Rankine Cycles, Modeling and Control

Thesis by

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Thesis information

This is a Master Thesis done at the Department of Energy Sciences, Lund University, Faculty of Engineering, and Modelon. Supervisors were Marcus Thern at the university and Pieter Dermont at Modelon. The examiner from the Univeristy was Magnus Genrup and opponent Filip Olsson. The project was formulated by and carried out at Modelon AB. The thesis was presented on the 18th of December 2015.

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Abstract

As the demand for decreased use of energy increases, new ways of using energy sources are investigated. One of these ways is to be able to use low temperature heat sources, which can be done with a rankine or organic rankine cycle. In a rankine cycle there is a working fluid which is pressurized, evaporated, expanded and then condensed again, and the energy release during the expansion can drive a generator which generates electricity.

The aims of this master thesis were to model a rankine cycle, parameterise it according to data received from a company, implement control strategies and gain knowledge about its behaviour. The external company provided 11 data sets from their rankine cycle test bench, and they were not all coherent; some had a mass flow that was too low, it required the turbine to have a mechanical efficiency that was above 100 %.

A new model of a pump was built, but otherwise models that already existed were used. The components were then parameterised to the different data sets and the simulations gave results that matched the data. For the data sets that were coherent, the simulations gave results that matched the data better. A power point tracking diagram done by the external company was also recreated. Control strategies that controlled the super heating, torque of turbine and working fluid charge in cycle were implemented. A tank before the pump was also added in order to prevent vapour to enter the pump. These control strategies as well as simulation results have created a base for knowledge about rankine cycles as well as organic rankine cycles. The results can be used by Modelon as a base for further investigation as well as for other projects.

Contents

Chapter 1

Nomenclature

RC = Rankine Cycle

ORC = Organic Rankine Cycle

WF = working fluid

SH = super heating/super heated

Δh = difference in enthalpy/specific enthalpy $[J]/[J/kg]$

ΔT = difference in temperature $[K]$

η = efficiency $[\%]$

ρ = density $[kg/m^3]$

Cp = specific heat capacity $[J/K]$

h = enthalpy $[J]$

h_{sp} = specific enthalpy $[J/kg]$

K = proportional coefficient

$\dot{m}$ = mass flow $[kg/s]$

P = power $[W][J/s]$

p = pressure $[Pa]$

s = entropy $[J/Kmol]$

T = temperature $[K]$

T_i = integral coefficient

t = time

$v_{specific}$ = specific volume $[m^3/kg]$

v = volume $[m^3]$

W = work $[J]$

Chapter 2

Introduction

The use of energy has increased enormously in the last centuries as technology has developed and become central to our daily lives. Due to use of fossil fuel emissions a global warming is occurring which can affect ecosystems and cause imminent climate changes [1]. Strategies to avoid changes that could jeopardize existing societies have been designed, and one of them is to utilize waste heat and unconventional heat sources. A common issue with these sources is that they usually are of low temperatures compared to conventional heat sources (such as oil or biomass), and are therefore hard to utilize with conventional methods.

2.1 Rankine Cycle

The method used in most power plants to extract power from a heat source is by the principle of a Rankine Cycle (RC). A Rankine Cycle models the process where a medium goes through different phases in order to convert heat into mechanical work. The medium used is often water, and the heat source can be anything from oil to biomass. By choosing a medium that can be evaporated at a convenient pressure for lower temperatures, it is possible to use heat sources of such temperatures. If the medium is organic, the temperature used can be very low, and this kind of Rankine Cycle is called Organic Rankine Cycle (ORC). In this report the medium is referred to as either working fluid (WF), or refrigerant. In figure 2.1 a RC can be seen and the steps are as following:

1 The medium is a saturated liquid at low pressure.

1→2 A pump increases the pressure of the medium so it can be evaporated at the temperature the heat source provides. This process is ideally isentropic (no heat is transferred between the system and its surroundings, and the transfer of work within the system happens frictionless), but in reality this is not the case.

2→3 An evaporator supplies the saturated liquid with heat to evaporate it. As the medium goes from saturated liquid to saturated vapour the pressure and temperature do not change.

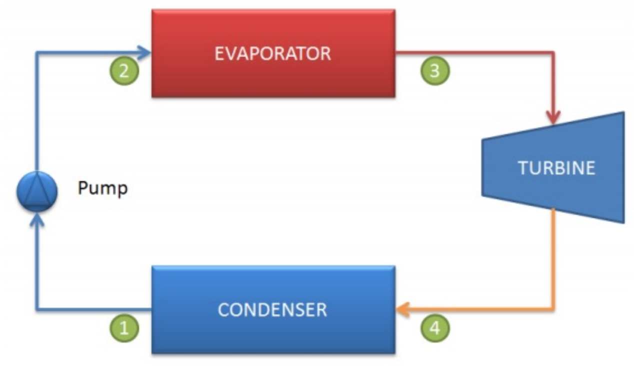

Figure 2.1: Different components and their placement in a Rankine Cycle [2].

3→4 The high pressure saturated vapour now enters the turbine which uses the pressure to produce power. This process is also isentropic ideally.

4→1 The low pressure gas now enters a condenser which will, without changing the temperature or pressure of the medium, condense it into saturated liquid, and the whole process starts again. In reality, the fluid is not saturated, and super heating and sub cooling is even desirable in order to spare damage to the components, and therefore a drop in pressure and temperature will be seen between the inlet and outlet of the evaporator and the condenser

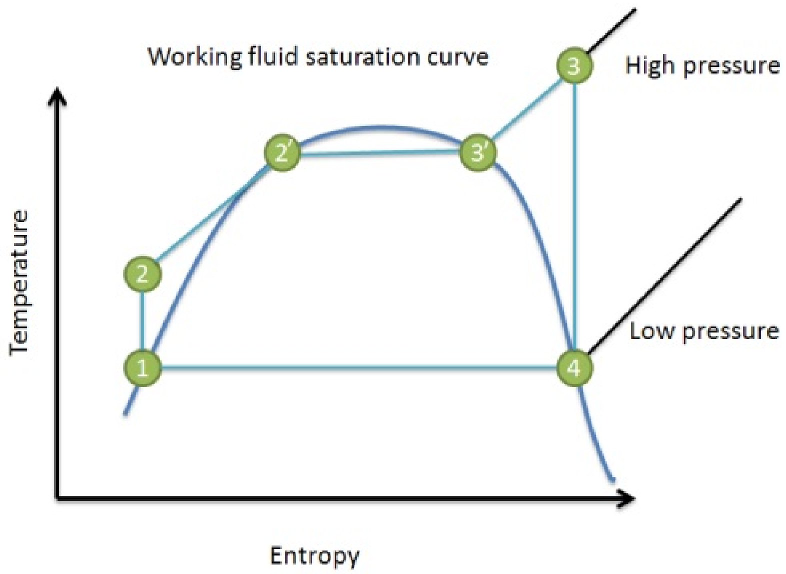

Figure 2.2: The RC on a temperature-entropy diagram for an arbitrary medium [2].

In figure 2.2 the temperature-entropy diagram can be seen for the Organic Rankine Cycle with an arbitrary medium. The vapour is in this case super heated from 3' to 3 in order to achieve a dry vapour to spare the blades of the turbine from condensation and erosion.

2.2 Problematize

Modelon is a global company that provides models and simulations to customers based on open standards. Modelon works with Modelica which is a component-oriented modeling language, and uses Dymola as the environment. Modelon always aspires to expand its catalogue and wishes to have a RC-model in their library. So far only a deficient model has been developed and Modelon has little knowledge about RCs.

2.3 Aim

The original aim of this master thesis was to simulate an ORC, but since the WF used in the test bench which provided data was not organic, a RC was simulated and worked with instead. The principle and behaviour is exactly the same, the only difference is a change to an organic WF, and therefore a change of temperatures the cycle could work with. The aim of this master thesis was to create a model of a RC, and to gain knowledge about the cycle and its behaviour. In order to complete the system the following aims were formulated:

- Model components if no suitable components already existed in Modelon's libraries
- Parameterise all components
- Build system model
- Set robust initialization of the cycle
- Gain insight in control strategies of RCs
- Implement appropriate control strategies

The existing model had some features that did not work properly and therefore special aims were made to correct these:

- The pump model was focused on automotive applications and it was investigated if a new model needed to be developed
- The turbine model was identical to the one used in steam cycles for power plants and it was investigated if a new model needed to be developed

- No control techniques were applied to the model
- There was an error in the model at the exit of the tank/the inlet of the pump

Data from an existing RC in another company was received in order for the model to be tested. The aims with the data was as following:

- To parameterise the model to mimic the properties of the existing RC
- Gain insight in how the control worked
- To test whether the model worked as expected with certain inputs and control

2.4 Focus and limitations

The most important aim of the master thesis was to achieve a working model with good initialization and good control, as well as gaining knowledge of how to model a RC. Since not much information about the existing RC was provided there were some limitations in how well the model could mimic the test bench RC. The project was also limited to 20 weeks of work.

Chapter 3

Method

3.1 Data from company

For this master thesis, data from an existing RC was received from an external company, which develops air-conditioning and heating systems for vehicles. The company wants to develop a turbine for the RC in order to use waste heat from a motor in a vehicle. The company provided Modelon with data from their test bench, but there were no obligations either way, neither promised the other results. The data consisted of 11 data sets with different pressure, mass flow, speed, power etc, at the different inlets and outlets of the components. The data can be seen in Appendix A.

The external company had also swept the torque of the turbine in different working conditions in order to find the maximum power point. This diagram can be seen in Appendix B.

3.2 Procedure of work

In order to understand the RC, the different components and how the system was controlled, academics were consulted and reports on similar projects and articles were studied. In order to create the model it was necessary to understand the Modelica language, which was done by attending a course in Modelica that Modelon held, studying the code and consulting the supervisor or other employees at Modelon.

To mimic the existing RC, information was gathered about the different components. If a suitable model of the different components already existed in Modelon's library, those models were parameterised to have the same features as the existing component, and if no suitable model existed already a new one was created. Then all the components were assembled in the right order to create a RC, and different initializations and control strategies as well as scenarios were investigated.

Chapter 4

Theory

4.1 Simulation

Modelica is an object oriented, equation based simulation language [3]. Modelon uses Dymola as its environment to work with Modelica, and that was used for this master thesis. Dymola has different solvers that can be used to run the simulations, and DASSL and Radau were the ones used mostly. All the simulations were dynamic; they modeled the time varying behavior of the system and did so by solving the equations with numerical methods.

4.2 Working fluid

A fluid called R134a, or 1,1,1,2-tetrafluoroethane, was used as the working fluid in the cycle. The reason was because the external company used that WF, but also because it is a fairly used WF with known properties. R134a is not an organic fluid why the cycle was a RC instead of an ORC. R134a will be replaced eventually because of its affects on the climate, but it could still serve the purpose of this thesis well [4]. In figure 4.1 the pressure/enthalpy diagram for the WF can be seen. This diagram was used to present most of the result and it shows among others at which pressures and enthalpies the WF is a liquid or a vapour.

