Thermodynamic Optimization of Organic Rankine Cycles for Co-Generation from Low Enthalpy Ultra-Deep Geothermal Reservoirs

R.F.O. van Wijk

January 31, 2019

Preface

This report was written commissioned by HermanDeGroot Ingenieurs en Vastgoedstrategen. I did this internship in the framework of my masters mechanical engineering at the University of Twente. The research was done during the period of september untill december of 2018. I very much enjoyed my experience at HermanDeGroot and would like to thank my supervisor Stijn Nefkens, for the advise he has given me in the past months. Furthermore, I would like thank Johan Herman de Groot, for making this internship possible. Lastly, I would like to thank all my colleagues at HermanDeGroot for the great atmosphere I have come to know at HermanDegroot

UNIVERSITY OF TWENTE.

Abstract

In this report an investigation was carried out into the applicability of an Organic Rankine Cycle electricity plant in combination with ultra deep geothermal energy. The first step was to determine the possible lay-out of the plants. Four different ORC systems were analyzed: A subcritical ORC, a subcritical regenerative ORC, a supercritical ORC and a regenerative supercritical ORC. From these four systems it was found that the regenerative supercritical system would lead to the largest electricity production. To analyze these systems, a detailed model of the ORC system was built in MatLab. This model optimized three thermodynamic parameters of the ORC system: The condensation temperature, evaporation pressure and the maximum temperature in the cycle. Furthermore, the model maximizes the working fluid mass flow, which is restricted by the pinch-point temperature difference in the heat exchangers and by the minimum outgoing temperature of the brine. A large selection of potential working fluids was analyzed. It was found that the properties governing the the maximum electricity output, were the working fluid critical temperature and the molar weight of the fluid. For the supercritical systems the molar weight appears to be the governing property determining the suitability of a working fluid. For a selection of suitable working fluids, the sensitivity of various parameters was mapped. n-Butane was selected as a working fluid and multiple concepts of co-generation plants were investigated. These concepts were all based on the subcritical non-regenerative design of the ORC. The combined electricity and heat production was analyzed for two different situations with different ingoing geothermal brine properties and different heat demands. A system consisting of two separate ORC cycles condensing at different temperatures was found to be the most effective in all situations. Furthermore, it was found that using a co-generation plant, delivering heat at high temperature approximately halves the produced electricity compared to the system with no heat production.

Keywords:
Organic Rankine cycle (ORC), Co-generation, Ultra deep geothermal energy, Thermodynamic Optimization, Sustainable energy production.

HermanDeGroot UNIVERSITY OF TWENTE.

Contents

HermanDeGroot

UNIVERSITY OF TWENTE.

Chapter 1

Introduction

In this chapter, the basic principles behind the working principles of an organic rankine cycle will be analyzed. Multiple adaptations to a basic ORC lay-out to improve the ORC performance are already known to at HermanDeGroot (Nguyen, 2017). A number of these improved configurations will be discussed in this chapter and finally, multiple concept lay-outs will be selected that will be analyzed and optimized using MatLab.

1.1 Motivation

Nowadays sustainability is becoming a major issue in the Netherlands. With the government aiming for new and innovative ways (Rijksoverheid, 2017) to reduce the production of CO_2, ultra deep geothermal energy might be a solution. The world wide energy consumption is growing (BP World Energy, 2018). This is another reason to explore new ways of electricity and heat generation. The Netherlands is also still dependant on natural gas, despite the fact that the gas production in Groningen is being halted (ENSOC, 2016). Renewable energy sources will play a big role in decreasing this dependence on natural gas and in supplying the Netherlands with it's increasing energy demand (Volkers and Wetzels, 2013). Ultra deep geothermal energy offers thermal and electrical energy with little environmental effects. The geothermal heat can either be used to generate heat for use in houses and industry or to generate electricity. The environmental policy of the government is reflected in the actions in the municipalities of the Netherlands. These municipalities have their own policies on increasing (local) sustainability in energy production (Zoeteman, 2014). HermanDeGroot has started a collaberation with the municipality of Soest to investigate the possibility of implementing a co-generation plant to supply the municipality with electricity and heat in a sustainable manner (Soester Courant, 2017).

1.2 Research objective

In this research, the thermodynamic properties of a co-generation plant are optimized to maximize the electricity production of the plant. The plant is designed to be applicable in the Netherlands. Previous research into the application of a geothermal energy plant was already done by HermanDeGroot (Nguyen, 2017). This research showed that the use of an organic rankine cycle is the most promising method of electricity generation. Based on previous findings of HermanDeGroot, this report aims to explore multiple ORC plant lay-outs, potential working fluids and methods of co-generation. First, an ORC is investigated using different working fluids and the electricity production is maximized. Then the ORC is redesigned to be suitable for co-generation.

Chapter 2

Basic ORC Principles

In this chapter, the basic principles and components of a simple ORC are identified.

When using geothermal heat at relatively low temperatures to produce electricity, the most common plant used is an organic Rankine cycle(DiPippo, 2016). An organic Rankine cycle works similar to a coal powered electricity plant, but it uses an organic working fluid rather than water, which is used in a normal Rankine cycle. Water is not feasible as a working fluid for this low temperature application due to economic reasons. Water would be able to achieve good efficiencies, however it requires a very complex turbine. This turbine would require a high peripheral speed, a high volumetric flow rate ratio and liquid content at the turbine discharge due to the inverse inverse relation between turbine enthalpy drop and molar mass, which is low for water.

Another difference is that in this case, no (fossil) fuels are burned to heat up the working fluid. Instead a geothermal well produces hot water which is used to vaporize the working fluid in a heat exchanger (or evaporator). A simplified representation of an ORC is shown below in figure 2.1.

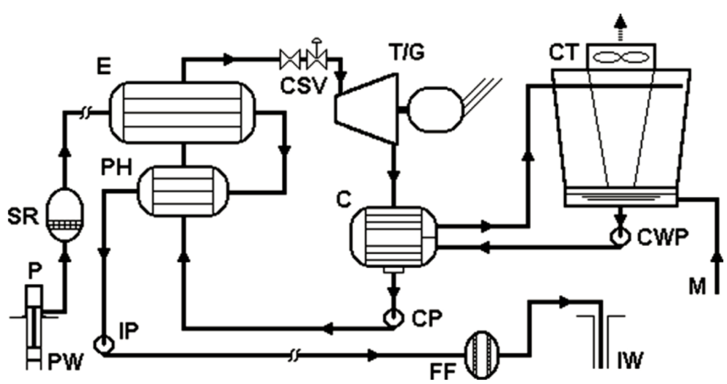

Figure 2.1: Simplified lay-out of a basic ORC binary plant (DiPippo, 2016)

In figure 2.1 three flows are recognized. The first flow is the flow of the geothermal water source. When it is pumped up to the ORC, the flow first goes through a sand remover (SR) which is done to prevent erosion of the piping. Then it continues through two heat exchangers. The first is called the evaporator (E) and the second is called the preheater (PH), here the geothermal fluid is used to evaporate and preheat the ORC working-fluid respectively. These heat exchangers are usually counterflow to increase the efficiency. After this the geofluid is injected back into the ground.

The second flow is that of the organic working fluid. This is a closed loop flow in the ORC. It is pumped to a high pressure at CP, after which it is preheated and evaporated by the hot geothermal fluid. In some cases it is also beneficial to superheat the vapour after it is evaporated. While this is common practice when using a "wet" fluid like water, it is not necessarily required for dry fluids which are often used in organic Rankine cycles. The hot gas is expanded in a turbine which powers a generator (T/G). After the generator the working fluid goes through a condenser (C) where it becomes liquid again. After the condenser the cycle starts anew.

The last flow shown in figure 2.1, is that of a cooling fluid through the condenser. After this fluid heats up in the condenser it cools down in a cooling tower(CT). Once cooled it is pumped (CWP) back to the condenser. The ORC system given in 2.1 uses a cooling tower, but other methods of condensation are also used. For example, the condensation can be achieved by using cooling water from a river or another water source directly or, which will be assumed in this report, by using an air-cooled condenser. This type of condenser uses fans to blow (cold) air past rows of finned tubes containing the working fluid vapour. Due to the heat exchanging between the air and the working fluid vapour inside the tubes, the working fluid will condense.

This report will focus on the thermodynamic performance of the ORC system. Therefore, components like the sand remover will not be analyzed. The pumps required to extract and reinject the geothermal fluid ('brine'), are also not analyzed. The thermodynamic properties of the fluids in the ORC system that are required are: The temperature, the pressure, the specific enthalpy and the entropy. When two of these properties are known, the other two can always be determined. The thermodynamic behaviour of a basic ORC is represented by the T-s and P-h diagram. These are typically similar to figure 2.2.

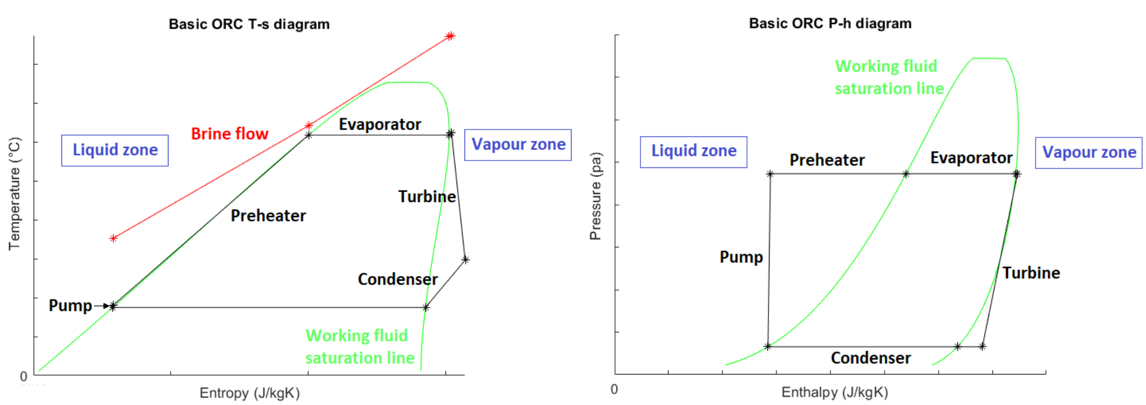

Figure 2.2: Temperature-Entropy and Pressure-Enthalpy diagram for a basic ORC lay-out

2.1 Concept lay-out selection

The basic ORC can be improved by changing the ORC lay-out or by changing the operating conditions. There are already various alternative ORC lay-outs known that can improve the efficiency of the ORC. For a detailed description of the working principles of the considered systems, refer to appendix A. Here, a literature study was done into existing techniques to improve the ORC performance.

Multiple lay-outs of the ORC plant are considered. A selection for multiple ORC lay-out concepts was made based on the findings in appendix A. The selected concepts are listed and explained below.

1. **Subcritical non-regenerative single pressure ('basic') ORC**
   This system was chosen because it is the most basic form of the organic Rankine cycle. It is a

good starting point for the MatLab model from where to add components that might improve it's performance and it will serve as a reference for the other concepts to quantify the performance increase as a consequence of the altered ORC lay-out.

2. **Subcritical regenerative single pressure ORC**
   Adding a regenerator to the ORC cycle is expected to increase the ORC efficiency. There are little downsides to the added heat exchanger, yet the degree in which the regenerator might increase the efficiency is not known. Therefore, this system is chosen to be analyzed and optimized.

3. **Supercritical non-regenerative single pressure ORC**
   To minimize the exergy loss in the heat exchangers, the temperature difference between the two flows in the heat exchanger needs to be minimized. Two methods were proposed to do this: using a dual-pressure system or using a supercritical system. A supercritical system was chosen, this choice was made based on the complete absence of an evaporation process. It is hypothesized that the supercritical system will be more beneficial for the efficiency of the system than the dual-pressure system.

4. **Supercritical regenerative single pressure ORC**
   The last system that will be analyzed uses both a supercritical fluid and a regenerator. It is expected that this system will perform best of all the proposed systems, since it uses the advantages of all proposed systems.

Since all systems operate using a single pressure system, this will no longer be mentioned when a reference is made to the concept lay-outs.

Chapter 3

List of Requirements

With the basic principles of an ORC known. A list of requirements can be made for the design of the ORC. There are four sets of requirements, one for the geothermal fluid, one for the ORC plant, one for the working fluid and one for co-generation of heat and electricity.

3.1 Requirements for the geothermal fluid

1. **Geofluid must be kept at a pressure above its flash point.**
   Since the geothermal fluid enters the ORC system at a high temperature, the pressure of the geothermal fluid inside the system should be high enough so that the fluid will not vaporize.

2. **Geofluid reinjection is 70 °C or higher.**
   The heat of the geofluid is used to power the ORC, so it is logical to extract as much heat as possible from this fluid. However, to avoid silica scaling and other negative effects, there is a minimum temperature at which the geofluid can be pumped back into the earth. For comparable situations, reinjection temperatures vary between 50 and 100 °C (Diaz et al., 2015). Another problem that might occur due to a low reinjection temperature is thermal stress around the point of reinjection (Vis, 2018). So a minimum reinjection temperature of 70 degrees is assumed for safety.

3.2 Requirements for the ORC plant

1. **The minimum condensation temperature of the cycle is 35 °C**
   Since no large amounts of water will be available on site (Nguyen, 2017), an air cooled condenser will be used in the system. The air cooled condenser should be usable even on the hottest day of the year, therefore a minimum condensation temperature of 35 °C is assumed. This is quite high compared to the average temperature in the Netherlands.

2. **Maximum pressure in the system should not exceed 11 MPa**
   The maximum system pressure will increase the cost of the plant. The maximum pressure of this system is based on the maximum working pressure of turbines in the MW-range (Siemens, 2014).

3. **The ORC system should be economically feasible**
   One major issue when designing something is always cost. The ORC plant should be lucrative. Therefore, the total investment of the ORC, including the construction of the geothermal well, should not be more than the proceeds of the electricity. However, the exact cost if the ORC plant is beyond the scope of this project. It can therefore, not be determined if the designed installations meet this requirement.

3.3 Requirements for the ORC working fluid

1. **The working fluid should have 0 ozone layer depletion (ODP)**
   According to the montreal protocol, these types of fluids must be phased out completely by 2030, because these fluids have a high ozone depletion potential (ODP) (Latrash et al., 2015). Since the plant is expected to be operating long after this time, the working fluid used can not be a hydro hlorofluorocarbon (HCFC).

2. **The working fluid should have a global warming potential (GWP) less than 2500**
   Due to the F-gas regulation in the European union, refrigerants with a GWP higher than 2500 are not allowed in the ORC system (PORKKA, 2018). Since a lower GWP is better for the environment, a GWP much lower than 2500 is preferred.

3. **The working fluid is desired to be non-flammable and non-toxic**
   The ORC plant should be as safe as possible. This means that a non-flammable non-toxic working fluid would be preferable. However, as long as the risks are manageable, this should not stand in the way of increasing the performance of the ORCe.

4. **The ORC working fluid should be inexpensive**
   For obvious reasons, decreasing the cost of the working fluid is desirable. As long as a more expensive working fluid does not lead to relevant increases in safety or performance of the ORC, inexpensive working fluids are preferred.

3.4 Requirements for co-generation

1. **Heat generated must be at least 39 MWatt**
   The ORC is designed to generate power in the Netherlands. However, the geothermal well and ORC waste heat will also be used to supply heat to a heating network. The amount of heat that needs to be supplied is based on the municipality of Soest and is approximately equal to 39MW on average during per year. (ten Voorde, 2018)

2. **Heat supplied to heating network should be 90°C**
   The heating network suited for the municipality of Soest works with water at a temperature of 90°C. Low-temperature heating networks do exist (van Vliet et al., 2016), but are not suited for this specific municipality, because this would mean a lot of buildings need better isolation and this would result in high costs (Smoor, 2017).

Chapter 4

Optimization of ORC Cycles

In this section the thermodynamic optimization process of various ORC systems will be analyzed. At the end of this section, the optimization protocols for these systems should have become clear. Using the thermodynamic optimization a working fluid will be selected in section 5. When a working fluid is selected, the maximum possible output of each of the ORC systems can be found.

