompressorControl)
384 %   figure(1)
385 %   pzplot(TF_massFlowRateCompressorControl)
386 %   grid on
387 % end
388
389 x = input('Press "Enter" to continue!');
390 clc; close all;
391
392
393 %% Stability and reachability
394 %
395
396
397 display('It is important that all eigenvalues/ poles have negative real part.')
398 display('If at least one eigenvalue has a positive part,')
399 display('the equilibrium will be unstable.')
400 System_eig = eig(A);
401 System_det = det(A); %If stable, than everyting is reachable
402
403
404
405
406 display('The eigenvalue or characteristic polynomial is:')
407 System_eig
408
409 display(' ')
410 x = input('Press "Enter" to continue!');
411 clc;
412
413 display('For all negative real parts the system is stable,')
414 display('on the other hand, all the positives are unstable,')
415 display('and need further adjustments to change pole placements.')
416 display(' ')
417 display('This system has unfortunate pole placement that needs to')
418 display('be addressed for it to be stable.')
419 %display('The determinants of the systems inputs are:')
420 %System_det
421
422 %   display('The A matrix has a zero determinant and is "called singular or non-invertible"')
423 %   display('There exists a vector v /= 0 such that Av = 0')
424 %   display('This means that NOT all components/ inputs are reachable')
425 %   display('We also know that this system is not stable!')
426
427 display(' ')
428
429 x = input('Press "Enter" to continue!');
430 clc;
431
432 Stability_from_trace = trace(A)
433 display('By calculating the sum of the diagonal line of the A matrix')
434 display('we can calculate the trace of the system,')
435 display('which again will describe stability of the system - ')
436 display('whether it converges or diverges. It preferably should be negative.')
437
438 x = input('Press "Enter" to continue!');
439 clc;
440
441 %% Observability
442 %
443 display('Observability depends on the A and C matrix, and not B!')
444 display('The A matrix has the dynamic components of the system,')
445 display('The C matrix is linear-combination of the output to the state variables!')
446 display('The A is a 10X10 matrix and C is a 3X10 matrix')
447 display('The matrix we will get is a 30X10 matrix of')

```

```
448 display('values that will determine observability!')
449 display(' ')
450
451
452
453 prompt = 'Do you want to see the observability matrix? Y/N [Y]: ';
454 x = input(prompt, 's');
455 if strcmp(x, 'y') | strcmp(x, 'Y')
456     clc;
457     observability = obsv(A,C)
458     rank = rank(observability)
459 end
460
461 x = input('Press "Enter" to continue!');
462 clc;
463
464 %% Controllability
465 %
466 display('Controllability depends on the A and B matrix, and not C!')
467 display('The A matrix has the dynamic components of the system.')
468 display('The B matrix is routing input values to the state variables!')
469 display('The A is a 10X10 matrix and B is a 10X5 matrix')
470 display('We will therefore get 10X50 values that will determine controllability!')
471 display(' ')
472 prompt = 'Do you want to see the controllability matrix? Y/N [Y]: ';
473 x = input(prompt, 's');
474 if strcmp(x, 'y') | strcmp(x, 'Y')
475     controllability = ctrb(A,B)
476     %Rank_of_con = rank(controllability)
477     %Determinant_of_cont = det(controllability)
478
479     display('Using the determinant to determine controllability only works if you')
480     display('only have only one input, i.e. the input matrix B has only one column.')
481     display('Namely, for more inputs the controllability matrix C is not square.')
482     display('The matrix CCT would always be square and if its determinant is non-zero')
483     display('when the controllability matrix has full rank. This is similar to checking')
484     display('whether the controllability matrix has "n" non-zero singular values.')
485     display('with "n" the size of the system state.')
486     display('So this has to be taken in consideration here!')
487 end
488
489 x = input('Press "Enter" to continue!');
490 clc;
491
492 %%
493
494 %% Thank you for your attention!
495 %
496 display('Thank you for your attention!')
497 display(' ')
498 x = input('The program is done. Press "Enter" to finish!');
499 clc; clear all; close all;
```