4.3 Components

4.3.1 Turbine

The turbine used by the external company was a scroll turbine. A scroll turbine has essentially two identical spirals, where one is fixed and the other is moving. In figure 4.2 they are the black and blue parts, where the blue spiral is the moving one. Hot, high pressured gas enters the scroll turbine in the middle, forcing the blue movable spiral to circulate. The gas will travel between the blue and

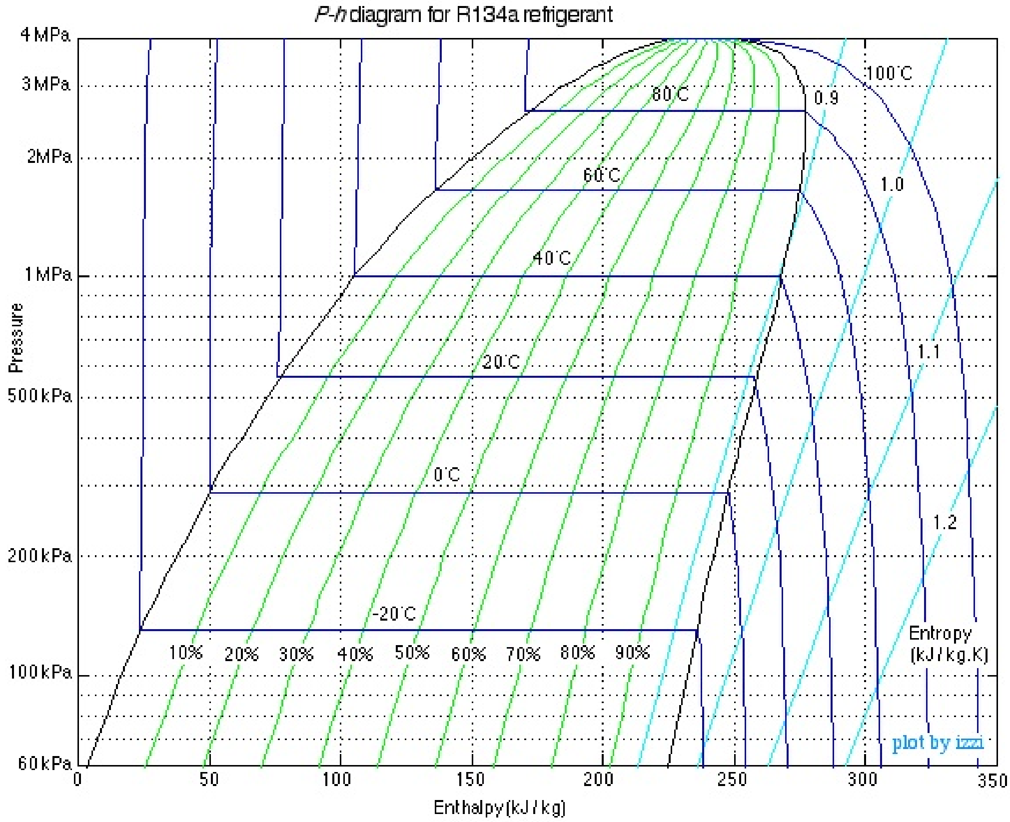

Figure 4.1: A pressure/enthalpy diagram for the WF [5].

black spiral in the cavity that is created between them, and will be released at the end where it has lost pressure.

Since the WF is a gas in the turbine the isentropic efficiency had to be calculated differently compared to the pump. It was calculated using a media model of the WF that Modelon had in their libraries. The media model contained data of the WF. Since the pressure and temperature for both the inlet and outlet of the turbine were given from the data, it was possible to calculate the enthalpy and entropy at the inlet, and the enthalpy at the outlet. Since the entropy at the inlet was calculated, it was possible to calculate what the enthalpy should be at the outlet, if the process had been isentropic. The efficiency was then calculated according to equation 4.1, where h_{outs} is the enthalpy at the outlet of the turbine if the process had been isentropic.

$$\eta_{is} = \frac{h_{in} - h_{out}}{h_{in} - h_{outs}} \tag{4.1}$$

The mechanical efficiency was then calculated according to equation 4.2, where P_{real} is the power actually produced by the turbine, and P_{ideal} is the power that was available.

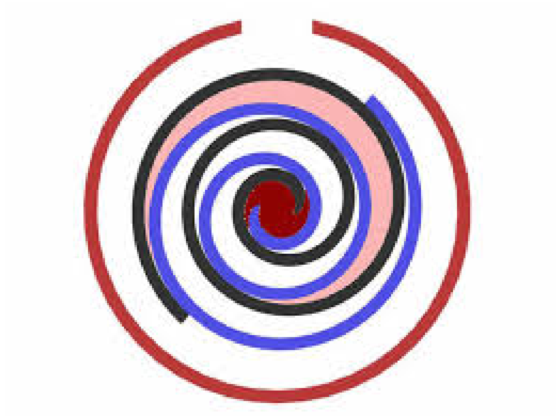

Figure 4.2: Cross-section of a scroll turbine with its two scrolls where one is moving and the other is still [6].

$$\eta_{mech} = \frac{P_{real}}{P_{ideal}} \tag{4.2}$$

The work that was available was calculated according to equation 4.3.

$$P_{ideal} = \Delta h_{sp} \dot{m} \tag{4.3}$$

All the efficiencies for the turbine were calculated from the data. Modelon had a model of a turbine that fitted with the purpose and requirements, except that it had constant values for efficiencies. The overall efficiency for a turbine will vary and is dependent on pressure ratio over the turbine as well as supply pressure. The curve will typically peak at a specific pressure ratio depending on the pump and pressure supply [7]. This was considered as a minor issue, hence a new model was not built.

4.3.2 Pump

To power of the pump needed to be a function of its speed, i.e. revolutions per minute, and different efficiencies, such as mechanical, volumetric and isentropic. It must be able to pump different WF depending on what kind of medium is used.

The pump used by the external company was a Hydracell G10-E. It is a diaphragm pump which works the same way as a human diaphragm. It increases its chamber's volume and thus creating low pressure which will suck in the medium. Then it will decrease the chamber's volume and consequently increase the pressure which will push out the medium and this completes one revolution.

In order to model the pump the displacement volume, revolution per minute, isentropic as well as volumetric and mechanical efficiencies were needed. The isentropic efficiency depends on the property of the WF, the volumetric depends on the geometry of the pump and the mechanical depends on the losses in the shaft. Ideally both the process in the pump and turbine would be isentropic, but alas it is not as can be seen in figure 4.3. In the figure, 2s is for the ideal process and 2 is the real process, and to achieve the same pressure an increase in enthalpy is required. This means that the pump does not manage to pump the fluid to the pressure which theoretically is possible, and the difference is the isentropic efficiency.

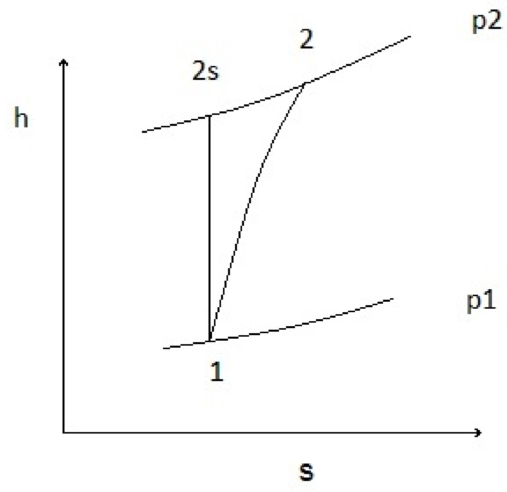

Figure 4.3: Enthalpy and entropy diagram where an isentropic process goes from 1 → 2s. Since the real process is not isentropic, it requires an increase in enthalpy in order to achieve the same pressure, 1 → 2.

The isentropic efficiency is calculated according to equation 4.4, where h_{outs} is the enthalpy out if the process had been isentropic.

$$\eta_{is} = \frac{h_{in} - h_{outs}}{h_{in} - h_{out}} = \frac{W_{ideal}}{W_{real}} \tag{4.4}$$

As it is harder to calculate the enthalpies, the isentropic efficiency was calculated with the the work done during the ideal (isentropic) and the real process. The ideal work was calculated according to equation 4.5, where v is the specific volume, the inverse of density.

$$W_{ideal} = \int_1^2 v_{specific} \mathrm{d}P \tag{4.5}$$

As the liquid is incompressible, the specific volume is not dependent on time and W_{ideal} can be calculated according to equation 4.6, where v is the volume of the liquid.

$$W_{ideal} = v\Delta p \tag{4.6}$$

The change in enthalpy in the real process is calculated according to equation 4.7, where Cp is the specific heat capacity.

$$\Delta h = Cp\Delta T \tag{4.7}$$

And the real work, W_{real} is calculated according to equation 4.8.

$$W_{real} = Cp\Delta T \tag{4.8}$$

The isentropic efficiency is therefore given by equation 4.9.

$$\eta_{is} = \frac{v\Delta P}{Cp\Delta T} \tag{4.9}$$

When a diaphragm pump compresses its chamber, it never manages to push out the whole volume. The volumetric efficiency consequently becomes: Actual volume displaced/Theoretical displacement volume of pump. The pump will due to losses in the shaft use more energy to pump the WF than the WF will gain in the pressure increase. The mechanical efficiency was calculated according to equation 4.11, where the P_{ideal} is the power ideally used by the pump, and P_{real} the power actually used by the pump, where the P_{ideal} is calculated according to equation

$$\eta_{mech} = \frac{P_{ideal}}{P_{real}} \tag{4.10}$$

$$P_{ideal} = \Delta h_{sp}\dot{m} \tag{4.11}$$

Since there was no data on the displacement volume for the pump that the external company had, the volumetric efficiency had to be set to 1 and the volume that was actually displaced was used as displacement volume. It was not possible to calculate the efficiencies of the pump since data at the outlet of the pump was missing. It was not either possible to use the data at the inlet of the boiler as the data for the outlet of the pump because the pressure and temperature changes when passing the pipe between the components.

The efficiencies were instead set by assumptions and experiments. The average pressure drop between the boiler and the turbine was 0.5 bar, hence the pressure at the outlet of the pump was assumed to be 0.5 bar higher than at the inlet of the boiler. For the result it didn't matter if the isentropic or mechanical efficiency fluctuated for the different data sets, and the isentropic efficiency table was chosen to be the one to fluctuate because it seemed more likely that the mechanical efficiency would be more robust. Then different isentropic efficiencies for different pressure ratios and speed were tried and set to match the power used according to data. By choosing a constant pressure drop for all data sets the temperature differences between the pump and the boiler fluctuated a lot. The calculated efficiencies were used in the places in the efficiency table that corresponded to their pressure ratios and speed, and in the other places a mean value was used.

Modelon did not have a model of a diaphragm pump so to be able to fully model the test bench RC that the external company had, a model of a diaphragm pump had to be made. The model was built using existing models that were changed to fit the need.

4.3.3 Evaporator

The external company had a Chevron plate heat exchanger with glycol as secondary fluid. The component was modelled with an already existing model in Modelon's libraries which was parameterised by setting boundary conditions and geometry. In a plate heat exchanger the hot and cold fluids enter and leaves the exchanger according to figure 4.4. In alternating cavities between the plates, hot or cold fluid passes along the plates according to the arrows in the picture, and large areas of heat exchange arises. In the picture the hot fluid has the same direction in all the passes, upwards in this case. The fluids do not have to go the same direction in all of the passes, for example the WF does not have to always go upwards, but can have 2 passes where the fluid go downwards, then 6 passes upwards etc.