4.1 Macroscopic thermodynamic optimization

Before the specific model is explained, the goal of the optimization should be clear. The goal of the MatLab optimization is to maximize the work output from the turbine based on a set geothermal flow. To do this some operating conditions of the organic rankine cycle can be "tuned". The optimizable operating conditions of the thermodynamic model are the same for all proposed ORC lay-outs. They are: the maximum pressure occurring in the cycle, the maximum temperature in the system and the condensation pressure of the system.

The aim of the MatLab model was to be able to quickly compare different ORC lay-outs and to find the optimal operating conditions for a chosen ORC lay-out for multiple working fluids in one simulation. This way, for a selection of working fluids, the best working fluid can be identified quickly.

When the optimization is run, the aim is to be able to quickly asses suitability of a certain working fluid and the effect of the given input conditions. This is done graphically through the generation of the T-s and P-h diagrams of the analyzed system(s) and by means of a table generated by the thermodynamic model. This table shows all critical information about the ORC system, such as the final values of the optimized variables, the net work of the system, parasitic electricity consumption of the condenser and pump and the thermal efficiency of the system. Lastly, a structure is created containing all thermodynamic information about each point in the ORC cycle.

4.2 Optimization of a basic subcritical ORC

First, a basic ORC will be optimized. The most simplified lay-out of a basic ORC is shown in figure 4.1. The important locations in the cycle are numbered.

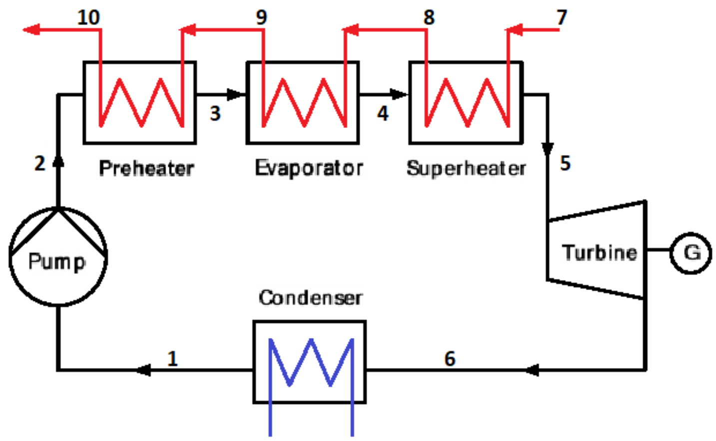

Figure 4.1: Simplified lay-out of a basic ORC binary plant (Kim, 2013)

4.2.1 Thermodynamic analysis

For each point in the cycle shown in figure 4.1, at least two variables are known:

Point	Fluid property 1	Fluid property 2
1	Saturated liquid	$P1 = $ t.b.d.
2s	$S2s = S1$	$P2s = $ t.b.d.
2	$H2 = H1 + \frac{(H2s - H1)}{\eta_{s,p}}$	$P2 = P2s$
3	Saturated liquid	$P3 = P2$
4	Saturated vapour	$P4 = P3$
5	$T5 = $ t.b.d.	$P5 = P4$
6s	$S6s = S5$	$P6s = P1$
6	$H6 = H5 - \eta_{s,t}(H5 - H6s)$	$P6 = P1$
7	$T7 = T_{b,in}$	$P7 = P_b$
8	$H8 = \frac{\dot{m}_b \cdot H7 - \dot{m}_{wf}(H5 - H4)}{\dot{m}_b}$	$P8 = P7$
9	$H9 = \frac{\dot{m}_b \cdot H7 - \dot{m}_{wf}(H5 - H3)}{\dot{m}_b}$	$P9 = P8$
10	$H10 = \frac{\dot{m}_b \cdot H7 - \dot{m}_{wf}(H5 - H2)}{\dot{m}_b}$	$P10 = P9$

Table 4.1: known thermodynamic properties for each point in the basic subcritical ORC

From this table it can be seen that for all but three of the ORC points, at least two variables are known. The missing parameters are: P1, P2(s) and T5. These are the parameters that will be optimized. However, these parameters also have some boundary conditions. These are:

1. P1: The lowest pressure in the cycle
   (a) P1 can't be lower than atmospheric pressure (10^5 Pa) to prevent in-leakage of ambient air.(Guillen and Zia, 2013)
   (b) The corresponding saturated liquid temperature can't be lower than 35°C, because it is cooled with ambient air.
2. P2: The highest pressure in the cycle
   (a) P2 can't be higher than the critical pressure of the working fluid.

(b) P2 is always higher than P1.

3. T5: The highest temperature in the cycle (superheated vapour).

   (a) T5 can't be higher than the brine ingoing temperature T7.
   (b) T5 is always higher than or equal to T4.
   (c) The chosen T5 can not lead to a wet situation after the turbine.

An overview of all boundary conditions for this system can be found in appendix B.

4.2.2 Mass flow

To determine the mass flow of this system, it is important to know the limiting factor. The mass flow is limited by a pinch point in the heat exchanger or by the minimum outgoing temperature of the brine. For normal subcritical ORCs a pinch point can occur at two different locations. A T-s diagram of a basic subcritical ORC is shown in figure 4.2. In figure 4.2a, the situation is shown if the mass flow is limited by the outgoing temperature. In figure 4.2.b, the mass flow is limited by a pinch point on the cold side of the preheater. In figure 4.2c the mass flow is limited by a pinch point on the cold side of the evaporator. The situation in figure 4.2d is likely to occur when the pressure in the heat exchangers is close to the critical pressure of the working fluid. When this happens, the isobaric lines can no longer be assumed to be linear. This can result in a pinch point at an unknown location in the preheater.

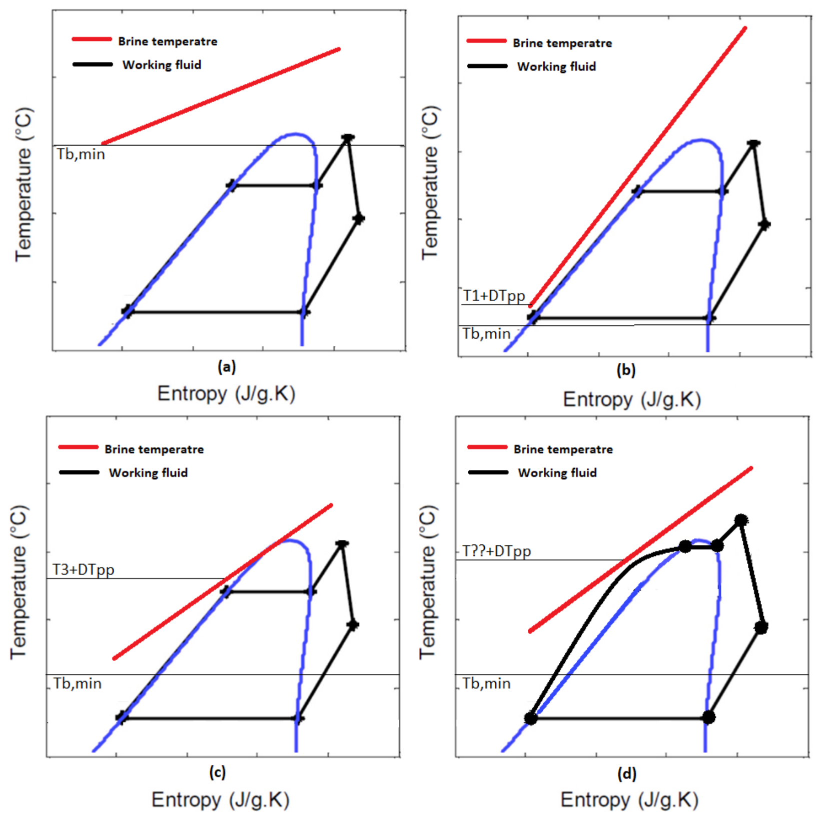

Figure 4.2: (a) no pinch point, mass flow limited by minimum temperature $T_{b,min}$. (b) Pinch point at cold end preheater. (c) pinch point located at hot end preheater (d) pinch point located before hot end preheater

For the situation in figure 4.2a, the mass flow can be computed easily, by using the energy balance for point 10 in table 4.1. In this case it is assumed that the outgoing temperature of the brine is equal to a given minimum temperature of the brine $T_{b,min}$. When the brine pressure and the brine temperature at point 10 are known, the brine enthalpy is also known. The maximum mass flow is found to be:

$$\dot{m}_{wf,a} = \frac{\dot{m}_b(H7 - H10)}{H5 - H2} \tag{4.1}$$

The parameters P2s, P1 and T5 are the parameters which will be optimized to find the highest possible output power. These are assumed to be known.

In the cases of figure 4.2a, 4.2b and 4.2c, the mass flow is limited by a minimum temperature difference between the brine and the working fluid: ΔT_{pp}. Because the location of the pinch point in case of figure 4.2d is not known. It is assumed that the pinch point can lie anywhere between the cold side and the hot side of the preheater. By doing this all possible situations from figure 4.2 are taken into account.

If the mass flow can be computed, assuming that the pinch point is at any random location between the cold and hot side of the preheater, the outgoing temperature of the brine can also be determined. For the system to work, the brine temperature at this point in the heat exchanger should at least be T_{pp} higher than the working fluid temperature at the pinch point temperature of the working fluid which will be called T_X. This means: The maximum mass flow in case of a pinch point at working fluid temperature T_X is determined by using an energy balance:

$$\dot{m}_{wf,pp@X} = \frac{\dot{m}_b(H7 - H_{brine@X})}{H5 - H_{wf@X}} \tag{4.2}$$

With the mass flow of the working fluid known, the enthalpy of the outgoing brine flow can also be determined:

$$H_{b,out,pp@X} = \frac{\dot{m}_{wf,pp@X} \cdot (H5 - H2) - (\dot{m}_b \cdot H7)}{\dot{m}_b} \tag{4.3}$$

All enthalpies in equations (4.1), (4.2) and (4.3) are functions of knowns, assumptions or optimized variables:

$S1 = f(P1, \text{saturated liquid})$
$H1 = f(P1, \text{saturated liquid})$
$H2 = H1 + (H2s - H1)/\eta_{s,p}$
$H2s = f(S1, P2)$
$H5 = f(T5, P2)$

$H7 = f(T_{b,in}, P_b)$
$H10 = f(T_{b,min}, P_b)$
$H_{brine@X} = f(T_X + \Delta T_{pp}, P_b)$
$H_{wf@X} = f(T_X, P2)$

With:

$$T2 < T_X < T3$$

When the resulting outgoing enthalpy is higher than the minimum outgoing enthalpy of the brine: $H_{b,out,min} = f(T_{b,min}, P_b)$, There is actually a pinch point for the system. By using an optimization function (fmincon) in MatLab. The maximum value for $H_{b,out,pp@X}$ can be found out of all possible values for T_X, If the highest found value for $H_{b,out,pp@X}$ is higher than $H_{b,out,min}$, this means that the actual maximum mass flow is equal to the mass flow corresponding to the maximum value of $H_{b,out,pp@X}$. If the maximum value of $H_{b,out,pp@X}$ is lower than the $H_{b,out,min}$, the mass flow is not limited by a pinch point. Instead, it is limited by the set minimum outgoing brine temperature, $T_{b,min}$. In this case the mass flow is determined using equation (4.1).

4.2.3 Output power

with the mass flow and the enthalpies for all points known, the work done by the system can be determined. The work done by the system is dependant on the work done by the turbine and the power required to operate the pump and the condenser fans as well as the mechanical efficiency of the system and of the generator. The net work is computed as follows:

$$\dot{W}_{Netto} = \left(\dot{W}_{turbine} - \dot{W}_{pump} - \dot{W}_{condenser}\right) \cdot \eta_{mechanical} \cdot \eta_{generator} \tag{4.4}$$

The work for each component is calculated using the the equations:

$$\dot{W}_{turbine} = \dot{m}_{wf}(H5 - H6)$$

$$\dot{W}_{pump} = \frac{\dot{m}_{wf}(P2 - P1)}{\eta_{pump} \cdot \rho_{water}} \tag{4.5}$$

$$\dot{W}_{condenser} = \dot{v}_{air} \cdot \frac{\Delta P_{fan}}{\eta_{fan}} \qquad (Woodlandet\ al.,\ 2014)$$

with:

$$\dot{v} = \frac{\dot{m}_{air}}{\rho_{air}} = \frac{\dot{m}_{wf}\,(H6 - H1)}{\rho_{air}\,(H_{air,out} - H_{air,in})} \tag{4.6}$$

and:

$\Delta P_{fan} = 120$ Pa (Yang et al., 2004)

All enthalpies in equations (4.5) and (4.6) are functions of knowns, assumptions or optimized variables:

$H1 = f(P1, \text{saturated liquid})$
$H5 = f(T5, P2)$
$S5 = f(T5, P2)$
$H6 = H5 - \eta_{s,t}(H5 - H6s)$
$H6s = f(S5, P1)$
$H_{air,in} = f(T_{air,in}, P_\infty)$
$H_{air,out} = f(T_{air,out}, P_\infty)$

It is assumed that the outgoing temperature of the air leaving the condenser is known. The ingoing temperature of the condenser air is equal to the temperature of the environment, T_∞. The pressure of the air remains constant at P_∞. With the temperatures and pressures of both the in- and outgoing air flows of the condenser known, the corresponding enthalpies are also known.

4.3 Optimization of a regenerative subcritical ORC

Next, the lay-out of the system will be changed to achieve a more efficient system. From the literature research it become apparent that the addition of a regenerator in the system has no downsides, except for the increased cost of the system. A regenerator is added to see if this leads to a significant increase in the performance of the system. The system lay-out and the important points are shown in figure 4.3.

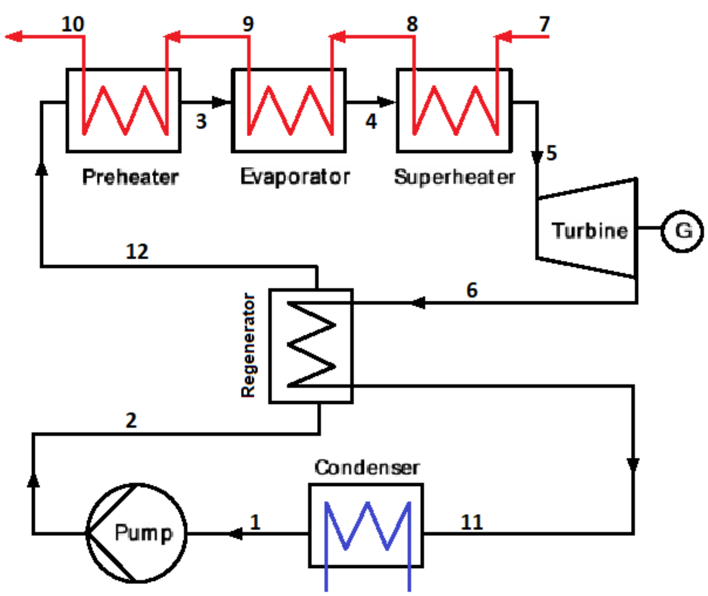

Figure 4.3: Simplified lay-out of a regenerative ORC binary plant

4.3.1 Thermodynamic analysis

For each point in the cycle shown in figure 4.3, at least two variables should be known. If this is the case, all information for each point can be determined. All points and known variables are shown in table 4.2.