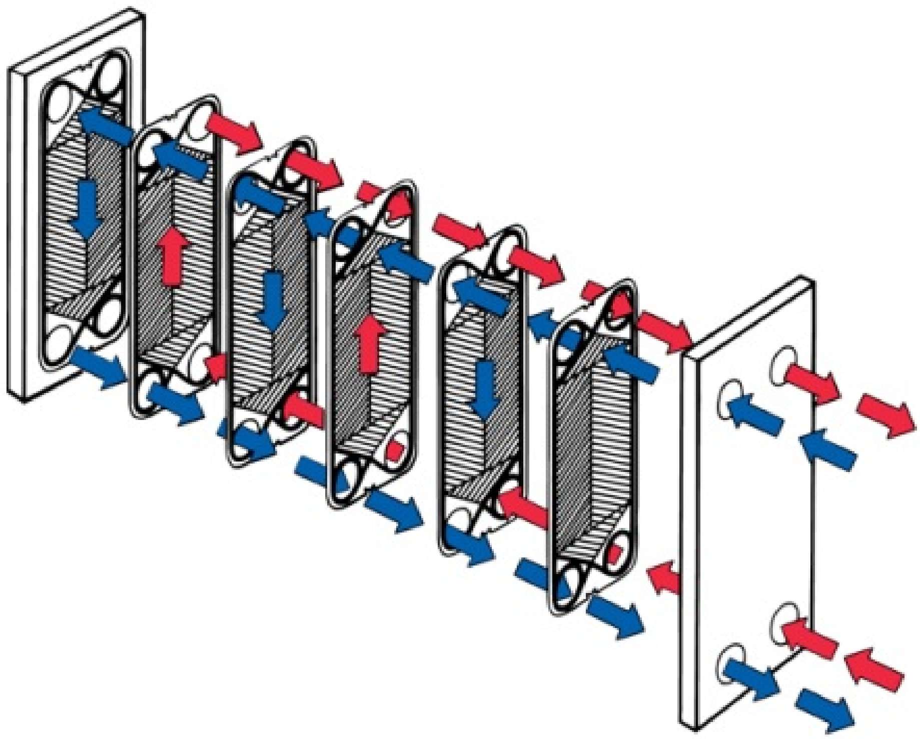

Figure 4.4: A plate heat exchanger [8].

Since some data regarding the geometry of the plate heat exchanger was missing, some estimates were used in the parameterisation. The information that was lacking regarded plate thickness, glycol ratio in the glycol-water mixture and plate chevron angle. The estimates were achieved by looking up typical values for the different parameters. The external company used glycol as the heat carrying fluid, but didn't specify in what concentration. The evaporator model was complex which meant that the simulations took a long time (more than half an hour). Therefore a simpler model was used for most simulations. The complex evaporator took into account and simulated how the WF passed through the gaps between the plates as shown i figure 4.4. The simpler evaporator simulated heat transfer between the two fluids as if they passed each other along two pipes in a counter flow.

The correlations used can be seen in table 4.1. Density profile meant that the boundary conditions for a specific set up was used.

Evaporator	Complex	Simple
WF friction loss model	Two-phase friction using density profile	Two-phase friction using density profile
WF heat transfer model	Two-phase heat transfer for Chevron plate heat exchangers by Martin [9]	evaporation by Chen [10]
Glycol pressure drop correlation	Single phase heat transfer for Chevron plate heat exchangers by Martin	Simple quadratic loss model
Glycol heat transfer correlation	Single phase pressure for Chevron plate heat exchangers by Martin	Constant heat transfer

Table 4.1: Correlations for the evaporators.

4.3.4 Condenser

The condenser was just like the evaporator best modeled by setting its geometry and boundary conditions in an already existing model from Modelon's libraries. The condenser used by the external company was a flat tube heat exchanger, with air as secondary fluid. In figure 4.5 a typical flat tube heat exchanger can be seen, where the warm WF passes through tubes, and between them cold air passes through fins leading to heat exchange and condensation of WF.

There was a lack in data for the condenser and estimates had to be used instead. The information regarded fin and tube geometry, as well as air pressure. The estimates were achieved by looking up typical values for the different parameters. The correlations used for the condenser can be seen in table 4.2

WF friction loss model	Two-phase friction using density profile
WF heat transfer model	Condensation, most refrigerants by Shah [12]
Air pressure drop correlation	Louvered fins after Chang, Hsu et al [13]
Air heat transfer correlation	Louvered fins correlation by Chang

Table 4.2: Correlations for the condenser.

Figure 4.5: A flat tube heat exchanger [11].

4.3.5 Tank

At the outlet of the condenser and before the pump a tank was situated to make sure that the WF was a liquid when it entered the pump, since vapour would damage it. The tank also regulated the amount of WF in the cycle. The tank worked as a buffer for the cycle so it could drain or fill WF, and for this to work the relative level in the tank had to be between 0 and 1. If the tanked reached a relative filling level of 1, the cycle got over charged with WF and the pressure increased in the whole cycle as the density was increased.

4.4 Control

There are different variables that can be controlled in the cycle, and the control strategy relevant depends on the setup of the cycle. The cycle can be controlled with different aims; to run the cycle in a thermodynamically optimal way, or find a configuration where the components work without being damaged. For the setup in this project, the following aspects were considered to control:

- Adequate super heating to let only gas into the turbine
- Adequate sub cooling or other measure to avoid gas entering the pump

- Preferred power or torque from the turbine
- Optimal evaporation temperature
- Low condenser pressure
- Appropriate amount of refrigerant in the cycle

Except for choosing appropriate components that will match the requirements, it is possible to change the following variables:

- The speed of the turbine
- The speed of the pump
- Add tanks after condenser/evaporator
- Drain or charge the cycle with WF
- Change the temperature of the heat source for the evaporator
- Change the temperature of the cooling air in the condenser

The super heating was achieved by changing the speed of the pump. More super heating required a lower speed of the pump as that lowered the mass flow as well as pressure, which meant that the evaporator had more time to bring the WF to a higher temperature. The evaporator temperature can be changed by varying the speed of the turbine [14] [15], as the isentropic efficiency is dependent on the speed. The speed, torque and power of the turbine are correlated to each other according to equation 4.12, and consequently the speed of the turbine was changed in order to obtain the desired torque, while the power was kept constant.

$$Torque(Nm) = \frac{Power(W)}{Speed(rad/s)} \tag{4.12}$$

The amount of WF in the cycle could either be changed by a flow source charge that would either fill or drain the cycle depending on the density desired for the cycle, changing the initial pressure or enthalpy in the components to change the amount refrigerant at the start in the cycle, or by changing the initial level of liquid in the tank. The former has the advantage that the cycle can be over charged. The temperature of the heat source or the cooling air is part of the boundary conditions, and not a way of controlling the cycle per say. It is relevant to know how the cycle behaves when these variables change, as that will most likely happen in real life (for example the temperature of the waste heat from a car motor changes, or the temperature of the outdoor air changes). Control strategies to react when these variable changes might have to be implemented in order to maintain an overall efficiency or power produced.

For this project, PI-controllers were used because they are easy to use and fulfill the requirements. The reason they were chosen over PID-controllers is because they are less sensitive to noise [16]. The equation for a PI-controller can be seen in equation 4.13 where $r(t)$ is the reference value, $y(t)$ the input, $u(t)$ is the output, K the proportional coefficient and T_i the integral coefficient. These coefficients are set by starting with a low value of K (<1), and a high value of T_i (>1), and then increasing respectively decreasing the values until satisfying values are obtained.

$$u(t) = K(r(t) - y(t) + \frac{1}{T_i}\int_0^t r(t) - y(t)\,\mathrm{d}t) \tag{4.13}$$

The external company controlled the test conditions like speed of the pump, ambient temperature, air velocity, mass flow and boiler coolant temperature among others. The speed of the turbine was controlled so that the torque was fixed at 9 Nm, and a tank was placed between the condenser and the pump. The controlling the external company was doing was yet not developed properly, they had acquired the maximum power point for the turbine by sweeping the torque for different pump speeds and air temperatures in the condenser, and collecting data.

4.5 Testing Scenarios

When the cycle was working satisfyingly, different scenarios were tested to gain knowledge about its behaviour. The following tests were carried out:

- Super heating control was tested.
- Means were carried out to ensure only liquid entered the pump.
- The cycle was tested with different torques on the turbine to match the maximum power point tracking diagram the external company had implemented (see Appendix B).
- The cycle was tested with varying amounts of refrigerant.
- The isentropic efficiency of the turbine was changed in order to see how the evaporation temperature changed, and with that also the overall efficiency of the cycle.
- The different data sets were simulated.

In this project the heat source was kept constant. The overall efficiency was calculated according to equation 4.14

$$\eta_{overall} = \frac{W_{turbine} - W_{pump}}{Q_{evaporator}} \tag{4.14}$$

4.6 Initialization

In order to achieve a model that works quickly, it is important to initialize the model well. For this model, good initial values had to be set in all the components as well as the proportional and integral coefficients in equation 4.13 for the PI-controllers. In this cycle the heat exchangers were the most complex models with most states to simulate, why the initialization of them was most important to do well.

4.7 Data

The data received had 11 sets of data and Modelon's data base was used for determining temperature, pressures, enthalpies etc for the different fluids. The different data sets were obtained by varying the pump speed and consequently the super heating. The first 5 data sets also had 8100 g refrigerant in the cycle, and the last 6 data sets had 8200 g refrigerant in the cycle. Then the inlet and outlet pressure and temperature, as well as turbine and pump speed and power was measured.

Chapter 5

Results

5.1 Components

5.1.1 Turbine

The efficiencies of the turbine were possible to calculate since data both at the inlet and the outlet of the pump existed. The isentropic efficiencies were relatively uniform and ranged between 43 % and 55 %, depending on pressure ratio between inlet and outlet pressure, and speed of the turbine. The isentropic efficiency was plotted against pressure ratio and speed to see if there was a good correlation between them. In figure 5.1 isentropic efficiency can be seen for all the 11 data sets as a function of the speed. The red line is a polynomial function of second order, fitted to the data points.

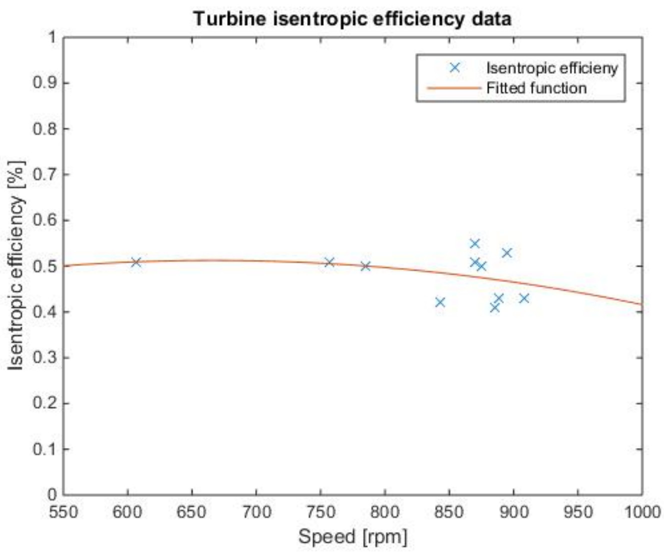

Figure 5.1: Isentropic efficiency for the turbine for the different data sets, as a function of speed.

In figure 5.2 the isentropic efficiencies can be seen for the different data sets as a function of pressure ratio. The red line is a polynomial function of second order, fitted to the data points. The form of the red line in figure 5.1 seems reasonable as you would expect it to have such a form. The fitted function was not accurate enough to be used to predict the isentropic efficiency.

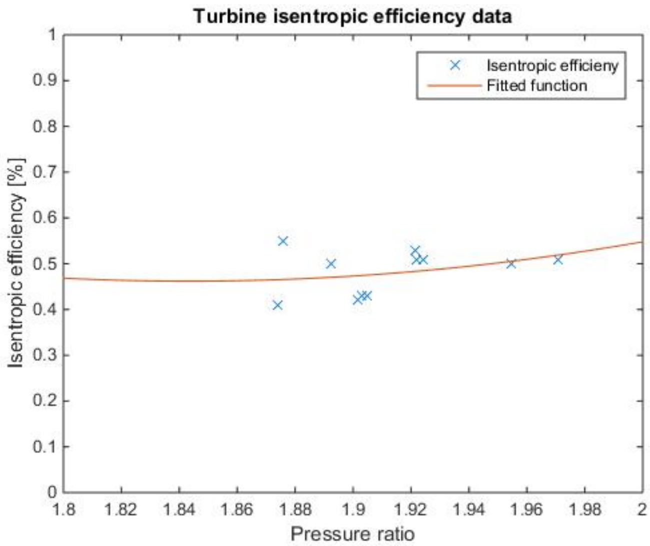

Figure 5.2: Isentropic efficiency for the turbine for the different data sets, as a function of speed.

The form on the red line in figure 5.2 does not seem to have a reasonable form, as it is not expected of the efficiency to have increasing efficiencies as the pressure ratios gets very large or small. It can be seen though that the range of the pressure ratios is not great why it could be assumed that for changed boundary conditions where the pressure ratio was not changed much, the isentropic efficiency could be held constant. Since the model of the turbine had a constant isentropic efficiency (as opposed to the pump where the values could be changed depending on the speed and pressure ratio) a mean value was used at the start of every simulation, which was calculated to 48 %, and then it was changed so that the simulated power of the turbine matched the power in the data. It is hard to know why the isentropic efficiencies are not more consistent with change in variables. It could be because the data was not perfect or because components seldom works according to theory.

The mechanical efficiencies varied more, where all except for 4 data sets were above 100 %. Out of the 4 under 100 % three of them were between 85 % and 91 %, and when inserting the variables for those three data sets into the Dymola model, the outputs fit well with the data. The other data sets stated a mass flow that was too low, in most cases about 5 % too low. If the cycle actually had that mass flow, the turbine would have a mechanical efficiency of more than 100 %. Mass flow is hard to measure which is probably the reason the data is wrong. An average of the good mechanical efficiencies were used, which was 87%.

5.1.2 Pump

As the mass flow in the data was too low in all cases but three, only these three data sets were used to calculate the isentropic and mechanical efficiencies. By experiment the mechanical efficiency was set to 0.73 and the isentropic efficiency ranged between 0.58 % and 0.7 %. Two of the three data sets where the mass flow was consistent with the rest of the data gave an isentropic efficiency of 0.58 % for pump speed 7.5 rad/s and 6.7 rad/s, and pressure ratio 2.27 and 2.33. The last data set gave an isentropic efficiency of 0.7 % for 8 rad/s and a pressure ratio of 2.48. For all other pump speeds and pressure ratio the average 0.62 % was used.

The pump model made was built upon an already existing model of a pump, but the equations were for the most part taken from a positive displacement compressor. The pump model can be seen in figure 5.3.

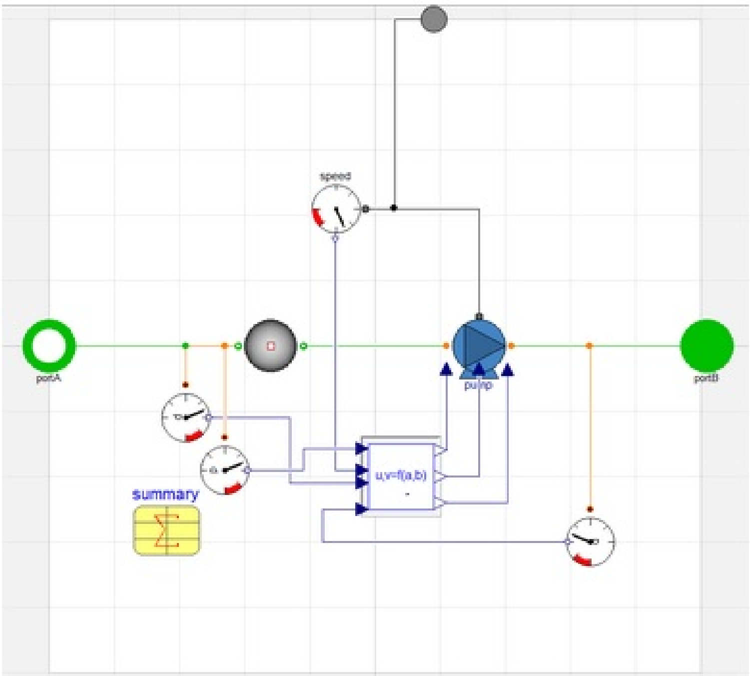

Figure 5.3: How the model of the pump looks like in Dymola.

The structure of the top model can be seen in figure 5.4. Speed of the shaft, density at the inlet and pressure at the inlet and at the outlet were measured and sent to the interface. The interface contained tables of the efficiency characteristics which were set by the user. The tables were dependent on speed and pressure ratio, and the efficiencies were then extrapolated from the tables. The interface also calculates the mass flow from density at the inlet, speed, maximum displacement volume and volumetric efficiency. The interface then sends isentropic and mechanical efficiencies, as well as mass flow to the input pump.

The structure of the input pump can be seen in figure 5.5. The input pump does not do that

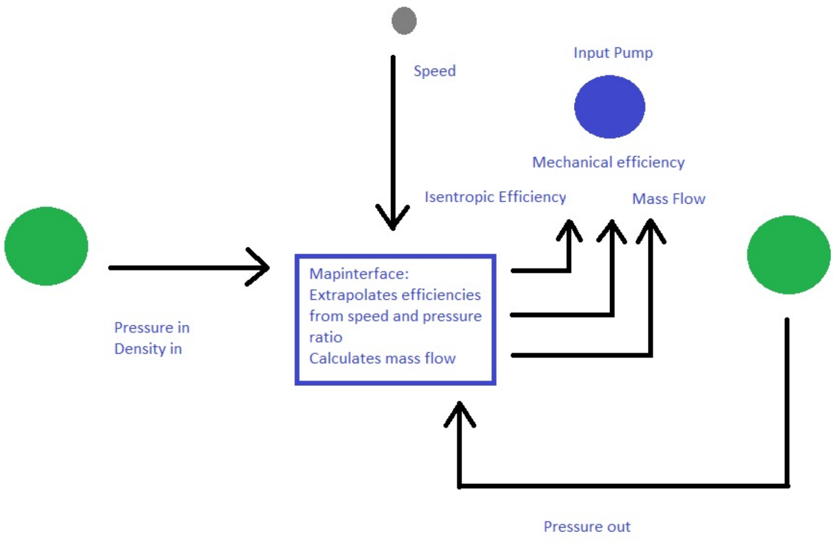

Figure 5.4: The structure of the top level of the pump model.

much in itself. It extends the partial pump and it holds the connection to the shaft and its angular velocity and torque. The partial pump calculates the enthalpy increase from mass flow and isentropic efficiency. It also calculates the shaft torque and power used from energy produced and mechanical efficiency, and leakage if that is present. The partial pump extends partial static flow which contains equations for states at the two ports, as well as energy and mass balance.

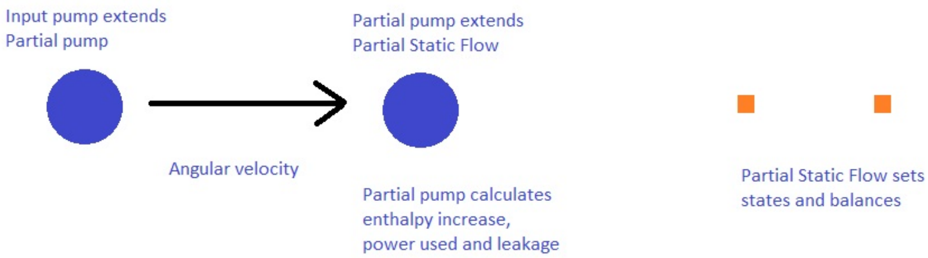

Figure 5.5: The structure of the lower level of the pump model.

Modelica is an equation based language, which means that the order of the equations does not matter, as long as there are as many as unknown variables. The code for the model can be seen in Appendix C.

5.1.3 Evaporator

The model used was very complex and took a long time to simulate (0,5-1,5h h in the whole cycle). Consequently, a simpler model of a plate heat exchanger was used where many of the parameters had to be estimated rather than set from the information from the external company. The simpler model of the heat exchanger meant that the simulations were quicker and control strategies and initialization could be investigated.

5.1.4 Condenser

The existing model used as a condenser served its purpose well.

5.2 The whole system

In figure 5.6, the whole system can be seen as it looks like in Dymola. This is the cycle with the simpler model of the evaporator and a tank between the condenser and the pump.

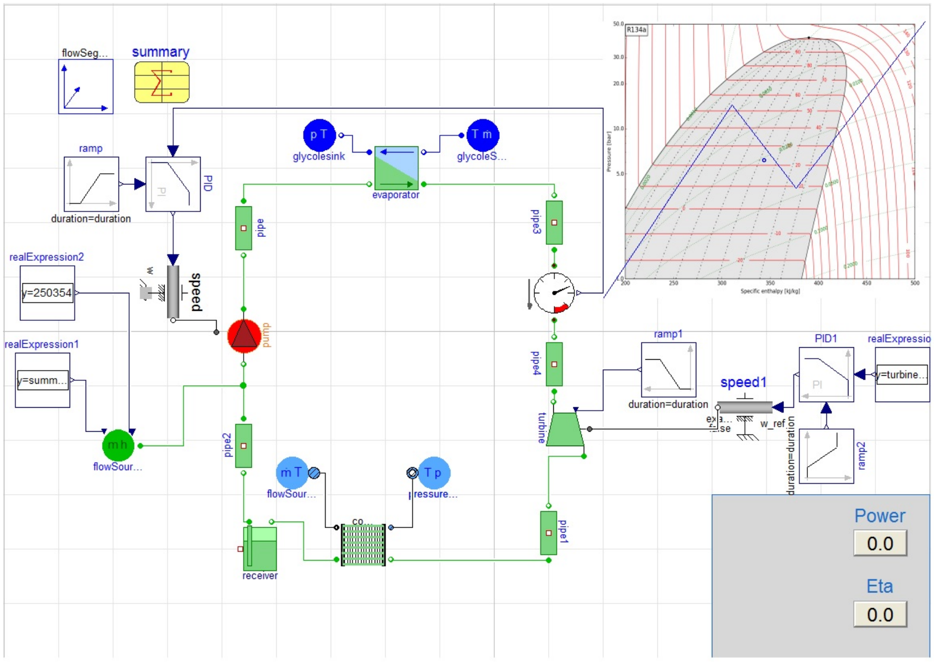

Figure 5.6: The whole RC cycle as seen in Dymola.

5.3 Control strategies

5.3.1 Super heating

In figure 5.6 the super heating control can be seen. The super heating is measured by the clock looking component after the evaporator. The result is then sent to a PI-controller which compares it to a set point, set by the user. The PI-controller will then increase or decrease the pump speed in order for the super heating to be according to preference. The coefficients in the PI-controller were set to $K = 1$ and $T_i = 3$. In figure 5.7 the result can be seen on a ph-diagram for different simulations where $0.5°C$, $5°C$ and $10°C$ super heating were obtained. The simulation used the simpler evaporator.