Point	Fluid Property 1	Fluid Property 2
1	Saturated liquid	$P1 = \text{t.b.d.}$
2s	$S2s = S1$	$P2s = \text{t.b.d.}$
2	$H2 = H1 + \dfrac{(H2s - H1)}{\eta_{s,p}}$	$P2 = P2s$
3	Saturated liquid	$P3 = P2$
4	Saturated vapour	$P4 = P3$
5	$T5 = \text{t.b.d.}$	$P5 = P4$
6s	$S6s = S5$	$P6s = P1$
6	$H6 = H5 - \eta_{s,t}(H5 - H6s)$	$P6 = P1$
7	$T7 = T_{b,in}$	$P7 = P_b$
8	$H8 = \dfrac{\dot{m}_b \cdot H7 - \dot{m}_{wf}(H5 - H4)}{\dot{m}_b}$	$P8 = P7$
9	$H9 = \dfrac{\dot{m}_b \cdot H7 - \dot{m}_{wf}(H5 - H3)}{\dot{m}_b}$	$P9 = P8$
10	$H10 = \dfrac{\dot{m}_b \cdot H7 - \dot{m}_{wf}(H5 - H2)}{\dot{m}_b}$	$P10 = P9$
11	$T11 = T1 + T_{pp}$	$P11 = P1$
12	$H12 = H2 + (H6 - H11)$	$P12 = P2$

Table 4.2: known thermodynamic properties for each point in the regenerative subcritical ORC

Table 4.2 is largely the same as Table 4.1 for the basic ORC. The only difference is the addition of two extra points of interest, these do not influence the rest of the thermodynamic properties and boundary conditions. These are the same as discussed in the section 4.2.1. The difference between the two systems can be seen when we add the brine temperature line to the T-s diagram of this system. This line now starts above point 5 and ends above point 12. This can be seen in figure 4.4 Which is different from the brine line in figure 4.2.

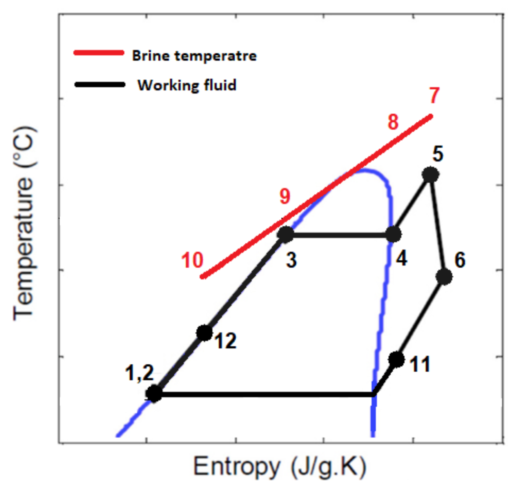

Figure 4.4: Ts-diagram including brine line for a regenerative subcritical ORC

Furthermore, there is one more boundary condition compared to the basic ORC. This boundary condition is one on point 12:

1. T12: The vapour temperature at the cold side of the regenerator
   (a) T12 can't be lower than T1+T_{pp}, because there must always be a minimum temperature difference of T_{pp} between the two fluids in the heat exchanger. It is assumed that the difference in c_p values for the two fluids in the regenerator is negligible.

An overview of all boundary conditions for this system can be found in appendix B.

4.3.2 Mass flow

The maximum mass flow for this system is determined in the same manner as for a basic ORC in section 4.2.2. However, the working fluid has a higher temperature at the cold side of the preheater. This is due to the regeneration process. This means that the equations for the mass flow change. First of all, in the case of no pinch point, the basic ORC mass flow was determined using equation (4.1). This equation now becomes:

$$\dot{m}_{wf,a} = \frac{\dot{m}_b(H7 - H10)}{H5 - H12} \tag{4.7}$$

In case of a mass flow limitation by a pinch point the procedure is the same as for a basic ORC. The pinch point can still only occur in the preheater. To determine the maximum maximum mass flow with the pinch point occurring at temperature T_X, equation (4.2) is still used. To determine the outgoing brine enthalpy, equation (4.3) must be altered slightly to the form:

$$H_{b,out,pp@X} = \frac{\dot{m}_{wf,pp@X} \cdot (H5 - H12) - (\dot{m}_b \cdot H7)}{\dot{m}_b} \tag{4.8}$$

where all enthalpies in equations (4.7) and (4.8) are functions of knowns, assumptions or optimized variables:

$H1 = f(P1, \text{saturated liquid})$
$S1 = f(P1, \text{saturated liquid})$
$H2 = H1 + (H2s - H1)/\eta_{s,p}$
$H2s = f(S1, P2)$
$H5 = f(T5, P2)$
$S5 = f(T5, P2)$
$H6 = H5 - \eta_{s,t}(H5 - H6s)$
$H6s = f(S5, P1)$
$H7 = f(T_{b,in}, P_b)$
$H10 = f(T_{b,min}, P_b)$
$H11 = f(T1 + T_{pp}, P1)$
$H12 = H2 + (H6 - H11)$

4.3.3 Output power

The output power of the system is also determined in the same way as for the basic ORC with one small difference: The power required by the fan in the condenser will be less, since the vapour leaving the turbine is already cooled in the regenerator. This means that equations of (4.5) still hold, but equation (4.6) changes slightly to the form:

$$\dot{v} = \frac{\dot{m}_{air}}{\rho_{air}} = \frac{\dot{m}_{wf}\,(H11 - H1)}{\rho_{air}\,(H_{air,out} - H_{air,in})} \tag{4.9}$$

All enthalpies in equation (4.9) are functions of knowns, assumptions or optimized variables:

$H1 = f(P1, \text{saturated liquid})$
$H11 = f(T1 + T_{pp}, P1)$
$H_{air,in} = f(T_{air,in}, P_\infty)$
$H_{air,out} = f(T_{air,out}, P_\infty)$

4.4 Optimization of a supercritical ORC

The lay-out of the supercritical system is not very different from the lay-out of a basic subcritical cycle, the only difference is that is has no separate preheater, evaporator and superheater. Instead, it has one supercritical heat exchanger. A schematic lay-out with all important points is given in figure 4.5. The numbers corresponding to each component are kept mostly the same (e.g. 2 still corresponds the fluid properties after the pump and 6 still corresponds to the fluid properties after the turbine). This way most of the equations used in the optimization of the basic subcritical ORC are also valid for the supercritical case.

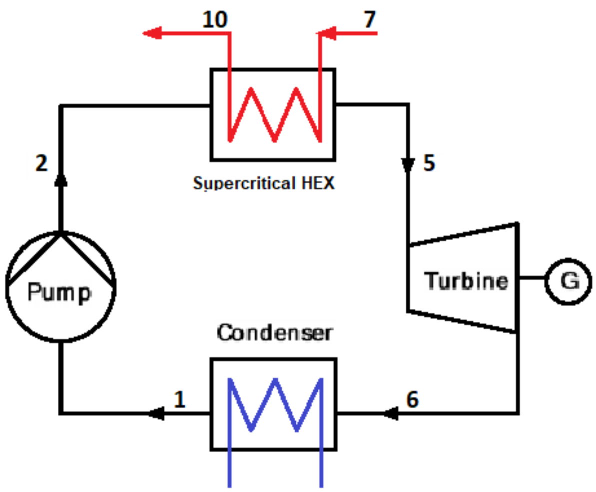

Figure 4.5: Schematic lay-out of a supercritical ORC

4.4.1 Thermodynamic analysis

For each point in the cycle shown in figure 4.3, at least two variables should be known. If this is the case, all information for each point can be determined. All points and known variables are shown in table 4.3.

Point	Fluid Property 1	Fluid Property 2
1	Saturated liquid	$P1 = \text{t.b.d.}$
2s	$S2s = S1$	$P2s = \text{t.b.d.}$
2	$H2 = H1 + \dfrac{(H2s - H1)}{\eta_{s,p}}$	$P2 = P2s$
5	$T5 = \text{t.b.d.}$	$P5 = P2$
6s	$S6s = S5$	$P6s = P1$
6	$H6 = H5 - \eta_{s,t}(H5 - H6s)$	$P6 = P1$
7	$T7 = T_{b,in}$	$P7 = P_b$
10	$H10 = \dfrac{\dot{m}_b \cdot H7 - \dot{m}_{wf}(H5 - H2)}{\dot{m}_b}$	$P10 = P7$

Table 4.3: known thermodynamic properties for each point in the supercritical ORC

Table 4.3 gives at least two fluid properties for all important points of the supercritical ORC. This means that all thermodynamic properties of the system can be determined. Again, P1, P2 and T5 are the parameters that will be optimized and these are considered to be known. The Ts-diagram for the supercritical ORC can be seen in figure 4.6

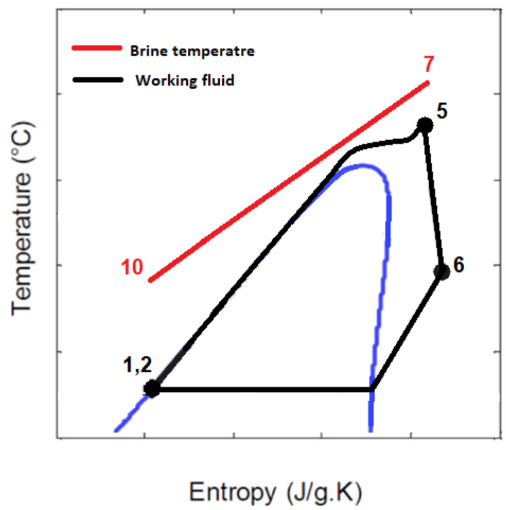

Figure 4.6: Ts-diagram of a supercritical ORC with the location of the important points.

The boundary conditions for this system are the same as those for the basic subcritical ORC, with one exception: The boundary conditions on the maximum value of P2 (boundary condition 2.a in section 4.2.1) is not valid since the system is supercritical. The system has a different boundary condition instead. The temperature after the heat exchanger also has a boundary condition to make sure the system operates in the supercritical region. These two boundary conditions are respectively:

1. P2: The vapour temperature at the cold side of the regenerator
   (a) P2 can't be lower than the critical pressure of the working fluid that is used. The system will not be supercritical if this would be the case.
2. T5: The highest temperature in the cycle
   (a) T5 can't be lower than the critical temperature of the working fluid that is used. The system will not be supercritical if this would be the case.

An overview of all boundary conditions for this system can be found in appendix B.

4.4.2 Mass flow

Again, the the mass flow for this system is determined with the method discussed in 4.2.2. For this system there are no changes in the equations used to determine the maximum possible mass flow. In section 4.3.2 it was stated that the pinch point occurs between the cold and hot side of the preheater. Since there is no preheater the region where the pinch point might occur must be redefined. Looking at the Ts-diagram in figure 4.6 it is assumed that the pinch point can occur anywhere inside the supercritical heat exchanger. This means that the domain of T_X changes, with the equations staying the same. The new domain for T_X is:

$$T2 < T_X < T5$$

4.4.3 Output power

The output power of the supercritical system is computed in the same way as in 4.2.3

4.5 Optimization of a regenerative supercritical ORC

The third system that will be analyzed is a regenerative supercritical system. This system is in all ways a combination of the the regenerative subcritical and the supercritical system. This system also has no separate preheater, evaporator and superheater, but a single supercritical heat exchanger. Furthermore, this system also has an extra heat exchanger between the working fluid after the turbine and the working fluid after the pump. A schematic lay-out with all important points is given in figure 4.7. Again, the numbers corresponding to each component are kept mostly the same.

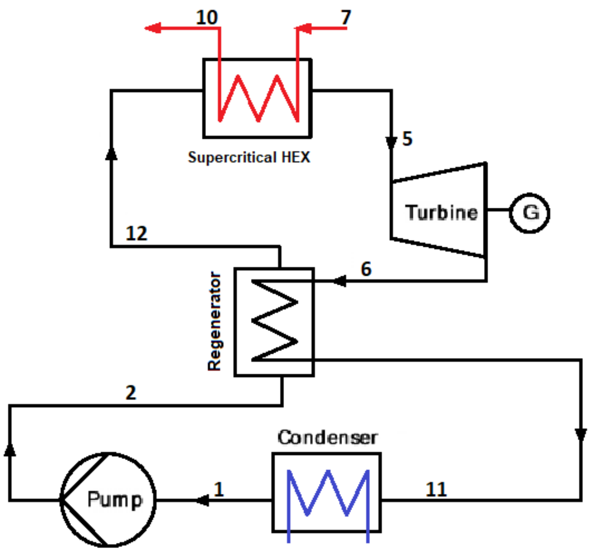

Figure 4.7: Schematic lay-out of a supercritical regenerative ORC

4.5.1 Thermodynamic analysis

For each point in the cycle shown in figure 4.7, at least two variables should be known. If this is the case, all information for each point can be determined. All points and known variables are shown in table 4.4.

Point	Fluid Property 1	Fluid Property 2
1	Saturated liquid	$P1 = $ t.b.d.
2s	$S2s = S1$	$P2s = $ t.b.d.
2	$H2 = H1 + \dfrac{(H2s - H1)}{\eta_{s,p}}$	$P2 = P2s$
5	$T5 = $ t.b.d.	$P5 = P2$
6s	$S6s = S5$	$P6s = P1$
6	$H6 = H5 - \eta_{s,t}(H5 - H6s)$	$P6 = P1$
7	$T7 = T_{b,in}$	$P7 = P_b$
10	$H10 = \dfrac{\dot{m}_b \cdot H7 - \dot{m}_{wf}(H5 - H2)}{\dot{m}_b}$	$P10 = P7$
11	$T11 = T1 + T_{pp}$	$P11 = P1$
12	$H12 = H2 + (H6 - H11)$	$P12 = P2$

Table 4.4: known thermodynamic properties for each point in the regenerative supercritical ORC

Table 4.3 gives at least two fluid properties for all important points of the regenerative supercritical ORC. This means that all thermodynamic properties of the system can be determined. Again, P1, P2 and T5 are the parameters that will be optimized and these are considered to be known. The Ts-diagram for the regenerative supercritical ORC can be seen in figure 4.8

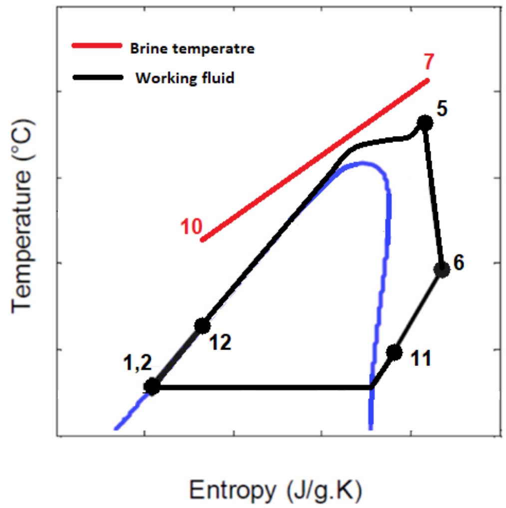

Figure 4.8: Ts-diagram of a supercritical ORC with the location of the important points.

This systems boundary conditions are equal to the boundary conditions of the (non-regenerative) supercritical ORC in section 4.4.1. With the addition of the boundary condition on point 12 from the regenerative ORC in section 4.3.1 An overview of all boundary conditions for this system can be found in appendix B.

4.5.2 Mass flow

The mass flow for this system is determined again using the same procedure as for the regenerative subcritical system in section 4.3.2. To determine the maximum mass flow again it must considered that the pinch point can occur anywhere in the supercritical heat exchanger. This is again due to the absence of a separate evaporator, preheater and superheater. This means: using the method from 4.3.2 the domain for T_X from section 4.4.2 is used. Which is:

$$T2 < T_X < T5$$

4.5.3 Output power

The output power for the regenerative supercritical system is calculated in the same way as for the regenerative subcritical system in section 4.3.3.

4.6 Optimization algorithm

As mentioned before, the parameters that the system will be optimized for are P1, P2 and T5. A model of all four systems was made in MatLab to calculate the net work output for all four binary systems. The net work is the parameter that was chosen to be optimized because the output of the power plant should be as high as possible. The fluid properties were determined in MatLab using CoolProp. CoolProp is an open-source database of fluid and humid air properties, formulated based on the most accurate formulations in open literature. It has been validated against the most accurate data available from the relevant references.
The optimization function offered by MatLab that was used to optimize the system is fmincon, which was mentioned before. fmincon is an optimization algorithm that finds the minimum value of a given

function. Since the net work should have a maximum value the net work is multiplied by -1, to change the required maximum into a minimum.

One disadvantage of fmincon is that it often finds a local minimum rather than the global minimum. To make sure the result is the global minimum the MatLab function globalsearch is used. This function uses fmincon, but starts the optimization procedure multiple times from different trial points, thus finding the global minimum.