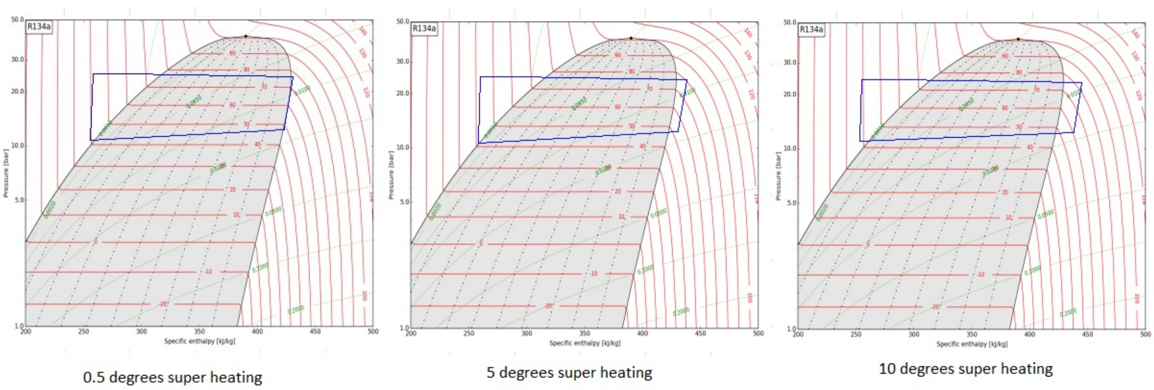

Figure 5.7: The simulation result in a ph-diagram where the speed of the pump was varied in order to achieve different super heating.

5.3.2 Sub cooling

To make sure that only liquid entered the pump a tank was placed after the condenser, where the outlet was collected at the bottom of the tank, making sure only liquid entered the pump. It is also possible to have the tank placed after the evaporator and some other type of sub cooling controlling after the condenser. A cycle with such a set up was created, but with no sub cooling controlling, and consequently the pump received liquid. Since the external company had a tank after the condenser, that set up was used.

5.3.3 Torque

In figure 5.6, the torque control can be seen. The turbine torque is sent in to a PI-controller which increases or decreases the speed of the turbine in order to achieve the desired torque.

In figure 5.8 the recreated power point tracking can be seen. Only 4 of the lines were recreated to show that it was possible to recreate the diagram. Since the only information available about the power point tracking was the temperature of the air in the condenser and the pump speed, the other boundary conditions had to evaluated. Data set 10 was used, and the result matched the data fairly well. The cyan line in the figure was supposed to be 500 rpm, but that was not possible to achieve in the model and still get the WF all the way over the two-phase dome, and consequently the highest value possible was used instead.

In table 5.1, 5.2, 5.3 and 5.4 some data from the simulations where the maximum power point tracking diagram was recreated can be seen.

Temperature in condenser $^\circ C$	15	15	15
Speed of pump rpm	416	416	416
Torque of turbine Nm	5	6	6.2
Power of turbine W	830	920	940
Isentropic efficiency of turbine %	0.45	0.5	0.51
Pressure ratio turbine	1.7	2.18	2.18
Speed of turbine rpm	1576	1461	1442
Overall efficiency %	1	1.3	1.4

Table 5.1: Simulation results for the power point tracking recreation. This table corresponds to the cyan line in figure 5.8.

Temperature in condenser $^\circ C$	15	15	15
Speed of pump rpm	400	400	400
Torque of turbine Nm	5	6	6.4
Power of turbine W	773	865	870
Isentropic efficiency of turbine %	0.42	0.47	0.47
Pressure ratio turbine	2.17	2.17	2.17
Speed of turbine rpm	1471	1375	1289
Overall efficiency %	1	1.3	1.3

Table 5.2: Simulation results for the power point tracking recreation. This table corresponds the orange line in figure 5.8.

Temperature in condenser $^\circ C$	25	25	25
Speed of pump rpm	400	400	400
Torque of turbine Nm	5	6	6.6
Power of turbine W	616	640	672
Isentropic efficiency of turbine %	0.43	0.47	0.49
Pressure ratio turbine	1.88	1.84	1.84
Speed of turbine rpm	1175	1022	964
Overall efficiency %	0.9	1.1	1.2

Table 5.3: Simulation results for the power point tracking recreation. This table corresponds the green line in figure 5.8.

5.3.4 WF drain

In figure 5.6 the WF control can be seen as the green component called flowSourceCharge connected to the inlet of the pump. By setting the set charge point in the charge, different refrigerant charge

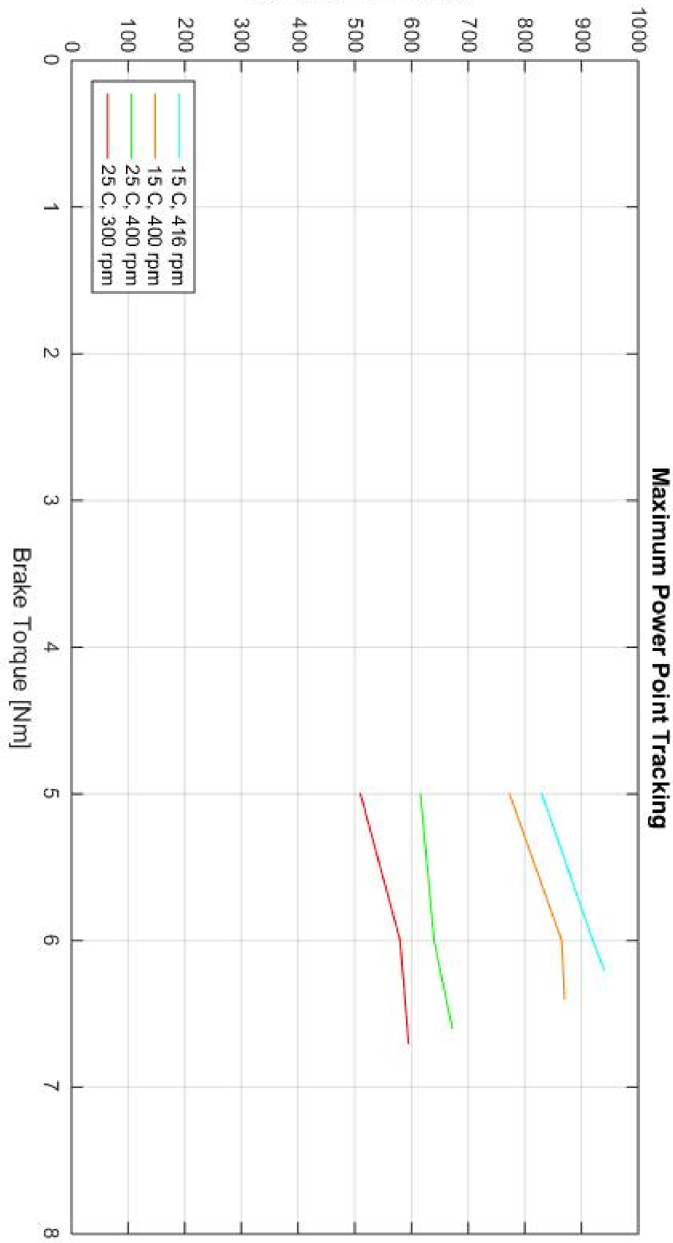

Figure 5.8: Recreated maximum power point tracking.

levels could be achieved in the cycle. The amount of WF in the cycle was only changed by the flowSourceCharge as the initialization of the tank did not seem to work.

Temperature in condenser $°C$	25	25	25
Speed of pump rpm	300	300	300
Torque of turbine Nm	5	6	6.7
Power of turbine W	510	580	595
Isentropic efficiency of turbine %	0.47	0.55	0.54
Pressure ratio turbine	1.71	1.71	1.67
Speed of turbine rpm	955	955	859
Overall efficiency %	1.4	1.8	1.8

Table 5.4: Simulation results for the power point tracking recreation. This table corresponds the red line in figure 5.8.

In figure 5.9 simulation results can be seen of different charges of the cycle. The top left figure shows the simulation result where no drain or filling was added, and the relative level of the tank is 1, which means that the cycle was overcharged and the density was $1088kg/m^3$. The top right was drained to $900kg/m^3$ from the start, the bottom left drained to $500kg/m^3$ after 60 s and bottom right drained to $300kg/m^3$ after 60 s.

As can be seen for the top right and the bottom left figures, the point for the inlet of the pump is in the two-phase dome, which means that the pump receives WF that is not only liquid, which will damage the pump. The relative level in the tank is above 0.4, and the suction of WF is below that, which means that the result is concerning since the behaviour of the tank is not as expected. Bernoulli's principle can be seen in equation 5.1, where v is the fluid flow speed at a point on a streamline, g is the acceleration due to gravity, z is the elevation of the point above a reference plane, p is the pressure at the chosen point, and ρ is the density of the fluid at all points in the fluid [17].

$$\frac{v^2}{2} + gz + \frac{p}{\rho} = constant \tag{5.1}$$

Because of Bernoulli's principle there might be an issue with the speed of the refrigerant before it enters the pump, which means that the pressure is too low to maintain the fluid as a liquid, and it evaporates. This could be the reason that the pump receives vapour in the cycles with under charge. The pipes have a way of introducing a static head in order to increase the pressure, which was tested but due to lack of time this could not be investigated more.

As can be seen in the bottom right diagram, when too little WF is present in the cycle, the cycle collapses. If the cycle is over charged too much the WF will never leave liquid phase and the cycle will not behave as desired. If the cycle is on the verge of being too over or under charged the overall efficiency will decrease as either the evaporator temperature required by the WF is too high, or too low. When the draining begins there is a slight increase in mass flow and power of the turbine.

In figure 5.10 the result from other draining simulations can be seen. The simulation to the left is the simulation which is closest to the data set. It is over charged, which means that the WF in the

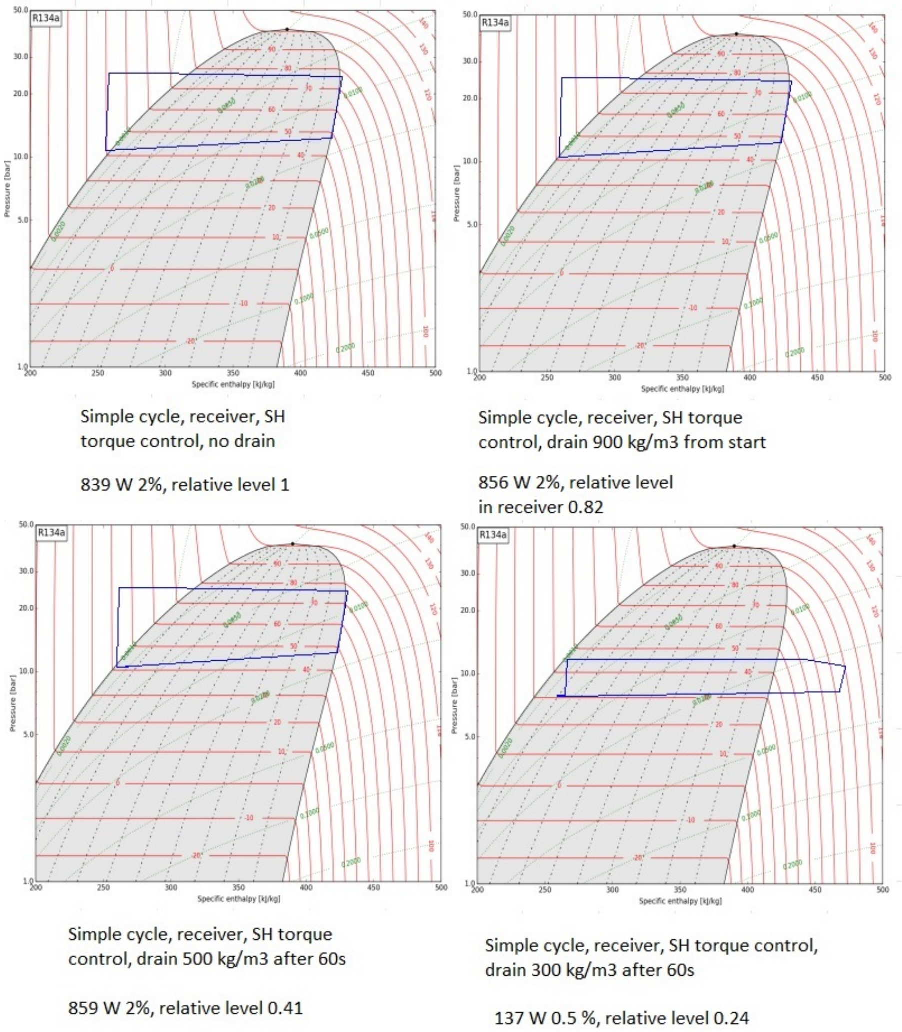

Figure 5.9: Simulation result when the cycle is drained of WF.

tank is not on the edge of the two-phase dome as it normally is, but pushed into the liquid area, but still within the boundaries of what the cycle can handle. The simulation result to the right shows the result where the cycle is not over charged, but the pump still only received liquid.

5.3.5 Varying evaporation temperature

According to articles [14] and [15], it would be possible to change the evaporation temperature by changing the speed of the turbine. A change in the speed of the turbine would mean a change in the isentropic efficiency, and therefore the evaporation temperature. This was tested but the result was that the evaporation temperature did not change.

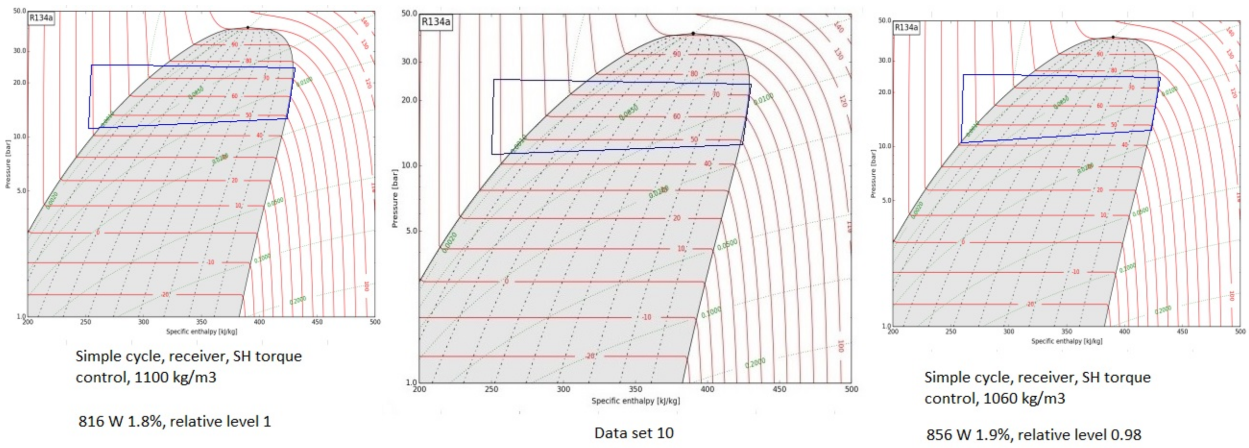

Figure 5.10: Simulation results with different draining and filling.

5.4 Simulation results of different data sets

Below in figure 5.11 the result from a simulation can be seen. The simulation is of data set 10, and the enthalpy - pressure diagram of the data points can be seen to the right, and the simulation result can be seen to the left. In this case the simple evaporator was used, and the tank was placed between the condenser and the pump. No control strategy was implemented in this simulation. The pump only received liquid and the overall efficiency for the whole cycle in this simulation was 1.8%

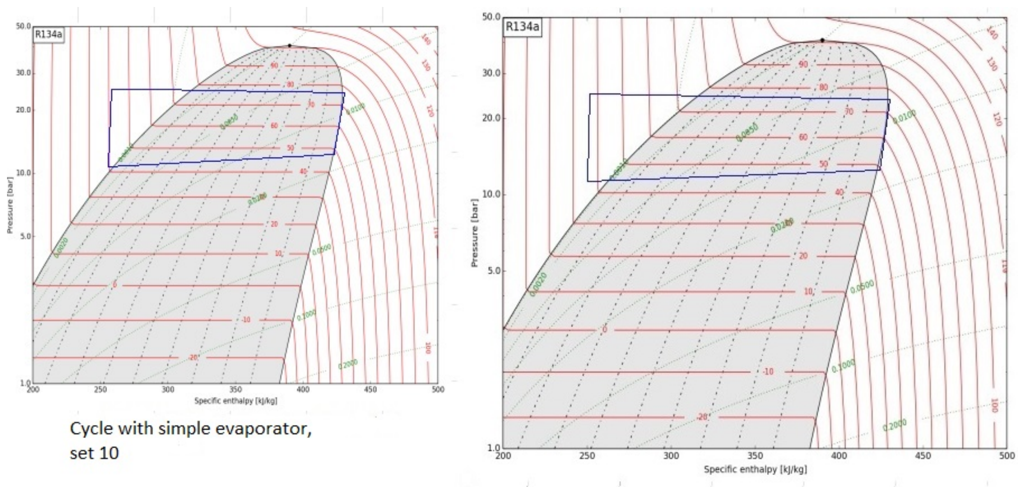

Figure 5.11: Simulation result for data set 10, on a pressure/enthalpy diagram to the left, data to the right.

In figure 5.12 the result from a simulation with the complex evaporator can be seen. The simulation is with data from data set 10, a tank is place between the condenser and the pump and no control was done. The simulation took half an hour.

In figure 5.13 the simulation result can be seen where data set 2 was simulated instead of data

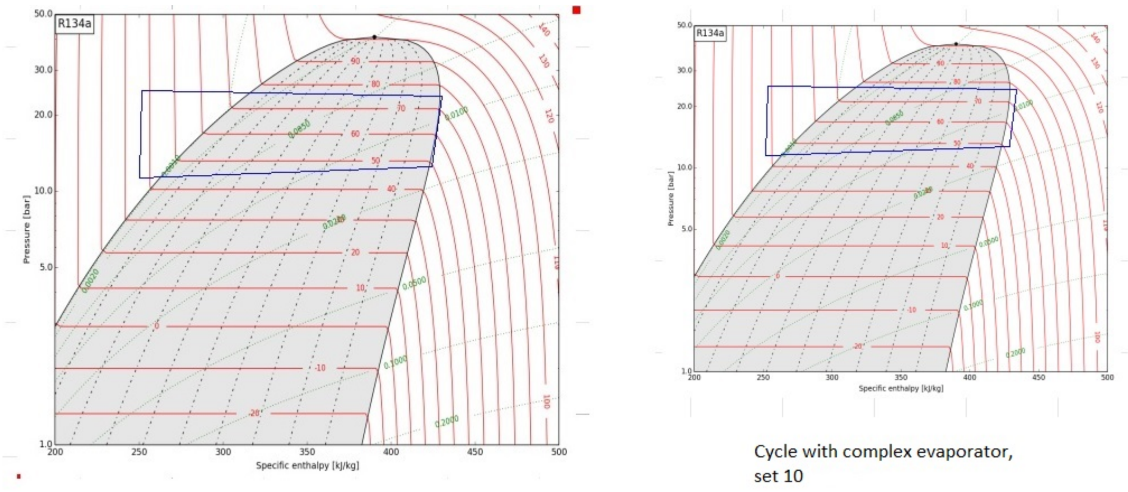

Figure 5.12: Simulation result on a pressure/enthalpy diagram to the right, data to the left. Cycle with complex evaporator.

set 10. The points according to the data can be seen in the left figure, in the middle the correlations in the heat exchangers were set for data set 10 but the SH was changed to fit data set 2. In the figure to the right the correlations were set to fit data set 2, and then the SH was changed to fit set 2. In figure 5.14 and 5.15 the simulation result for data sets 6 and 9 can be seen.

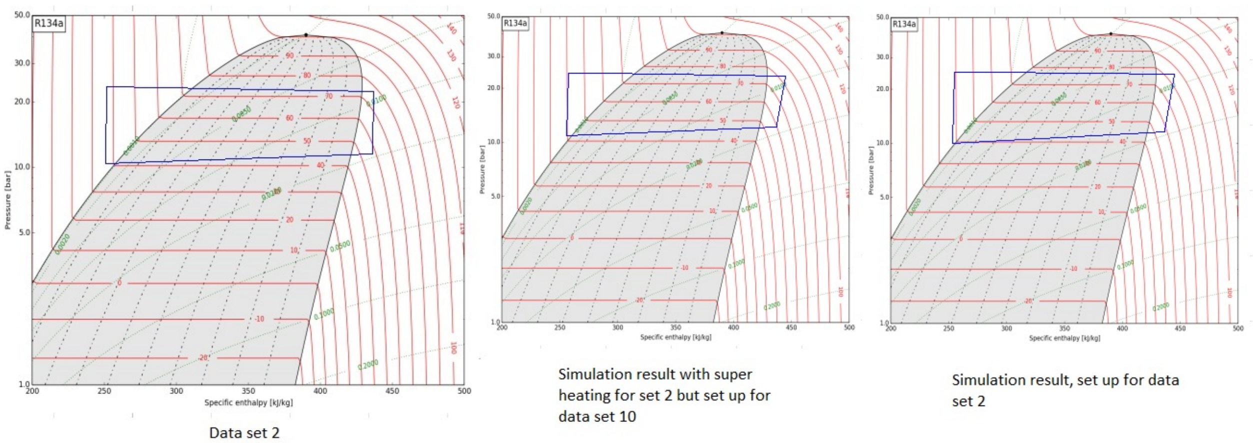

Figure 5.13: Simulation result for data set 2.

In table 5.5 and 5.6 simulation results from simulations of data set 2 and 6, and 9 and 10 can be seen. The pressure ratio error was calculated according to equation 5.2, where Pr is the pressure ratio. Due to the fact that some data sets were not coherent, data set 2, 6 and 9 out of the sets presented here, it was expected that the simulation result would no match the data as well as data set 10, which also can be seen in the tables. Focus for the simulations was to achieve matching

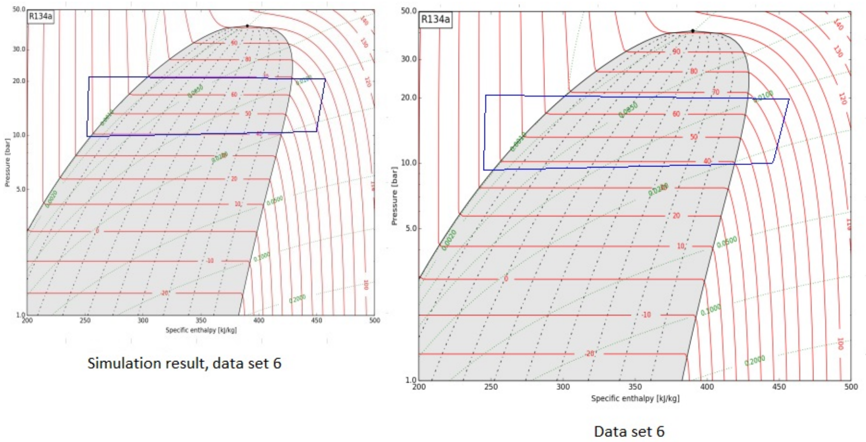

Figure 5.14: Simulation result for data set 6.