The MatLab code for the optimization can be found here, or by following the URL in appendix G.1

4.7 Mass flow as an optimizable paramater

Because the power required to run the pump and the condenser fans is also linearly dependent on the mass flow of the working fluid, it is conceivable that the the maximum net power of the system is not necessarily at the highest possible mass flow rate. To exclude this possibility, the mass flow is also modelled as an optimization parameter in the MatLab function fmincon. It was still limited to the maximum value set by the pinch point or the minimum outgoing brine temperature. To avoid an outcome that is a local maximum instead of a global maximum, the matlab function globalsearch was used. However, this function greatly increased computation time. Therefore the simulation was only done for the three working fluids with the best performances. From this analysis it became apparent, that the maximum possible mass flow, yields the highest net work

Chapter 5

Working Fluids

With the functionality of the basic ORC identified, research will be done into the selection of a working fluid. In this section a large number of potential working fluids will be selected and their physical properties will be examined to see if a relation can be found between their performance in the ORC cycle and their physical properties. After an analysis of a large amount of different working fluids, a handfull will be selected. These working fluids will be used to determine the sensitivity of the system to parameters other than the working fluid.

5.1 Assumptions

To be able to fully define the MatLab model, some assumptions must be made. The assumptions done are listed below.

1. Brine mass flow is 50 kg/s
2. Brine ingoing temperature 200 °C
3. Turbine isentropic efficiency is 80% (Vanslambrouck et al., 2011)
4. Pump isentropic efficiency is 80 % (Borsukiewicz, 2013)
5. The mechanical efficiency of the generator is 98% (Radovic, 1997)
6. The pressure loss in the heat exchangers is negligible
7. The minimum pinch point temperature difference in the heat exchangers is 5 °C (Lisheng and Weixiu, 2016)
8. Environmental temperature is 10 °C
9. The environmental pressure is 10^5 Pa
10. The geothermal fluid properties are equal to those of water

5.2 Working fluid behaviour

In the first analysis, demands on the environmental and potentially hazardous properties of the working fluid will be neglected. From this analysis, an estimate can be made about the suitability for the ORC system of working fluids outside of the CoolProp directory. All trial working fluids and their properties can be found in appendix C. For all trial working fluids the optimization algorithm was run to find the maximum possible work output. This was done for four different ORC systems: the basic ORC, regenerative ORC, supercritical ORC and regenerative supercritical ORC. The results can be found in appendix D.

5.2.1 Working fluid behaviour in a subcritical ORC

From the results it was found that the critical temperature and the molar mass are the dominant fluid properties determining the maximum work output for the basic and regenerative subcritical ORC's. To find out in which way the work is dependent on these parameters a 3D plot was made. In this plot the critical temperatures and molar mass of all working fluids are plotted vs the maximum possible work. this was turned into a surface to clearly show the correlation between the parameters. The results can be seen in figures 5.1a, 5.1b.

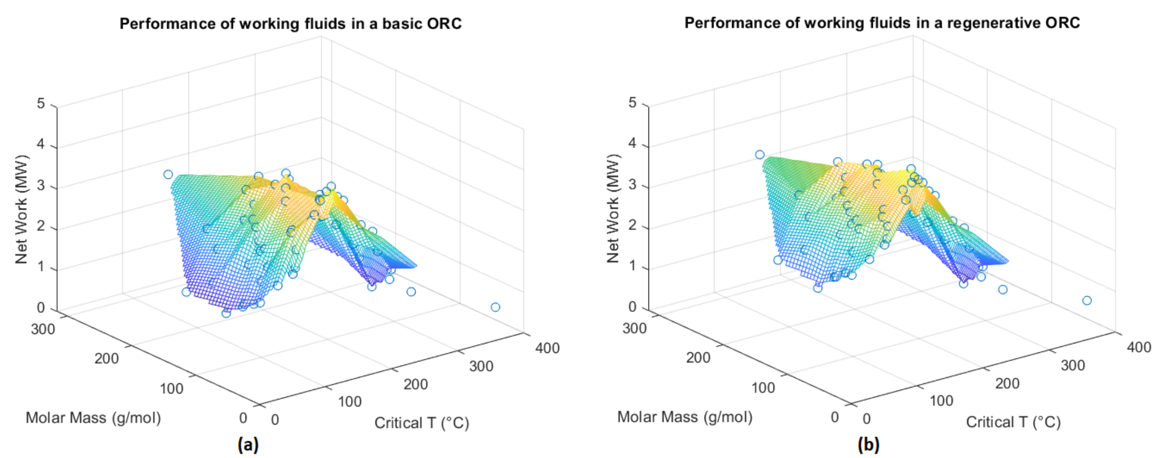

Figure 5.1: Net work output versus working fluid critical temperature and molar mass for a (a) basic subcritical ORC (b) regenerative subcritical ORC.

From figure 5.1 it can be noted that adding a recuperative system to the ORC does not change the suitability of a working fluid very much. It does however, increase the performance of the system significantly. For the subcritical systems in 5.1a and 5.1b, a peak in performance can be seen at a critical temperature of approximately 435 K. It can be seen that for the subcritical system, the critical temperature is the dominant factor determining the work output. From the figure, note that increasing the molar mass leads to a decrease in net power. However, this effect is much smaller than the effect of the critical temperature. Since there is no direct thermodynamic explanation for this last behaviour, consider it a purely statistical relation.

The behaviour of the found peak is investigated further. The location of the peak is likely to be dependent on the in- and outgoing temperatures of the brine. The optimization was run again with a varying brine ingoing temperature and constant outgoing temperature (of 70 °C) and with a varying outgoing temperature and constant ingoing temperature (of 200 °C). The approximate optimal critical temperatures are plotted versus the ingoing temperature 5.2a and versus the outgoing temperature 5.2b. This was done for the basic ORC system only.

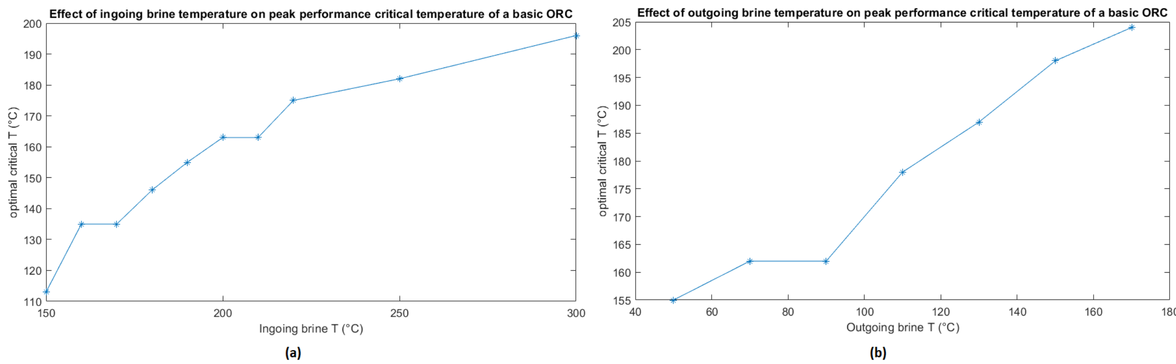

Figure 5.2: Effect of ingoing brine temperature (a) and maximum outgoing brine temperature (b) on working fluid optimal critical temperature.

5.2.2 Working fluid behaviour in a supercritical ORC

For the non-regenerative and regenerative supercritical ORC systems, only working fluids with a critical temperature below 440 K were analyzed. This was done because higher critical temperatures would not allow a supercritical system to function or would result in very low work outputs. When the results for these systems were analyzed it became apparent that the critical properties of the fluids are no longer a major factor. The only analyzed property that showed a significant relation with the net work output was the molar mass of the working fluid. The results are plotted in figure 5.3a and 5.3b.

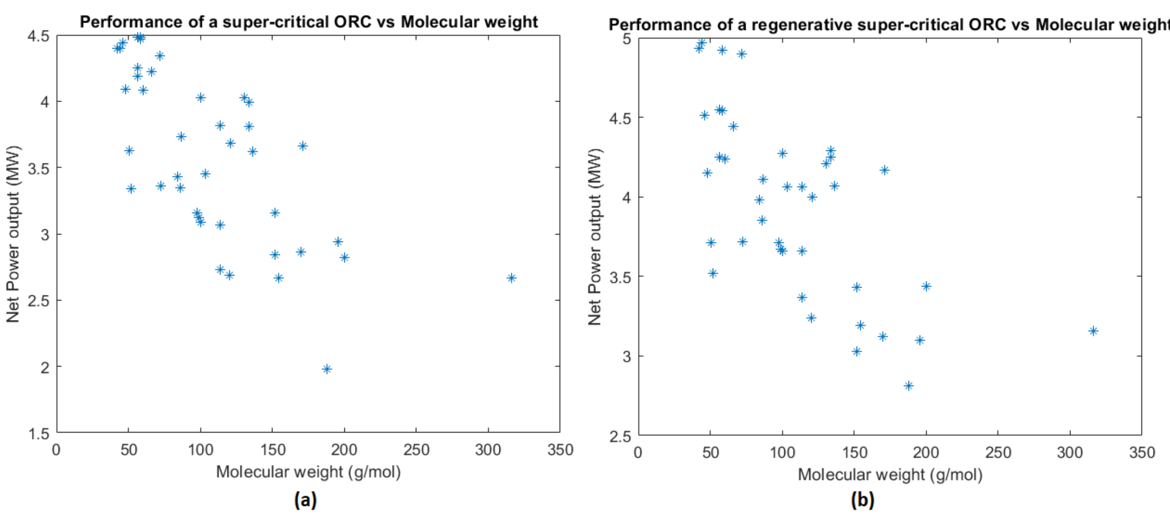

Figure 5.3: Net work output versus molar mass for a (a) supercritical ORC (b) regenerative supercritical ORC.

From figure 5.3 again similar behaviour is noted for the non-regenerative and regenerative system. A behaviou that seems linear. An outlier is caused by a substance called Novec649, this refrigerant has a very high molecular mass and performs better than one might expect if the relation between the net power and molecular weight is indeed linear. Because of this one might expect the relation to be logarithmic, rather than linear. This would mean that with an increasing molecular mass, the net work output would not decrease in a linear rate, but increasingly slowly. Because there are not enough datapoints (working fluids) with a molecular mass between 200 and 300 g/mol to verify this theory, a linear relation is assumed only on the interval [40,200] for the molecular mass.

For both supercritical systems Pearsons correlation coefficient is around 0.7, which indicates a strong linear relation. For more information about the determination of Pearsons correlation coefficient see appendix E.

5.3 Working fluid selection

For the subcritical systems as well as the supercritical systems five potential working fluids will be selected. It is important that they fulfill all requirements set in section 3 they should also have a high net work output. The selected working fluids for each system are:

For the subcritical systems:

1. Cis-2-Butene
2. Neopentane
3. R1233zd(E)
4. R245fa
5. n-Butene

For the supercritical systems:

1. n-Propane
2. Propylene
3. R152A
4. Dimethyl ether
5. Trans-2-butene

There are some properties of the working fluids that are important, but that have not been looked at. These properties are stated in chapter 3. The important non-thermodynamic properties of the working fluids are listed below in table 5.1.

Table 5.1: non-thermodynamic fluid properties of selected working fluid

Working fluid	Toxicity	Flammable	GWP	ODP
Cis-2-Butene	Low	Highly	20	0
Dimethyl ether	Low	Highly	1	0
n-Butane	Low	Highly	20	0
Neopentane	Low	Highly	0	0
n-Propane	Low	Highly	3	0
Propylene	Low	Highly	2	0
R1233zd(E)	Low	No	4.5	0
R152A	No	Moderately	124	0
R245fa	No	No	1030	0
Trans-2-butene	Low	Highly	20	0

These fluids are all deemed suitable for use in the ORC system. Their performance will be analyzed more thoroughly in the the next section.

Chapter 6

Sensitivity Analysis

In this section the behaviour of the different ORC systems will be determined when certain variables are changed. For example, the efficiencies of the pump and turbine could be different than the assumed values. Also the in and outgoing brine temperatures are far from certain and may even vary over the years. To make sure the system will still function under different operating conditions, this sensitivity analysis was done.

6.1 Brine properties

As mentioned before, the brine properties are very important to the functioning of the ORC. To determine the ORC output, three main properties of the brine need to be known to be able to determine the functioning of the ORC. These are: the ingoing brine temperature, the outgoing brine temperature and the mass flow of the brine. So far, assumptions were done for the quantities of these variables. However, since these assumptions will never be entirely correct and the fact that the brine properties might actually change during the operation period of the ORC. The behaviour of the ORC will be identified when these variables change.

6.1.1 Brine ingoing temperature

The first brine parameter that will be varied is the brine ingoing temperature. All working fluids were compared at a brine ingoing temperature of 200 Celsius (473 K). However, it might be possible that the brine temperature is slightly higher or lower than expected. Furthermore, it is conceivable that during the operation of the plant, the brine temperature decreases over the years. Therefore the maximum output is determined as a function of the ingoing temperature. The results of the sensitivity analysis for the brine ingoing temperature can be seen in figure 6.1

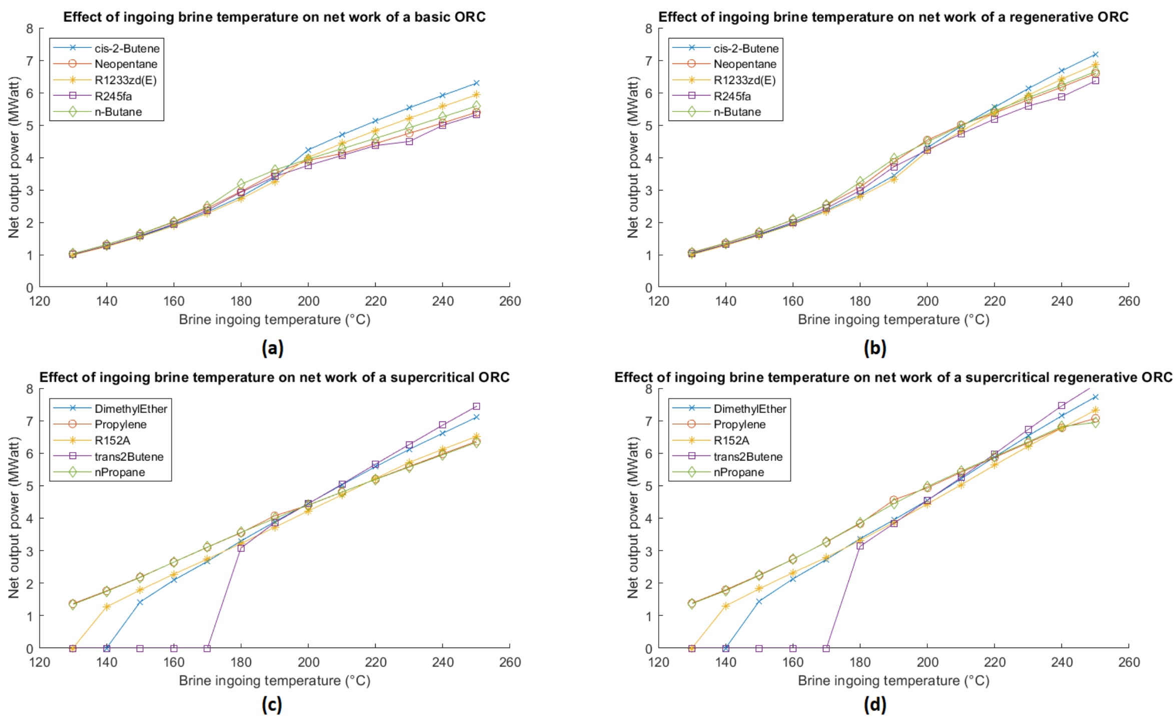

Figure 6.1: Effect of different ingoing brine temperature on output of a (a) basic ORC (b) regenerative ORC (c) supercritical ORC (d) regenerative supercritical ORC

From figure 6.1 it can be seen that for the subcritical systems the choice of working fluid seems to matter more when the ingoing temperature increases. Furthermore, the supercritical systems generally show a more linear behaviour than the subcritical systems.