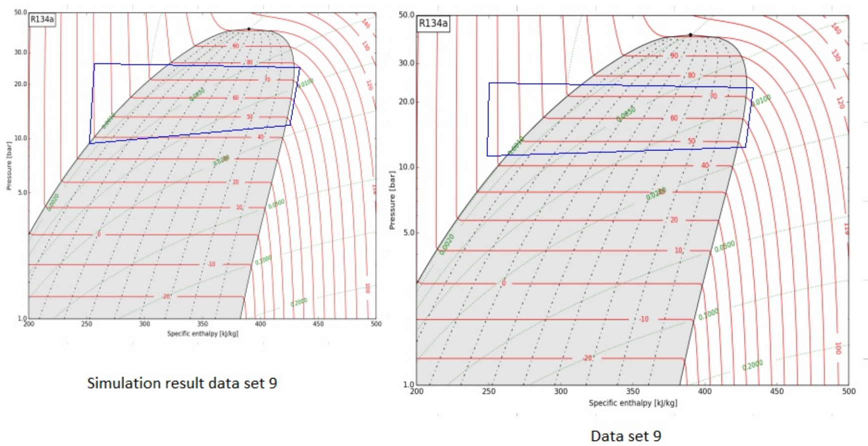

Figure 5.15: Simulation result for data set 9.

turbine power and SH, and in order to do that, the isentropic efficiency for the turbine was varied.

$$error = \frac{|Pr_{data} - Pr_{simulation}|}{Pr_{data}} \tag{5.2}$$

Data set	2	Simulation	6	Simulation
Power W	820	816	572	580
Overall efficiency %	2.7	1.4	2.6	2.3
Mass flow kg/s	0.102	0.108	0.066	0.085
SH $^{\circ}C$	9.6	9.6	20.7	20.7
Speed of turbine rpm	870	859	607	611
Speed of pump rpm	300	318	200	250
Torque of turbine Nm	9	9	9	9
Turbine pressure ratio	1.92	2.02	1.99	1.9
Pressure ratio error %	-	5	-	4.5
Isentropic efficiency turbine	-	0.59	-	0.54

Table 5.5: Simulation results for different data sets.

Data set	9	Simulation	10	Simulation
Power W	834	836	820	827
Overall efficiency %	2.2	1	1.7	1.8
Mass flow kg/s	0.12	0.12	0.138	0.138
SH $^{\circ}C$	2.5	2.5	0.5	0.5
Speed of turbine rpm	885	886	870	879
Speed of pump rpm	350	343	400	406
Torque of turbine Nm	9	9	9	9
Turbine pressure ratio	1.87	2.03	1.89	1.84
Pressure ratio error %	-	8.6	-	2.6
Isentropic efficiency turbine	-	0.61	-	0.6

Table 5.6: Simulation results for different data sets.

Chapter 6

Discussion

The aims of the master thesis were fulfilled, and the project was carried out within its time limit. The result was a working model of a rankine cycle, and basic understanding of its behaviour was obtained. There are suggestions for future work concerning the rankine cycle but the result from this project is a base that can be built upon.

The original aim of this master thesis was to simulate an organic rankine cycle, but the WF used, R134a, was not organic, and hence the cycle was not an organic rankine cycle, but a rankine cycle. The cycles work principally the same way, with the same behaviour, and therefore it should be possible to change the WF and some boundary conditions in order to have a simulation of an ORC instead of a RC.

A source of error for the parameterisation of the components was the error in the data. Some data was missing and some was wrong which meant that it was hard to calculate the efficiencies. Since a constant pressure drop between the pump and evaporator was assumed the isentropic efficiency for the pump was uncertain since the pressure drop most likely fluctuates. The values achieved were reasonable and as accurately calculated as possible according to the data. Since only three data sets were used to calculate the mean efficiencies for the pump and the mechanical efficiency for the turbine, the accuracy of the result is not as high as it could be.

The incoherence of some of the data sets also meant that the simulations of those sets could not match the data exactly. It was decided that the power of the turbine was focus, as well as the SH, and the cycle was controlled in order to achieve that. Apart from that the simulations matched the data well.

There was a lack of information regarding the components. Some facts about them were missing and estimates were used if no other method was available. Another source of error was the fact that the existing turbine model used constant values for mechanical and isentropic efficiencies. Consequently there would be inconsistencies in the model compared to the data received from the external company. This was avoided by manually changing the efficiency for different simulations, but in order to have a turbine that would act realistically for varying boundary conditions a turbine with an efficiency that was dependent on speed and pressure ratio is needed. If that kind of turbine had existed, it would have required more data regarding the isentropic efficiency's dependence of speed and pressure ratio, if other boundary conditions was to be tested.

the external company had not yet developed their control strategies which meant that it was not possible to mimic exactly any existing control. Instead control strategies talked about in other reports were used and tested. Because of this it was harder to understand how the cycle behaved for the data points we obtained from the external company, and consequently how the cycle behaved in general.

The complex plate heat exchanger, that worked as an evaporator, was ideal for the cycle but almost impossible to use since every simulation took 1,5 h. The simpler evaporator was parameterised to give sufficiently good values in less time (2-10 min), but was only a simple reflection of reality, instead of actually simulating what happens in a plate heat exchanger.

When a tank was used after the condenser, the states of the WF would not be exactly according to the data which is because of tank property. Even though the simulation did not give exact result, the pump only received liquid which is more important. Some issues were encountered regarding the tank when the cycle was charge or drained of WF. The first issue was that vapour left the tank when it was expected to only discharge liquid when the cycle was under charged. The other issue was that when the initial liquid level was changed, nothing happened.

The tank model used was the only one existing, and there was a theory that introducing a static head to the tank could solve the problem. Due to lack of time, this issue was not investigated further in this project, and other means had to be implemented to get around the problem.

The overall efficiency for the simulations of the data sets was around 2 %. As the cycle is quite small it would be expected to not be high. The external company was interested in developing turbines, or positive displacement machines, and not achieving a high efficiency which is another reason the efficiency was so low. The overall efficiency could have been increased by for example using the heat of the gas after the turbine to heat the gas after the pump. The overall efficiency could also, according to some articles, be changed by changing the speed of the turbine and hence varying the evaporating temperature. This correlation was not found to happen in this simulation, and there was no time to investigate it further.

It was possible to recreate the maximum power point tracking diagram to some extend, but not completely copy the results. There was not much information regarding the diagram, and because of this assumptions had to be made as to what the conditions were for the different tests, which meant that it was not possible to know why the recreation did not match the original tracking. The main difference between the simulation and the diagram provided by the external company is that they had pump speeds of 500 rpm, which was not possible to achieve while still completely evaporate the WF. In the tracking diagram, the company also only tests torques up to $7Nm$, and then from the diagram assumes that the optimal torque is $9Nm$. It would have been useful for this project to have a complete power point tracking diagram.

Chapter 7

Conclusion

A model of a rankine cycle was created and it was possible to parameterise it to fit the data that the external company had provided. The model worked better than the original model, and knowledge about its behaviour was obtained. The original model was only used to gain knowledge about the cycle, and to work as a source of inspiration. A new model of a pump was made, otherwise models that already existed were used. All components were parameterised according to the data received by the external company, and the cycle was built and initialized. Insight in control strategies were obtained and basic control was implemented. The model that was parameterised according to the data gave results that matched when simulated . The original aim was to create a model of an organic rankine cycle, but the working fluid used was not organic why a rankine cycle was modeled. The cycles work principally the same, why the knowledge gained from this project can be used for an organic rankine cycle.

It was found out that the data was not consistent, some data sets seemed to have a mass flow that was too low. The lack of some data, especially about the components, meant that some parameters had to be evaluated. This meant that the components were parameterised to fit the specific data sets and would potentially not work accurately for other boundary conditions. Because the amount of data was not large it was not possible to parameterise the components to work for a large variety of boundary conditions, but it showed that it was possible to simulate a RC.

The tests and simulations carried out were adequate in order to acquire a basic understanding of the behaviour of the cycle. If more data had been available, it would have been possible to establish how the turbine's isentropic efficiency was dependent of speed and pressure ratio. There was also an issue with the model of the turbine used, that it had constant efficiencies which made it hard to model varying efficiencies.

It was possible to recreate the maximum power point tracking diagram the external company had created to some extent, but it did not match perfectly. It was not possible to investigate the cause of this because of missing information. A complete diagram would have been useful for this project.

Different data sets were simulated and the ones where the data was coherent the result matched well. For the sets where the data was not coherent, the result did not completely match the data set, and the simulation was focused on achieving a correct turbine power. It is not unusual that the data is not coherent, it is for example hard to measure mass flow.

Different super heating was tried and the model adjusted accurately in order to achieve the desired super heating. Knowledge about how the cycle behaved with less or more WF was obtained. There was a range of WF charge where the cycle worked well, more or less WF made the cycle collapse. A slight over charge resulted in a simulation result that matched the data best. The tank did not behave entirely as expected which needs to be investigate further.

The control strategies implemented worked well, and the initialization was satisfactory. The overall efficiency for the cycle, albeit being low (2%), is legit, and could potentially be changed by the speed of the turbine. This correlation could not be confirmed and further investigation is needed. The simulations with the simple evaporator took between 2 and 10 minutes which is fast enough to work with for this project. The result from this project can be used by Modelon as a base for further investigations as well as to be used in other projects.

Chapter 8

Future work

There is work that could be carried out in order to develop the rankine cycle further. A turbine model which has efficiencies that can be varied depending on the speed and pressure ratio, is crucial for a model that can be used with different boundary conditions. The initialization worked well for this project, especially the simulations with the simple evaporator where the simulations took between 2 and 10 minutes. The initialization could be improved in order to achieve even faster simulations. In this project, start up and close down have not been investigated at all, and would be relevant in order to gain more knowledge about the cycles behaviour.

In this project, the behaviour of the cycle was investigated for basic control, but in order to gain more knowledge other types of control strategies or cycle set ups should be tried out. For example: varying heat source in evaporator and heat sink in condenser, changing WF, adding turbines, heat the WF after the pump with the heat from the WF after the turbine and changing place of the tank. The behaviour of the model of the tank needs to be investigated as it behaved in a way that was not expected; the initialization didn't seem to work as it should and the fluid in the outlet was vapour when it should have been liquid.

A simpler version of an evaporator was used for this project in order to save time, but to achieve more accurate results the complex one should be investigated more. All the results from simulations in this report come from cycles with the simple evaporator and it would be beneficial for Modelon to redo them with the complex evaporator. The correlation between the speed of the turbine (and therefore the isentropic efficiency of the same) and the evaporating temperature needs to be investigated.

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Appendices

Appendix A

Data from the external company

The data received from the external company.

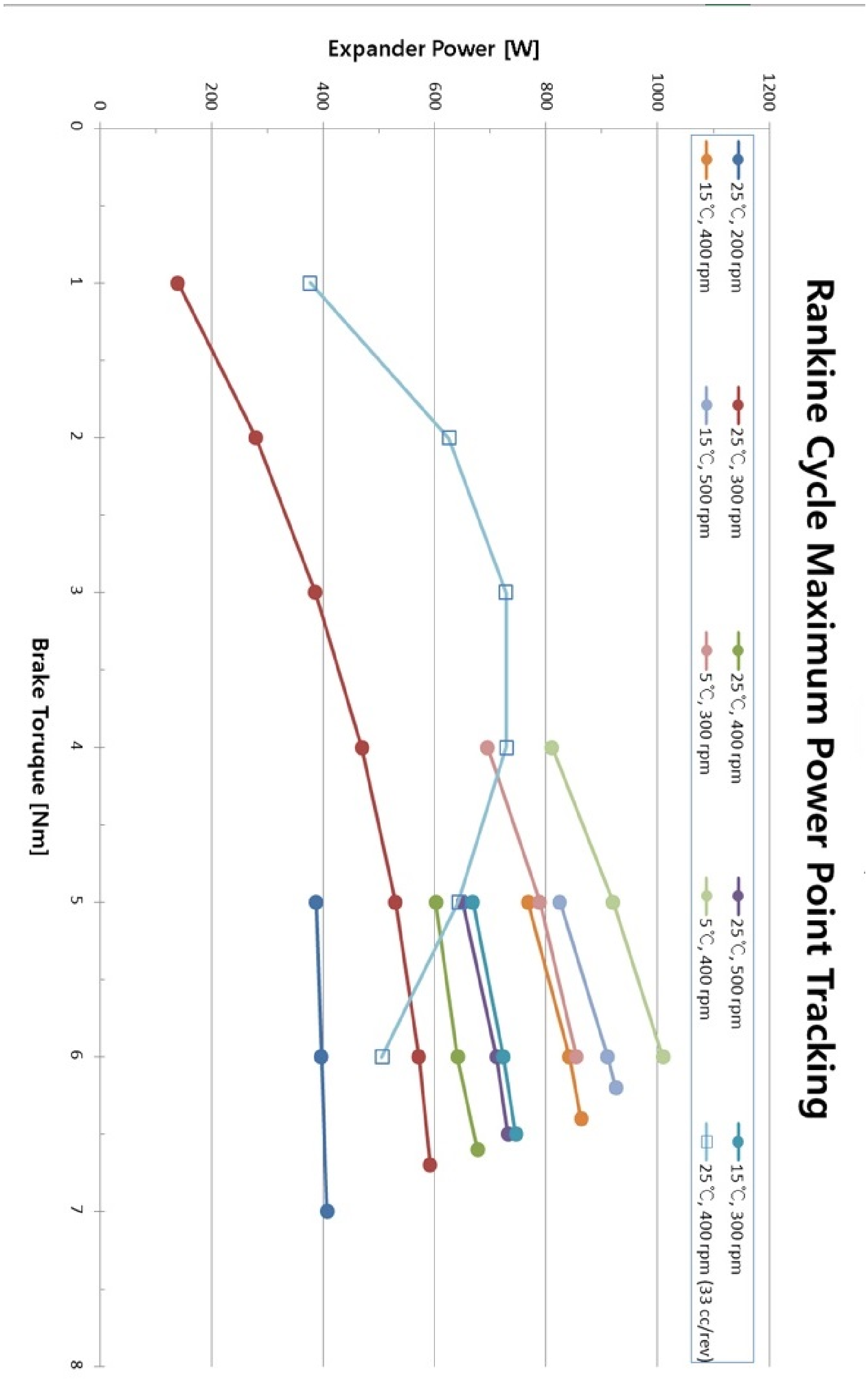
Performance (@Pump SPEED)

EXP	MTR	EXP	EXP	PUMP		NET	Glycol			Cond In Air		Refrig erant mass	PRESSURE & TEMP (R134a)																				Ref
SPD	RPM	Torque	POWER	V	A	Power	Temperature		volume flow	DB	Air		PUMP			Boiler				Expander					COND				Reservoir				Flow
.										Temp	Flow	매	IN		S.C	In		Out		In		S.H	Out		In		Out		In		Out		
rpm	rpm	N.m	W	V	A	W	IN	OUT1	ℓ	°C	SCMH	g	kg/㎠G	°C	°C	kg/㎠G	°C	kg/㎠G	°C	kg/㎠G	°C	°C	kg/㎠G	°C	kg/㎠G	°C	kg/㎠G	°C	kg/㎠G	°C	kg/㎠G	°C	kg/hr
B0243BXB03_2	R134a _90'C_30LPM COND Room 25'C Air Velocity 4.5m/s												2900	3080		HVCC Boiler				HVCC 75ABR1.83 Expander ; 75cc, Asymmetric scroll, thrust Rac													
785	250	9,0	740	67	3,9	479	90	82,2	30	24,9	3586	8100	9,0	36,0	2,7	21,5	36,6	21,0	85,9	20,5	86,0	16,1	10,0	63,7	10,0	61,8	9,0	36,1	9,0	36,1	9,0	35,4	302
870	300	9,0	820	78	3,8	524	90	80,9	30	25,1	3585	8100	9,6	36,5	4,4	23,0	37,1	22,4	82,7	21,9	82,3	9,6	10,9	58,5	10,8	57,6	9,7	36,7	9,7	36,7	9,6	36,0	366
908	350	9,0	856	91	3,9	501	90	79,9	30	25,0	3585	8100	9,9	37,4	4,6	23,9	38,2	23,2	77,3	22,6	77,3	3,2	11,4	52,8	11,3	52,1	10,0	37,5	10,0	37,5	10,0	36,9	427
889	400	9,0	838	102	3,9	440	90	79,4	30	25,0	3585	8100	9,9	38,1	3,9	24,5	39,0	23,7	76,1	23,0	75,4	0,6	11,6	48,8	11,5	48,3	10,0	38,2	10,0	38,2	10,0	37,6	492
895	450	9,0	844	114	3,9	399	90	79,6	30	25,0	3586	8100	9,7	39,1	2,2	25,0	40,0	24,1	77,0	23,4	76,1	0,5	11,7	48,8	11,6	48,3	9,9	39,2	9,9	39,2	9,8	38,5	552
607	200	9,0	572	55	3,8	363	90	83,8	30	24,9	3586	8200	8,5	32,5	4,4	20,1	33,2	19,7	87,2	19,3	88,0	20,7	9,2	66,4	9,2	63,8	8,5	32,6	8,5	32,6	8,5	32,0	237
757	250	9,0	713	68	3,9	448	90	81,5	30	25,0	3586	8200	9,6	33,2	7,7	22,0	33,9	21,5	85,1	21,1	85,0	13,9	10,5	62,7	10,4	61,1	9,6	33,2	9,6	33,2	9,6	32,8	309
843	300	9,0	795	79	3,9	486	90	80,5	30	24,9	3585	8200	10,2	34,0	9,0	23,3	34,9	22,7	81,7	22,2	81,3	8,0	11,2	58,4	11,1	57,1	10,2	34,1	10,2	34,1	10,2	33,7	370
885	350	9,0	834	91	3,9	479	90	79,9	30	25,0	3586	8200	10,5	35,1	8,9	24,1	36,0	23,4	76,9	22,8	77,0	2,5	11,7	52,8	11,6	52,0	10,6	35,2	10,6	35,2	10,6	34,7	432
870	400	9,0	820	103	3,9	418	90	79,4	30	25,0	3586	8200	10,5	36,0	8,0	24,6	37,0	23,8	76,4	23,2	75,7	0,5	11,9	49,5	11,8	49,1	10,7	36,1	10,7	36,1	10,6	35,6	497
875	450	9,0	825	115	4,0	365	90	79,4	30	25,0	3586	8200	10,4	37,0	6,7	25,1	38,0	24,1	77,2	23,6	76,3	0,3	12,0	49,6	11,9	49,1	10,6	37,0	10,6	37,0	10,6	36,5	559

Appendix B

Maximum Power Point Tracking

The maximum power point tracking diagram carried out by the external company.



Rankine Cycle Maximum Power Point Tracking
Expander Power [W]
Brake Toruque [Nm]
25°C, 200 rpm
15°C, 400 rpm
25°C, 300 rpm
15°C, 500 rpm
25°C, 400 rpm
5°C, 300 rpm
25°C, 500 rpm
5°C, 400 rpm
15°C, 300 rpm
25°C, 400 rpm (33 cc/rev)

Appendix C

Pump code

Equations and code that made up the pump model.

C.1 Interface

```
massFlow = max(d_in*eta_Vol*V_MaxDisplacement*speed, 0);
```

C.2 Input pump

```
  der(shaft.phi) = omega;
  shaftTorque = shaft.tau;

  N=Modelica.SIunits.Conversions.to_rpm(omega);

  m_flow_pump = massFlow;
  eta_is = isenEff;
  eta_mech = mechEff;
```

C.3 Partial pump

```
 eta = eta_is*eta_mech;

  h_in = Medium.specificEnthalpy(stateA_inflow);
  h_is_lim = max(h_in, h_is);
  h_out = (h_is_lim - h_in)/max(0.01,eta_is) + h_in;

  if Medium.analyticInverseTfromh then
```

```
  state_outflow = Medium.setState_phX(portB.p, h_out,
     portB.X_outflow);
  T_out = Medium.temperature(state_outflow);
else
  state_outflow = Medium.setState_pTX(portB.p, T_out,
     portB.X_outflow);
  Medium.specificEnthalpy(state_outflow) = h_out;
end if;

if torqueFromDisplacement then
  shaftTorque = (h_out-h_in)*dA_inflow*eta_Vol*Vd/(eta_mech*2*
     Modelica.Constants.pi);
  W_tot = shaftTorque * omega;
else
  W_tot = P_ext/eta_mech + (h_out-h_in)*m_flow_leakage;
  shaftTorque = W_tot/max(1, omega);
end if;

dp = portB.p - portA.p "Pressure increase";

m_flow_leakage = if internalLeakage then  mflow0_leakage*dp/
   dp0_leakage else 0;
m_flow = m_flow_pump - m_flow_leakage;
```

C.4 Partial Static Flow

```
if Medium.analyticInverseTfromh then
   stateA_inflow = Medium.setState_phX(portA.p, inStream(
      portA.h_outflow), inStream(portA.X_outflow));
   T_in = Medium.temperature(stateA_inflow);
 else
   stateA_inflow = Medium.setState_pTX(portA.p, T_in, inStream(
      portA.X_outflow));
   inStream(portA.h_outflow) = Medium.specificEnthalpy(
      stateA_inflow);
end if;
```

```
// Densities for inflowing fluid
  dA_inflow = Medium.density(stateA_inflow);
  h_is = Medium.isentropicEnthalpy(portB.p, stateA_inflow);
  h_out = portB.h_outflow;

  V_flow = m_flow/dA_inflow;

// Instantaneous propagation of enthalpy flow between the two
   ports with
//defined enthalpy increase in flow direction (no storage and no
   loss of energy)
  portA.h_outflow = inStream(portB.h_outflow);
  portA.X_outflow = inStream(portB.X_outflow);
  portB.X_outflow = inStream(portA.X_outflow);
  portA.C_outflow = inStream(portB.C_outflow);
  portB.C_outflow = inStream(portA.C_outflow);

// Mass balance
  0 = portA.m_flow + portB.m_flow;
  0 = portA.m_flow*actualStream(portA.h_outflow) + portB.m_flow*
     actualStream(
    portB.h_outflow) + P_ext;
// Momentum balance, to be supplied in subclass
  m_flow = portA.m_flow;
```

```
equation
  if Medium.analyticInverseTfromh then
    stateA_inflow = Medium.setState_phX(portA.p, inStream(
       portA.h_outflow), inStream(portA.X_outflow));
    T_in = Medium.temperature(stateA_inflow);
   else
    stateA_inflow = Medium.setState_pTX(portA.p, T_in, inStream(
       portA.X_outflow));
    inStream(portA.h_outflow) = Medium.specificEnthalpy(
```

```
        stateA_inflow);
  end if;
```