6.1.2 Brine outgoing temperature

Next, the sensitivity of the systems the outgoing brine temperature is tested. In chapter 3 it was stated that the injection temperature is typically between 50 °C and 100 °C. For most calculations up untill now, the outgoing brine temperature was assumed to be 70 °C, but it would be beneficial for the system if this temperature could be lowered. It is also interesting to see how an increased outgoing temperature effects the output of the system, since the allowed injection temperature might be higher than 70 °C, if the brine has a higher outgoing temperature from the ORC than the allowed reinjection temperature, this heat might also be used for co-generation of heat and electricity. The results of the sensitivity analysis for the brine outgoing temperature can be seen in figure 6.2.

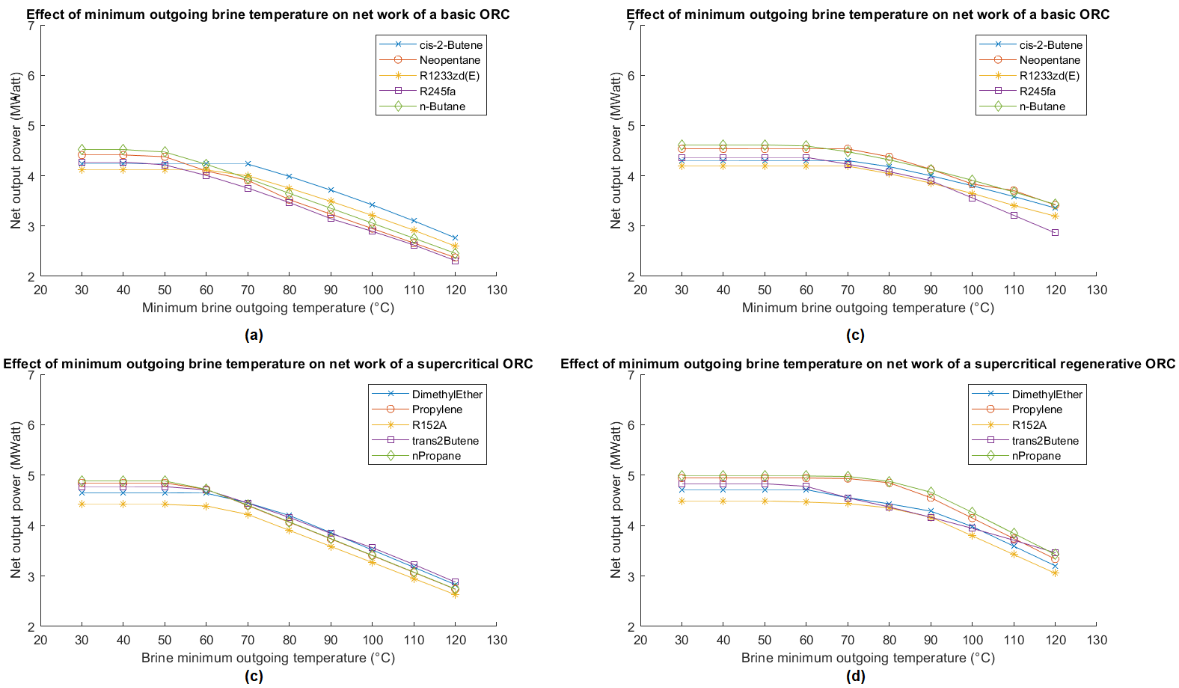

Figure 6.2: Effect of different outgoing brine temperature on output of a (a) basic ORC (b) regenerative ORC (c) supercritical ORC (d) regenerative supercritical ORC

From figure 6.2 it can be observed that the regenerative systems are less sensitive to an increasing outgoing brine temperature than the systems without a regenerator. Another interesting observation is that there is no real increase in power to be gained from decreasing the outgoing brine temperature from 50 °C to 30 °C. This is the case for the regenerative systems as well as for the non-regenerative systems. Should a regenerator be used (and the other variables kept the same to the assumed values), no real increase in power can achieved from lowering the outgoing brine temperatur below 70 °C.

6.1.3 Brine mass flow

The mass flow coming from the brine is a difficult thing to predict. The effects of the rate of extraction of geothermal water, and the temperature and rate at which it is injected are still being investigated (Clotworthy et al., 2010). Extracting water from the well too fast might lead to unwanted seismic activity (Vis, 2018), but will increase the proceeds of the ORC plant. Finding the best extraction rate is beyond the scope of this project, but the effect of different mass flow rates from the geothermal well are presented below in figure 6.3.

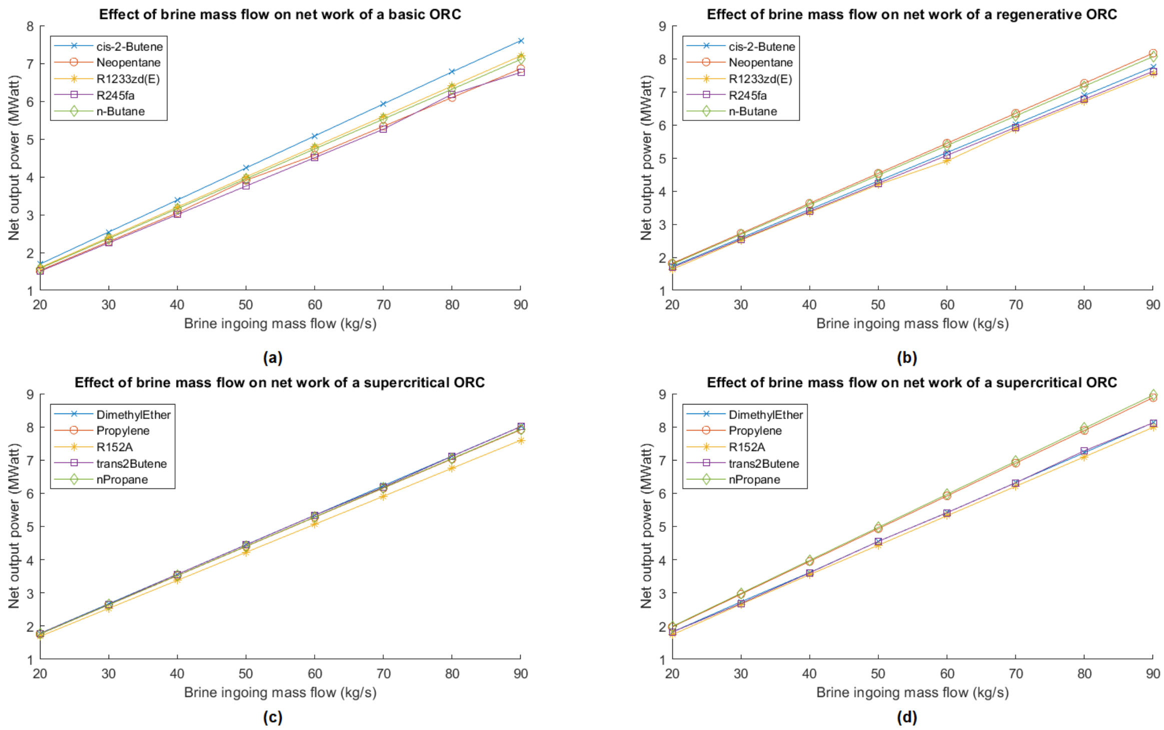

Figure 6.3: Effect of different brine mass flow rates on output of a (a) basic ORC (b) regenerative ORC (c) supercritical ORC (d) regenerative supercritical ORC

From figure 6.3 it can be seen that as to be expected, there is a linear relation between the net power output of the system and the brine mass flow. What can also be seen is that when the brine mass flow increases, the absolute difference in performance of the working fluid becomes more clear.

6.2 ORC properties

The ORC system itself is also based on a few critical assumptions. As mentioned in the start of this chapter, the efficiencies of the pump and turbine will looked at as well as the minimum allowable pinch point temperature difference in the heat exchangers. These are all properties of ORC components. Except for these component characteristics a few thermodynamic limitations will also be investigated. These are the minimum condensation temperature and the maximum operating pressure of the cycle. Normally the maximum operating pressure is determined by the limitation of the components. However, since these limitations are not calculated, it is considered a property of the ORC cycle.

6.2.1 Pump efficiency

The assumed pump efficiency of 0.8 is an efficiency that is often encountered in literature. However pumping efficiencies ranging between 0.65 and 0.85 can be found (Borsukiewicz, 2013). To see the effect of an incorrect assumption a sensitivity analysis was done for the efficiency of the pump and can be seen in figure 6.4.

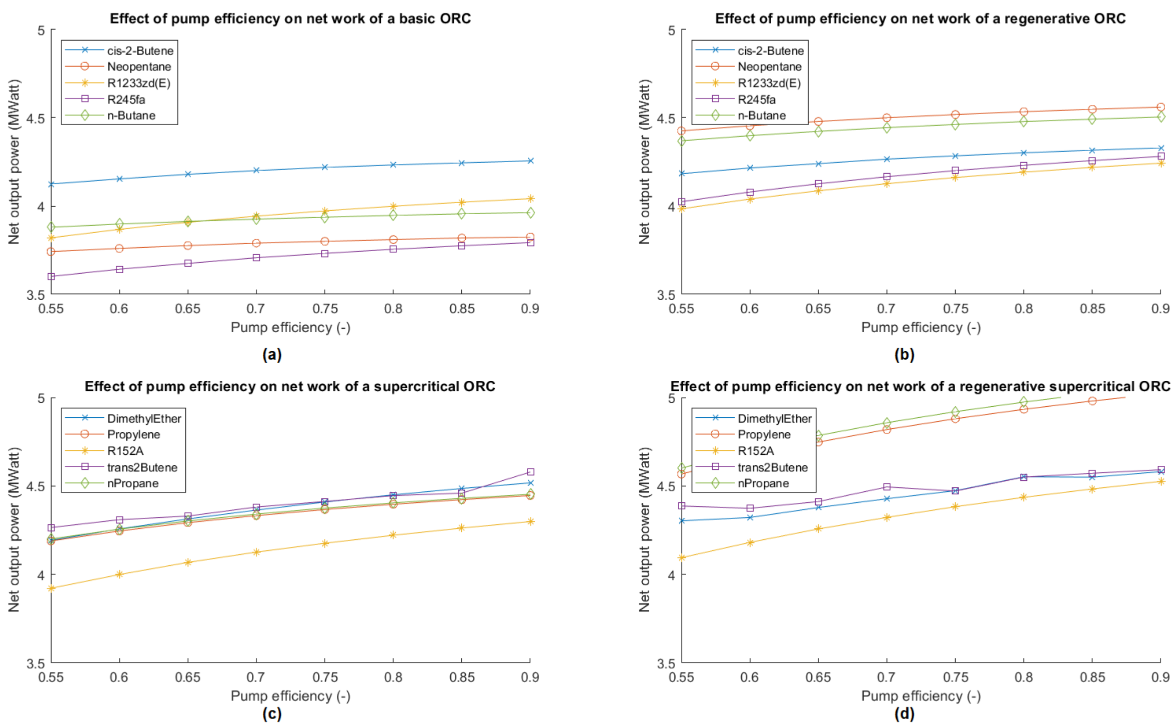

Figure 6.4: Effect of different pump efficiencies on output of a (a) basic ORC (b) regenerative ORC (c) supercritical ORC (d) regenerative supercritical ORC

From figure 6.4 it can be concluded that the efficiency of the pump does not have a very large impact on the total electricity generation of the plant. Especially increasing an already high efficiency will hardly result in any noticeable improvement of the ORC. The regenerative systems seem to be more effected by the pump efficiency, because these work with higher pressures.

6.2.2 Turbine efficiency

The turbine efficiency was estimated to be equal to the pump efficiency at 0.8. Efficiencies of dedicated turbines can reach as high as 0.9, in practice small scale turbines working with lower pressures often have efficiencies lower than 0.75 (Vanslambrouck et al., 2011). To see in what magnitude this effects the output power the turbine efficiency was analyzed. The results can be seen in figure 6.5.

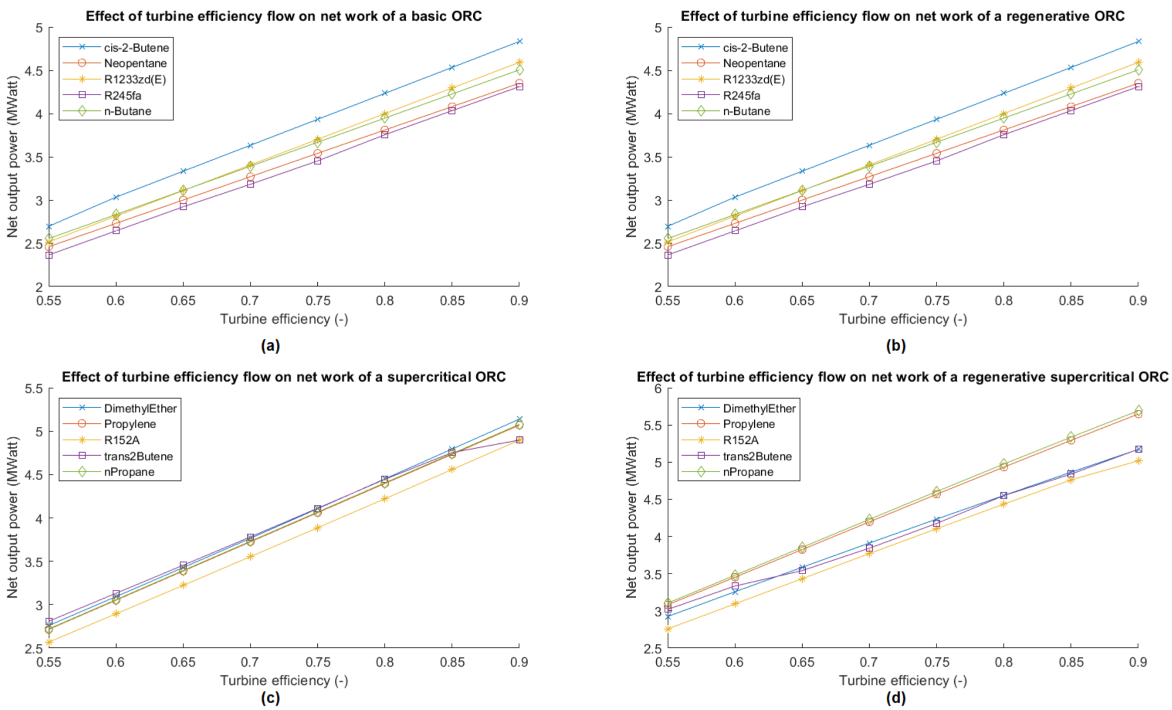

Figure 6.5: Effect of different turbine efficiencies on output of a (a) basic ORC (b) regenerative ORC (c) supercritical ORC (d) regenerative supercritical ORC

Figure 6.5 shows an almost perfect linear relation between the turbine efficiency and the net output power of the system in nearly all cases. Increasing the efficiency of the turbine will result (much) higher outputs. If the turbine would have an efficiency of 0.9, which should theoretically be achievable, this would mean an increase of 0.5 up to 1 MWatt for any of the systems. Maximizing the turbine efficiency should be a priority when selecting the actual turbine for the ORC system, this will likely outweigh the increased cost of the turbine itself.

6.2.3 Pinch point temperature difference

There are quite a few heat exchangers present in the ORC systems and they also influence the performance of the ORC. For example, the determination of the maximum mass flow in chapter 4.2.2 is dependent on the minimum allowable pinch point temperature difference. A lower pinch point temperature difference might mean increased performance of the system, but also means increased cost of the heat exchanger (van Erdeweghe et al., 2017) . To see the magnitude of this effect, the net output power was plotted for different minimum pinch point temperature differences in figure 6.6.

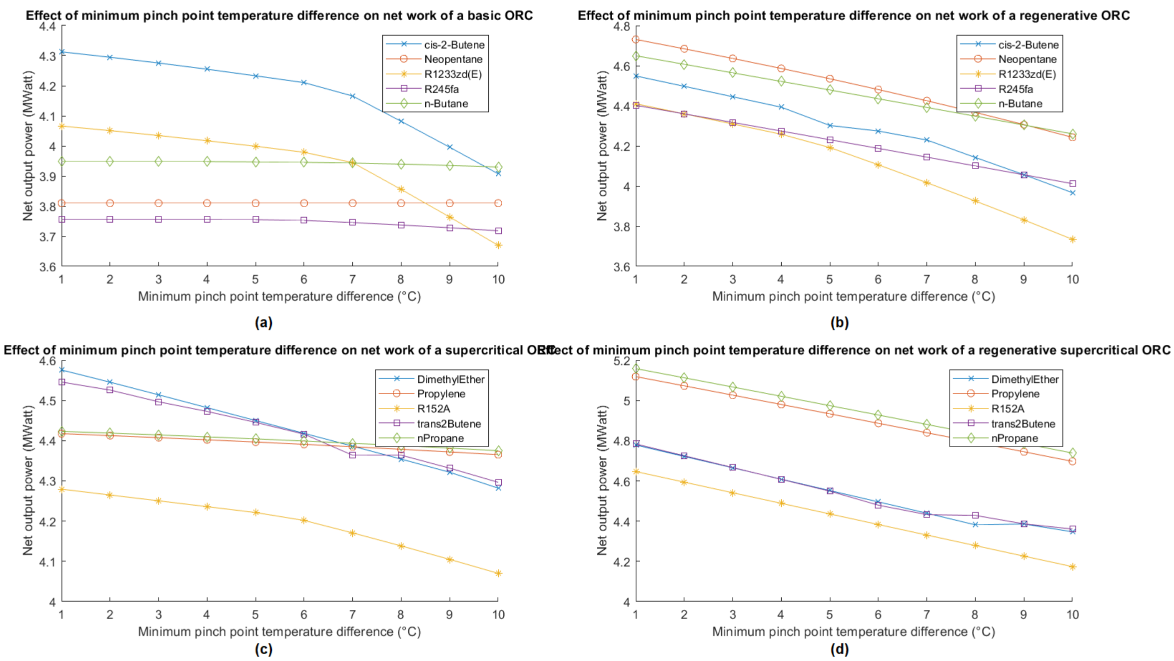

Figure 6.6: Effect of different minimum pinch point temperature differences on output of a (a) basic ORC (b) regenerative ORC (c) supercritical ORC (d) regenerative supercritical ORC

Figure 6.6 shows that not all working fluids react to the minimum pinch point temperature in the same way. For some fluids in the non-regenerative system we see hardly any effect on the output power at all. However, the most common behaviour seems to be a linear decrease in performance. This linear behaviour is likely due to the effect that the working mass flow is decreased to compensate for the increases pinch point temperature difference, while the optimal working pressure, condensation temperature and maximum cycle temperature stay the same. For some cases we see a bend in the curve where the rate of decrease in performance increases. This is explained by the fact that one or more of the aforementioned optimized factors change.

6.2.4 Condensation temperature

The last parameter that will be analyzed is the condensation temperature of the cycle. The assumed value seems realistic, but it might not be achievable on very hot days. Furthermore, in chapter 7 co-generation will be examined. If the latent heat of condensation would be used for the generation of heat for use in a heating network, this also means that the condensation temperature needs to be raised. The results of the analyses of the condensation temperature can be seen in figure 6.7.

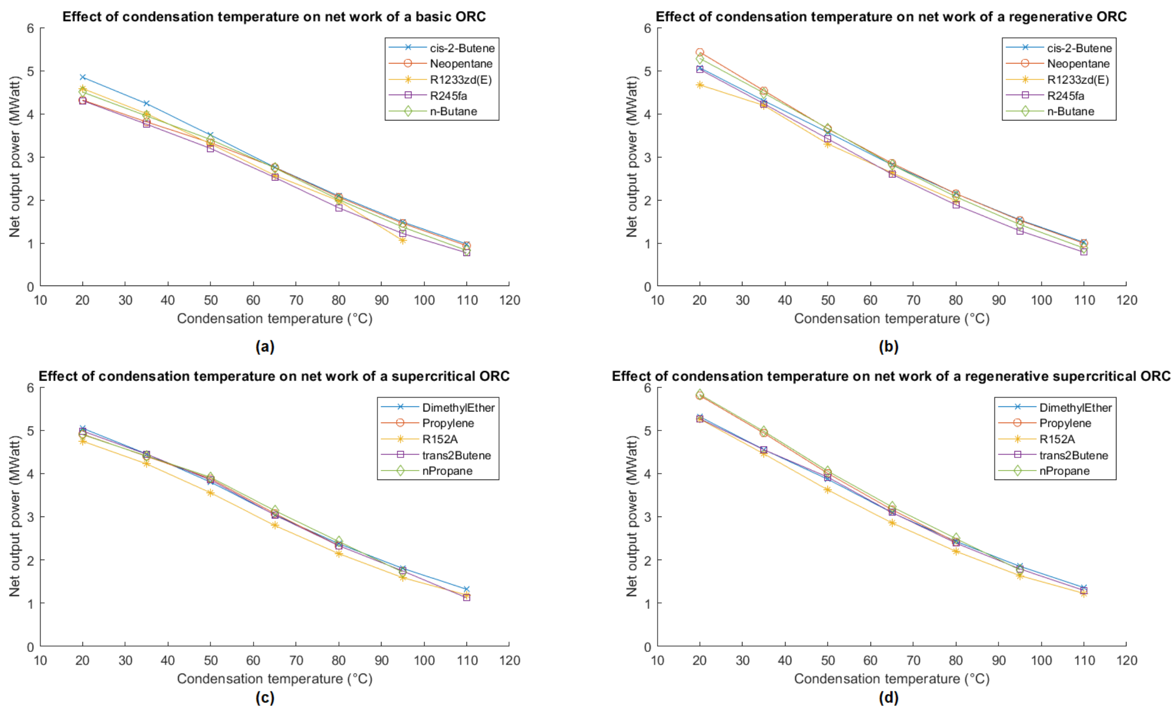

Figure 6.7: Effect of different condensation temperatures on output of a (a) basic ORC (b) regenerative ORC (c) supercritical ORC (d) regenerative supercritical ORC

In figure 6.7 similar behaviour can be noted for all systems. The behaviour of the systems is close to linear. Having a lower condensation temperature will lead to better performance of the ORC, because the turbine will generally be able to expand to a lower pressure. However, with a condensation temperature close to environmental temperature, the latent heat can not be used for other applications.

Chapter 7

Optimization for Co-Generation

With the functioning of multiple systems known, co-generation can now be investigated. In this chapter three different ways of generating enough heat to provide heat to a heating network are proposed. Two different situations will be looked at: The first will be providing heat to a heating network that accepts all heat that the co-generation plant can produce. In the second situation a specific amount of heat needs to be provided. This amount was specified in chapter 3 to be 39 MWatt. This is enough to service the entire municipality of Soest.

Since a lot of energy is lost in the condenser, it is vital that this energy can be harvested for use in the heating network. However to comply with the list of requirements in chapter 3, the heat must be delivered at a high temperature. This makes raising the condensation temperature inevitable. The three proposed co-generation systems are: High temperature condensation co-generation (HTCC), after-heating co-generation (AHC) and dual condensation co-generation (DCC).

For these systems a basic ORC using the working fluid n-butane is assumed. These methods of co-generation will be explained in this chapter. Furthemore, the analysis was done for two sets of brine ingoing properties. The different brine properties (situation A and B) are listed in table 7.1:

Situation name	Brine mass flow (kg/s)	Brine ingoing Temperature (°C)
A	50	200
B	60	175

Table 7.1: Different ingoing brine properties for co-generation

The MatLab code used for the optimized plants can be found here or by following the URL given in appendix G.2.

7.1 High temperature condensation co-generation

A schematic lay-out of the first type of co-generation plant is shown in figure 7.1. The heating network (HNW) is shown in blue. The cold flow from the heating network enters at 55 °C at the left of the figure. It is divided (not equally) into two flows. The first flow is heated by the remaining heat in the brine, which is kept at a minimum of 95 °C (90 °C+T_{pp}) to be able to heat the flow from 55 °C to 90 °C. The second flow passes through the condenser. The condensing process occurs at 95 °C and the latent heat from the condensing process is used to heat the second flow from 55 °C to 90 °C. The two HNW flows then merge and are then sent back for heat consumption. The advantages and disadvantages of this co-generation system are listed below

advantages:

1. Due to the use of the latent heat in the condenser, the required fan power of the air cooled condenser is reduced or even eliminated.

2. Because of the fact that the heating of the HNW split flows are independent of each other. When the required amount of heat fluctuates during operation of the plant the heat produced can be reduced (somewhat) at the brine side, without increasing the required fan power.

3. Due to the simplicity of the plant, the purchased equipment cost will be relatively low.

Disadvantages:

1. Due to the high condensation temperature, electricity production will be limited.

2. When this system would be designed for a lower heat load, this can only lead to a decrease in electrical performance.

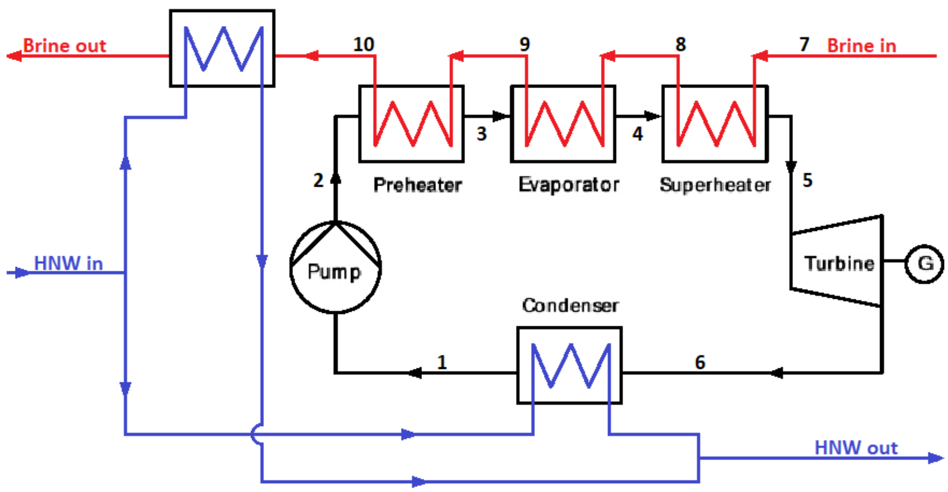

Figure 7.1: Schematic lay-out of the co-generation plant with integrated heating network using high temperature condensation co-generation

7.1.1 Maximum heating with HTCC

When the maximum heating is to be achieved, all heat from the condenser and all remaining heat in the brine is used to heat up water for the HNW. The system was optimized for a condensing temperature of 95 °C. The resulting output power in MWe and MWth are shown below, in table 7.2. For more information about this ORC cycle, see appendix F.

Since both the condensation temperature and the outgoing brine flow are at least T_{pp} higher than the required temperature of the HNW. The total heat production can be determined easily by adding up the heat available between the exit of the turbine and the exit of the condenser and the heat available from the brine. The brine can be cooled to a temperature of 70 °C:

$$P_{th} = \dot{m}_{wf}(H6 - H1) + \dot{m}_b(H10 - H_{b,out,min}) \tag{7.1}$$

The electrical power is determined using the process described in chapter 4 only with a parasitic condenser fan power equal to 0, since all the latent heat is used for the HNW. The total heat and electricity production for situations A and B can be found in table 7.2.

Situation	Electrical power (MW_e)	Thermal power (MW_{th})
A	1.59	26.46
B	1.14	25.81

Table 7.2: maximum heat and corresponding electricity production when HTCC is used

7.1.2 Restricted heating with HTCC

In chapter 3 it was specified that the co-generation plant would be designed to fit the heat requirements of the municipality of Soest. The municipality has an avarage heat consumption of 39 MW_{th}, since the total heat available from one UDG well is equal to 27.9 MW_{th} in situation A and 26.8 MW_{th} in situation B, at least 2 UDG wells are needed in both situations. Therefore, it is assumed that the required heat from one UDG well is equal to 19.5 MW_{th}.

To produce the specific amount of heat, the heat production needs to be reduced in comparison the the maximum heating situation. This is done by increasing the reinjection temperature of the brine. If the reinjection temperature becomes equal to the temperature of the brine after the preheater (T10), the HNW flow through the condenser is reduced. This would mean that the working fluid flow is not completely condensated by the HNW flow, meaning air cooled heat exchangers are required to condensate the remaining working fluid vapour. These air cooled heat exchangers consume electricity and this must be taken into account. The new electricity and heat production for the specific demand can be found in table 7.3.

Situation	Electrical power (MW_e)	Thermal power (MW_{th})
A	1.59	19.5
B	1.14	19.5

Table 7.3: Restricted heat and maximum electricity production when HTCC is used

There is no difference in electricity production between the maximum and restricted heat situations. This is due to the fact that the heat required is higher than, or equal to the latent heat produced by the condensation process in the ORC. The only effect of the reduced heat production is a higher reinjection temperature: 103 °C in situation A and 88 °C in situation B, compared to 70 °C for both situations when maximum heat is extracted from the brine.

7.2 After-heating co-generation

The second co-generation system that will be analyzed is called after-heating co-generation (AHC). This system is based on the fact that optimized (basic) ORC systems with high condensation temperatures (e.g. 90 °C), often show outgoing brine temperatures higher than the condensation temperature. It is proposed that using the temperature difference between the outgoing brine temperature (T10) and the condensation temperature (T1) lower the condensation temperature of the system, will increase the performance of the co-generation plant. The schematic lay-out of the system can be seen in figure 7.2. The latent heat of the brine is extracted by the incoming cold flow of the HNW. However, the condensation process occurs at a temperature below the required temperature for the HNW. Therefore the HNW flow is after-heated with the outgoing brine flow to 90 °C. The advantages and disadvantages are:

advantages:

1. Due to the use of the latent heat in the condenser, the required fan power of the air cooled condenser is reduced or even eliminated.

2. When this system would be designed for a lower heat load, the electricity output will be higher.
3. Due to the simplicity of the plant, the purchased equipment cost will be relatively low.

Disadvantages:

1. This system is not suited for reinjection temperatures far below the HNW maximum temperature.
2. When the required amount of heat fluctuates during operation of the plant, the amount of heat produced can not be reduced without the use of electricity consuming air cooled condensers.

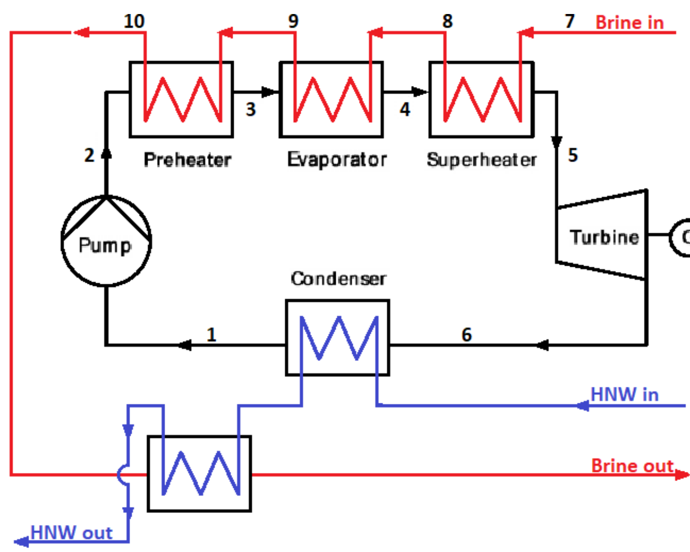

Figure 7.2: Schematic lay-out of the co-generation plant with integrated heating network using after-heating co-generation

For an assumed condensation temperature T1, The HNW temperature is raised from 55 °C to $T1 - T_{pp}$, furthermore the HNW is assumed to be kept at a constant pressure of 20 bar (Alliander, 2018). The maximum massflow of the heat network, $\dot{m}_{HNW}$, depends on the heat load and will be determined later.

The amount of after-heating that is required is determined with:

$$P_{AH,required} = \dot{m}_{HNW}(H_{HNW,out} - H_{HNW,cond,out}) \tag{7.2}$$

The thermal power available (from the brine after the preheater) for after-heating can be computed, assuming the brine outgoing temperature is equal to the condensation temperature, with:

$$P_{AH,available} = \dot{m}_b(H10 - H_{b,out}) \tag{7.3}$$

7.2.1 Maximum heating with AHC

Again a situation is determined where all possible heat is extracted from the brine. This means that the brine outgoing temperature should be 70 °C. This means that for the AHC co-generation system, the condensation temperature of the ORC cycle is also 70 °C. The HNW flow will be heated with the latent heat from 55 °C tot 65 °c and using the leftover heat in the brine after the preheater, it is heated from 65 °C to 90 °C. Unfortunately, because the HNW flow is (much) larger than the brine flow, the minimum temperature of the brine after the preheater will be very high. Resulting in low electricity outputs.

The maximum mass flow of the HNW past the condenser is determined with:

$$\dot{m}_{HNW} = \frac{H6 - H1}{H_{HNW,cond,out} - H_{HNW,in}} \tag{7.4}$$

Assumptions can be done for the brine temperature after the preheater (T10) and for the condensation temperature. The ORC properties can be determined with the model described in chapter 4. The required and available power for after-heating can be determined based on this model. Based on the results, the assumptions for condensation temperature and T10 are changed to better fit the available power for after-heating to the required power for after heating. This iterative process can be repeated untill the difference between the required and available power is negligible. When this is done, the total heat produced is equal to:

$$P_{th} = \dot{m}_{HNW}(H_{HNW,out} - H_{HNW,in}) \tag{7.5}$$

The final electricity and heat output of the AHC system with maximum heat output can be found in table 7.4.

Situation	Electrical power (MW$_e$)	Thermal power (MW$_{th}$)
A	0.92	27.01
B	0.87	25.98

Table 7.4: maximum heat and corresponding electricity production when AHC is used

7.2.2 Restricted heating with AHC

The AHC system is more suited for lower heat outputs. When the reinjection temperature is allowed to be higher than it's minimum allowable value, the heat output will become lower, but it will still be able to satisfy the average heat demand of 19.5 MW$_{th}$. The advantage of this system is that the remaining heat in the brine can be used to lower the condensation temperature of the ORC system, thus increasing the electricity output.

The approach for the system with restricted heat output is slightly different than the approach for maximum heat production with AHC. Since the required thermal power is known, the required HNW mass flow can be determined by rewriting the expression from equation 7.5:

$$\dot{m}_{HNW} = \frac{P_{th}}{H_{HNW,out} - H_{HNW,in}} \tag{7.6}$$

again an assumption is done for the condensing temperature and minimum brine temperature after the preheater. The after-heating required and available are computed using equations 7.2 and 7.3. One of the two assumptions is changed to find the condensing temperature corresponding to a certain minimum brine temperature after the preheater where the after-heating required is equal to the after-heating available. Finally, the required fan power must be computed. This is done with equations 4.5 and 4.6, but the heat that must dissipated in the air cooled condenser is equal to:

$$P_{th,lost} = \dot{m}_{wf}(H6 - H1) - \dot{m}_{HNW}(H_{HNW,in} - H_{HNW,cond,out}) \tag{7.7}$$

Here: $H_{HNW,cond,out}$=f($P_{HNW}, T6 - T_{pp}$)

The performance of the system for the situations A and B, with the corresponding condensation temperature and brine temperature after the preheater (T10) can be found in table 7.5. Note that the reinjection temperatures are equal to the condensation temperatures. For more imported data about this co-generation system, please refer to appendix F.

Situation	Electrical power (MW_e)	Thermal power (MW_{th})
A	1.71	19.5
B	1.48	19.5

Table 7.5: Restricted heat and maximum electricity production when AHC is used

Compared to the HTCC system, a performance increase of 7.5% can be seen for situation A and 7.7% in situation B. Which is quite significant. From this analysis it has become apparent that this system is not ideal, when the reinjection temperature should be as low as possible.

7.3 Dual condensation co-generation

The final co-generation plant that is analyzed utilizes a dual condensation principle. It is similar to the HTCC plant, but it utilizes two smaller ORC systems to heat up the HNW flow in two steps. Before entering the system, both the HNW flow and the brine flow are split into two flows (again: not equal flows). The first of the two ORC systems heats the HNW flow from 55 °C to T_{pp} below it's condensation temperature in the low temperature condenser. Then, the HNW flow is heated further to 90 °C in the high temperature condenser of the second ORC cycle. This means that the condensation temperature of the second ORC cycle, must be equal to 95 °C, similar to the HTCC plant. Again, the leftover heat in the brine after the preheater(s), can be used to heat a HNW flow from 55 °C to 90 °C, just like the HTCC plant. This is done in two steps. Since the leftover heat in the brine from the LT condensation cycle will be lower than 95 °C, the HNW flow is first heated using this leftover heat from low temperature condensation (LTC) cycle and then heated to 90 °C using the leftover heat in the brine from the high temperature condensation (HTC) cycle. A schematic lay-out of the plant can be seen in figure 7.3.

This system system combines most of the advantages of the HTCC system and the AHC system:

advantages:

1. Due to the use of the latent heat in the condenser, the required fan power of the air cooled condenser is reduced or even eliminated.
2. When this system would be designed for a lower heat load, the electricity output will be higher.
3. Because of the fact that the heating of the HNW split flows are independent of each other. When the required amount of heat fluctuates during operation of the plant, the heat produced can be reduced (somewhat) at the brine side, without increasing the required fan power.

Disadvantages:

1. This system is quite complicated, which will make it relatively expensive.

Determining the heat and power of the system works the same as for an HTCC system, only the condensation temperature and distribution of the incoming brine mass flow must be assumed. With these assumptions the latent heat of the HTC cycle and LTC cycle can be determined. Using MatLab, an optimization can be done where the input variables are the brine massflow distribution and the condensation temperature of the LTC cycle. Using this optimization the optimal values can be found depending on the required heat load.

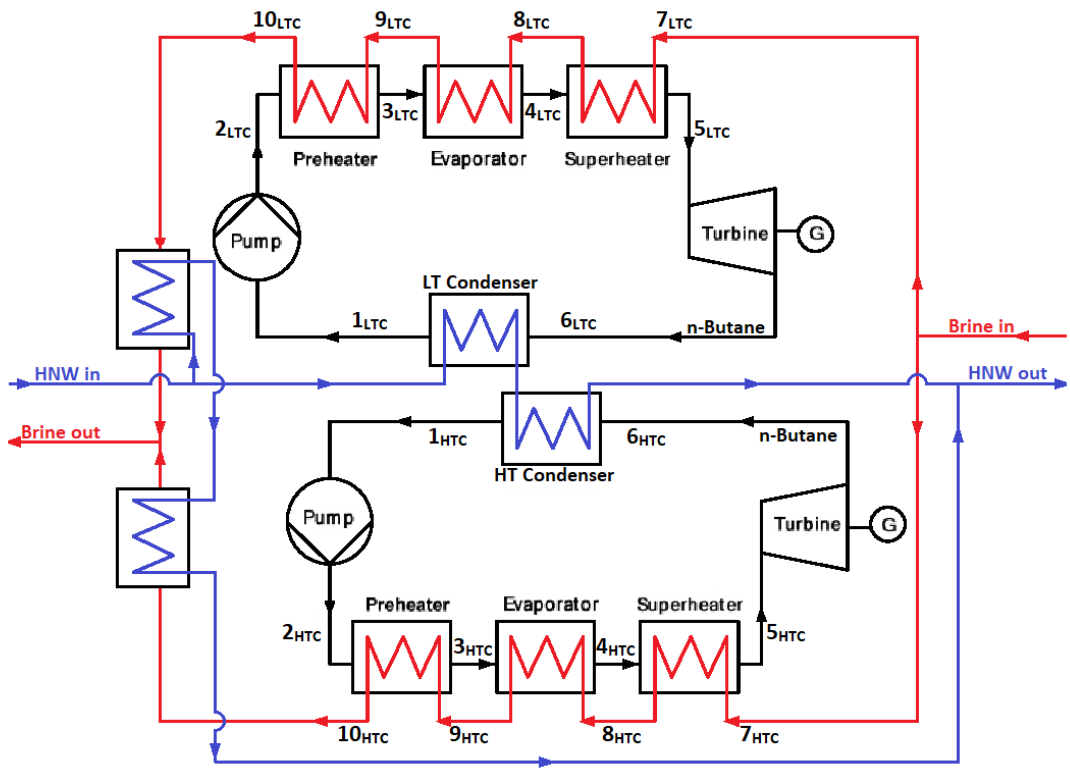

Figure 7.3: Schematic lay-out of the co-generation plant with integrated heating network using dual condensation co-generation

7.3.1 Maximum heating with DCC

To acquire the maximum possible heat load from the DCC system, the maximum possible massflow of the HNW flow through the low temperature condenser should be equal to the maximum possible massflow through the high temperature condenser:

$$\dot{m}_{HNW} = \frac{\dot{m}_{wf,LTC}(H6_{LTC} - H1_{LTC})}{H_{HNW,LTC,out} - H_{HNW,in}} = \frac{\dot{m}_{wf,HTC}(H6_{HTC} - H1_{HTC})}{H_{HNW,out} - H_{HNW,LTC,out}} \tag{7.8}$$

Using the optimization fmincon in Matlab, the values for the mass flow distribution and condensation temperature can be found for which this equation holds. In theory, for each mass flow distribution a condensation temperature can be found to satisfy this condition. The combination with the highest electricity output was chosen. The produced electricity and heat can be found in table 7.6. The specifications of the ORC cycles in the system can be found in appendix F.

Situation	Electrical power (MW_e)	Thermal power (MW_{th})
A	1.96	26.1
B	1.40	25.5

Table 7.6: maximum heat and corresponding electricity production when DCC is used

7.3.2 Restricted heating with DCC

Finally, the DCC system's performance will be examined for a situation where less than the maximum possible heat is required. When less heat is required, the condensation temperature of the LTC cycle can be reduced even further. The maximum amount of heat will be extracted from the leftover heat in the brine after the preheaters of the ORC cycles and the remaining heat will be produced using the the condensation process. Because the amount of heat that must be delivered at 90 °C is lower, meaning that the incoming brine mass flow through the HTC cycle can be reduced and the flow through the LTC cycle can be increased. This should lead to an increase in electrical power. An air cooled heat exchanger will be needed to dissipate the excess heat coming from the condensation process of the LTC cycle.

The total leftover heat in the brine after the preheaters is found with:

$$P_{th,b} = \dot{m}_{b,LTC}(H10_{LTC} - H_{b,min}) + \dot{m}_{b,HTC}(H10_{HTC} - H_{b,min}) \tag{7.9}$$

Since the total required thermal power. $P_{th,total}$ is known, the remaining heat that must be provided by the condensation processes of the combined ORC systems to the HNW equals:

$$P_{th,wf} = P_{th,total} - P_{th,b} \tag{7.10}$$

This means the required HNW mass flow through the the condensers can be computed. This must be equal to the mass flow that can be heated in the high temperature condenser:

$$\dot{m}_{HNW,cond} = \frac{P_{th,wf}}{H_{HNW,out} - H_{HNW,in}} = \frac{\dot{m}_{wf,HTC}(H6_{HTC} - H1_{HTC})}{H_{HNW,out} - H_{HNW,LTC,out}} \tag{7.11}$$

Again fmincon is used to find the combination of the lower condensation temperature and the mass flow distribution for which this equation holds. In theory, for each mass flow distribution a condensation temperature can be found to satisfy this condition. The combination with the highest electricity output was chosen. The resulting power and heat outputs for the two situations can be found below in table 7.7

Situation	Electrical power (MW_e)	Thermal power (MW_{th})
A	2.05	19.5
B	1.50	19.5

Table 7.7: Restricted heat and maximum electricity production when HTCC is used

Since it was found that the optimal evaporation pressure and degree of seperheating is equal in the HTC and LTC cycles in all cases, an alternative plant lay-out it suggested. The flows can be heated together and split after the superheater, this reduces the amount of heat exchangers required. The suggested improved plant lay-out can be found in figure 7.4.

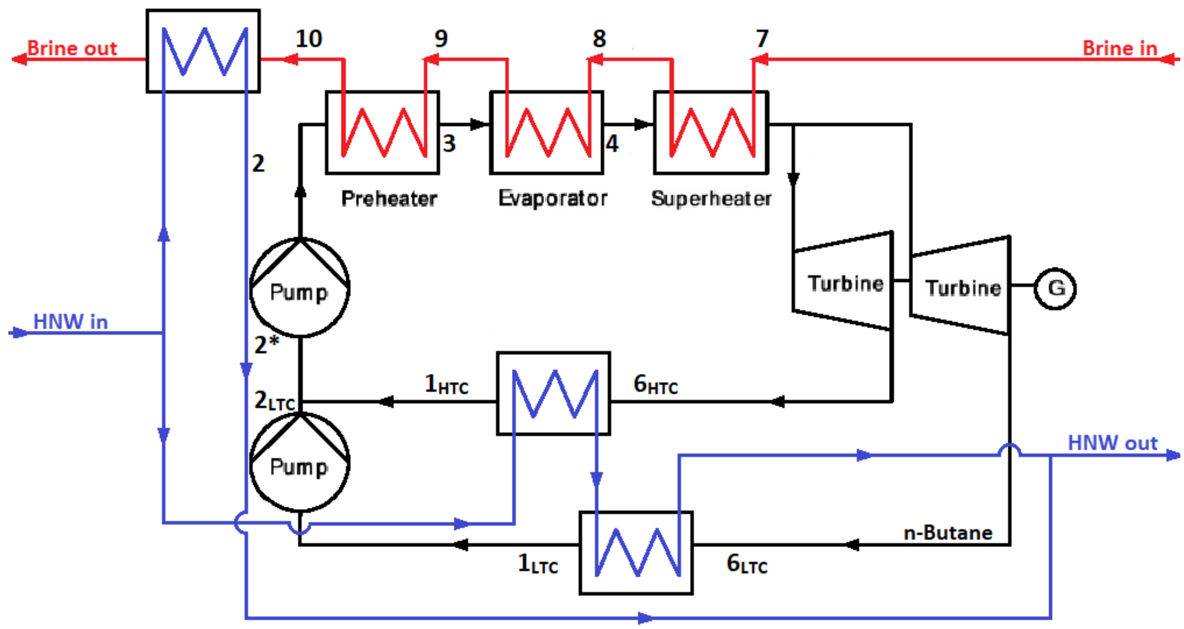

Figure 7.4: Schematic lay-out of the improved co-generation plant with integrated heating network using dual condensation co-generation

Chapter 8

Conclusions and Recommendations

8.1 Conclusions

In this report, an investigation was done into the implementation of an organic Rankine cycle in the Netherlands powered by ultra deep geothermal energy. Multiple systems were analyzed. It was found that the supercritical regenerative ORC yields the most promising net work output. However, this system is not yet widely used because of the sometimes extreme operating conditions. Therefore, a regenerative subcritical system could also be considered.

From an analysis of a large amount of working fluids it was found that for a sub critical system, the suitability of a working fluid is mainly dictated by it's critical temperature and to a lesser extent by the molecular weight of the working fluid. For the supercritical fluid a fairly strong correlation was found between the maximum electricity production of the system and the molecular weight of working fluid.

An extensive sensitivity analysis was done to determine the dependence of the system on the made assumptions and to be able to estimate the ORC behaviour when the brine conditions might be different then expected.

Finally, three co-generation plant lay-outs were suggested. In all three cases, a subcritical non-regenerative system was used for electricity production. Using multiple condensers at different temperatures showed the highest electricity production combined with the production of an adequate amount of heat. A disadvantage of this system is that it will require a (large) air cooled condenser in addition to the condenser cooled by the city's heating net work.

8.2 Recommendations

After the finishing of this investigation, there is still a lot more to be examined to make an adequate prediction about the potential use of an ORC system with UDG in the Netherlands.

1. First of all, the examined ORC systems are still not very complicated. There are numerous configurations that should be examined before a final decision about the plant lay-out is made. One example of this, is the dual pressure system discussed in appendix A.

2. From a thermodynamic point of view, the regenerative supercritical system would be the best combined with ultra deep geothermal water. However, a research could be done into the limitations of ORC components such as heat exchangers when the relatively large pressure is considered in the mechanical design of these components. The high pressures might increase the cost of the ORC and might also entail increased safety concerns. Therefore, the applicability of a supercritical ORC should be examined further

3. In this study, the considered working fluids are all part of the CoolProp directory. In practice there are a lot more fluids available, which could be more beneficial for this specific situation. In some cases, fluids can be mixed to obtain more favorable properties, these mixtures are not part of the CoolProp library and should be examined

4. The examined co-generation systems were designed using a basic ORC only. They could also be designed using a more efficient system, like a regenerative or regenerative supercritical ORC. This is likely to improve the performance of the co-generation system.

Bibliography

Alliander (2018). Warmtenet nijmegen - veelgestelde vragen. warmtenetnijmegen.nl.

Borsukiewicz, A. (2013). Pumping work in the organic rankine cycle. Researchgate.com.

Boz, B. and Diez, A. (2017). Comparative study of sub-critical and supercritical orc applications for exhaust waste heat recovery. Waset.org.

BP World Energy (2018). Bp statistical review of world energy. bp.com.

Clotworthy, A., Lawless, J., and Ussher, G. (2010). What is the end point for geothermal developments: Modeling depletion of geothermal fields. Sinclair Knight Merz Ltd.

Diaz, A. R., Kaya, E., and Zarrouk, S. J. (2015). Reinjection in geothermal fields: A worldwide review update. Proceedings World Geothermal Congress 2015.

DiPippo, R. (2016). Geothermal power plants. Elsevier Ltd.

Duan, Y. Y. (2018). Parametric optimization and thermodynamic performance comparison of single-pressure and dual-pressure evaporation organic rankine cycles. Sciencetrends.com.

ENSOC (2016). Nederland blijft voorlopig afhankelijk van aardgas. ensoc.nl.

Goodarzi, M. and Dehghani Soltani, H. (2015). Thermal performance analysis of a reheating-regenerative organic rankine cycle using different working fluids. researchgate.net.

Guillen, D. P. and Zia, J. (2013). Modifications and optimization of the organic rankine cycle to improve recovery of waste heat. Idaho National Laboratory.

Kim, K. H. (2013). Effects of superheating on thermodynamic performance of organic rankine cycles.

LaerdStatistics (2018). Pearson product-moment correlation. statistics.laerd.com.

Lai, N. A., Wendland, M., and Fischer, J. (2010). Working fluids for high-temperature organic rankine cycles. Elsevier Ltd.

Latrash, F. A., Agnew, B., Al-Weshahi, M. A., and Eshoul, N. M. (2015). Optimal selection of using fluids (hfc, hcfc, hfc) for an organic rankine cycle utilising a low temperature geothermal energy source. 2015 5th International Conference on Environment Science and Engineering.

Lisheng, P. and Weixiu, S. (2016). Investigation on the pinch point position in heat exchangers. Journal of Thermal Science Vol.25, No.3.

Nguyen, D. (2017). Een geothermische elektriciteitscentrale in de omgeving van amersfoort. Herman de Groot Ingenieurs.

PORKKA (2018). F-gas regulation and refrigerant restrictions. porkka.com.

Radovic, L. (1997). Efficiency of energy conversion.

Rijksoverheid (2017). Regeerakkoord: 3.1 klimaat en energie. Rijksoverheid.nl.

Siemens (2014). Power generation: Siemens organic rankine cycle waste heat recovery with orc.

Smoor, P.-P. C. (2017). Warmtevisie gemeente almere. Gemeente Almere.

Soester Courant (2017). Eerste geothermiecentrale in nederland die zowel elektriciteit, warmte en koude kan opwekken. Soester Courant.

ten Voorde, L. (2018). Integratie van een warmtenet voor de gemeente soest. Intern onderzoek Larderel Energy.

van Erdeweghe, S., van Bael, J., Laenen, B., and D'Haeseleer, W. (2017). Influence of the pinch-point-temperature difference on the performance of the preheat-parallel configuration for a low-temperature geothermally-fed chp. IV International Seminar on ORC Power Systems.

van Vliet, E., de Keijzer, J., Slingerland, E., van Tilburg, J., Hofsteenge, W., and Haaksma, V. (2016). Collectieve warmte naar lage temperatuur: Een verkenning van mogelijkheden en routes. GREENVIS Energy Solutions.

Vanslambrouck, B., Vankeirsbilck, I., van den Broek, M., Gusev, S., and de Paepe, M. (2011). Efficiency comparison between the steam cycle and the organic rankine cycle for small scale power generation. Ghent University.

Vis, K. (2018). Thermal stimulation of the subsurface. HermanDeGroot Internal Research.

Volkers, C. and Wetzels, W. (2013). Nieuwste inzichten nederlands gasverbruik. ecn.nl.

Woodland, B. J., Braun, J. E., Groll, E. A., and Horton, W. T. (2014). Methods of increasing net work output of organic rankine cycles for low-grade waste heat recovery. Pardue University.

Yang, L., Braun, J. E., and Groll, E. A. (2004). The impact of fouling on the performance of filterevaporator combinations. Pardue University.

Zoeteman, B. (2014). Duurzaamheid leeft volop in gemeenten, maar kan effectiever. Groenekwesties.nl.

Appendix A

Literature research

A.1 Heat regeneration

One way of increasing the efficiency of an ORC is by using heat regeneration. When the working fluid leaves the turbine, it is still hot. Instead of cooling it down by using a condenser right after the turbine, the hot low pressure gas can be used to preheat the high pressure liquid later in the cycle. This way a lot of energy that would otherwise be lost, is put back in the cycle. The lay-out of a regenerative cycle is shown in figure A.1.

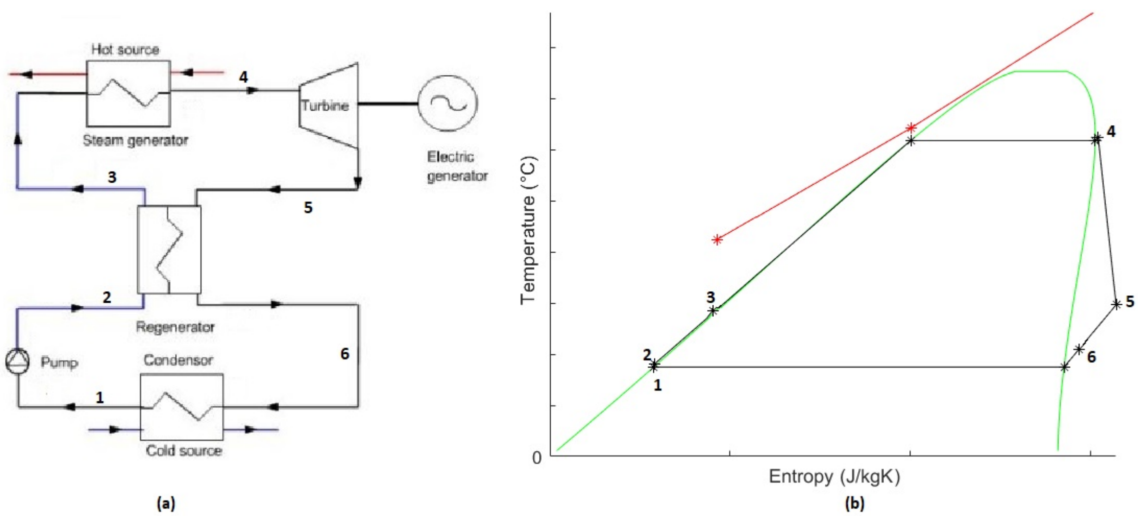

Figure A.1: Simplified lay-out of a basic ORC with heat regeneration and it's corresponding T-s diagram

A.2 Dual-pressure

Another way to decrease the exergy loss in the ORC plant is by using a low and high pressure turbines. In previous research it was already determined that most of the exergy loss occurs in the condenser and heat exchanger(Nguyen, 2017). While using regeneration helps with reducing the exergy loss in the condenser, by using high and low pressure turbines, the exergy loss in the heat exchangers (and evaporator) can also be reduced. A schematic overview of the dual-pressure ORC is shown in figure A.2a the Temperature-Entropy diagram for this system is shown in A.2b.

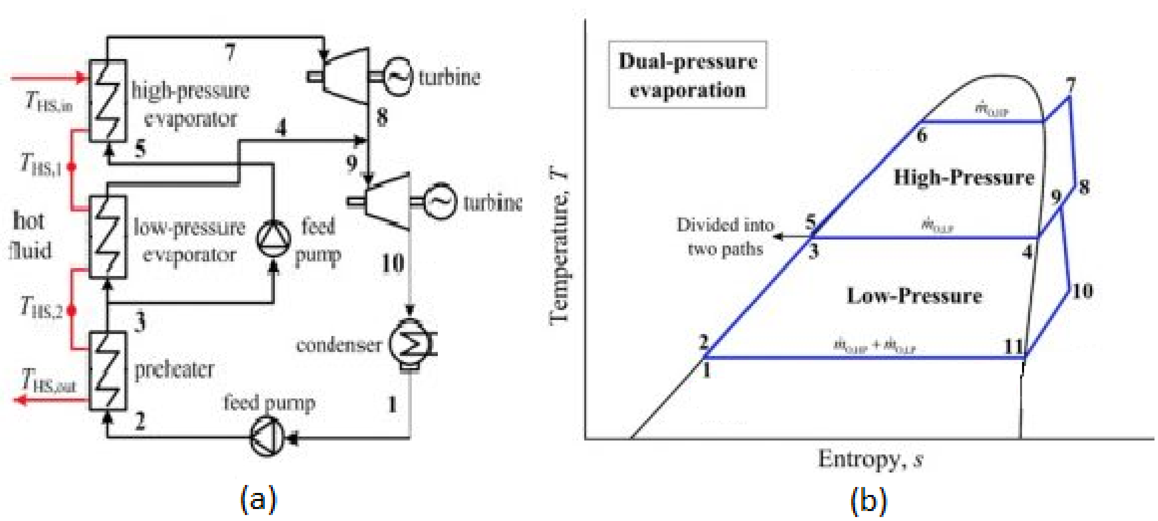

Figure A.2: Simplified lay-out of an ORC with dual pressure turbine (a) and the T-s diagram of the same system (b)(Duan, 2018)

The large amount of exergy loss is a consequence of a large temperature difference between the working fluid and geothermal fluid in the heat exchanger(Duan, 2018). This occurs due to flashing that happens in the evaporator. When using turbines at different pressures the saturated vapor state will be achieved at different temperatures for different pressures. By using the geothermal fluid first to vaporize the high pressure working fluid and using it to vaporize the low pressure working fluid. The temperature difference between the two fluids is minimized. The geothermal fluid temperature and working fluid temperature is shown for the basic ORC in figure A.3a and for the dual-pressure ORC in figure A.3b.

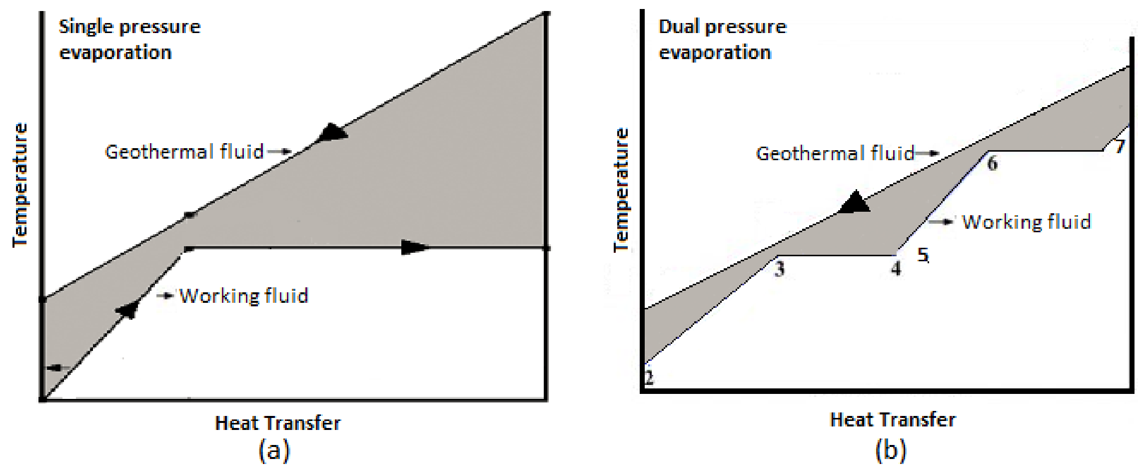

Figure A.3: Temperature difference between working fluid and geothermal fluid in a heat exchanger for a basic ORC (a)(DiPippo, 2016) and for a dual-pressure ORC (b)(Duan, 2018)

As one can see, adding another turbine working again on a different pressure would theoretically also decrease exergy loss. However the increase efficiency might not be enough to outweigh the added cost of an extra turbine, this is something that could be investigated. In theory, adding infinite heat exchangers and infinite turbines, this would lead to no loss in the heat exchangers. Because the supercritical state can only be achieved when the system operates on very high pressures, the system costs are expected to increase.

A.3 Reheating

Reheating is a process where the high pressure working fluid is partly expanded through a turbine, then reheated and then expanded to condenser pressure in a low pressure turbine. Reheating is mainly useful for normal rankine cycles or rankine cycles using a wet fluid. However, studies have shown that using reheating is also beneficial for the efficiency in an ORC with a dry working fluid (Goodarzi and Dehghani Soltani, 2015). For certain working fluids, reheating increases the performance of the ORC up to 7.8 percent. An ORC system with reheating and regeneration is shown in figure A.4a and the corresponding T-s-diagram is shown in figure A.4b.

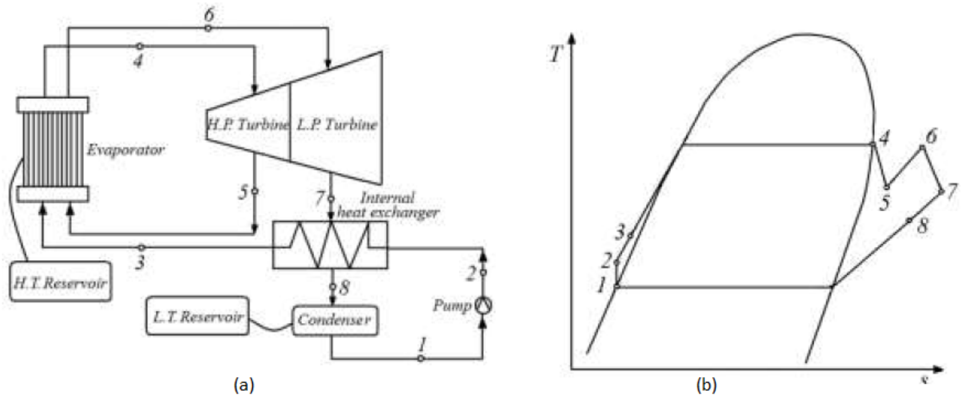

Figure A.4: Simplified lay-out of a regenerative reheating ORC (a) and it's corresponding T-s-diagram (b)(Goodarzi and Dehghani Soltani, 2015)

A.4 Supercritical ORC

A supercritical ORC is in essence no different than a basic ORC, but it operates with supercritical fluids. This means that there is no actual evaporating happening in the ORC. Instead the fluid becomes super-critical, a state where distinct liquids and gases don't exist. Using a super-critical ORC can lead to improvements in thermal efficiency (Boz and Diez, 2017). However, the working fluid's critical temperature should be below that of the brine temperature. This means only certain working fluids can be used. The advantage of the supercritical ORC is that no evaporation process occurs. This yields a similar advantage as the dual-pressure system: The temperature of the geothermal fluid can be kept close to the temperature of the working fluid. resulting in less exergy loss.

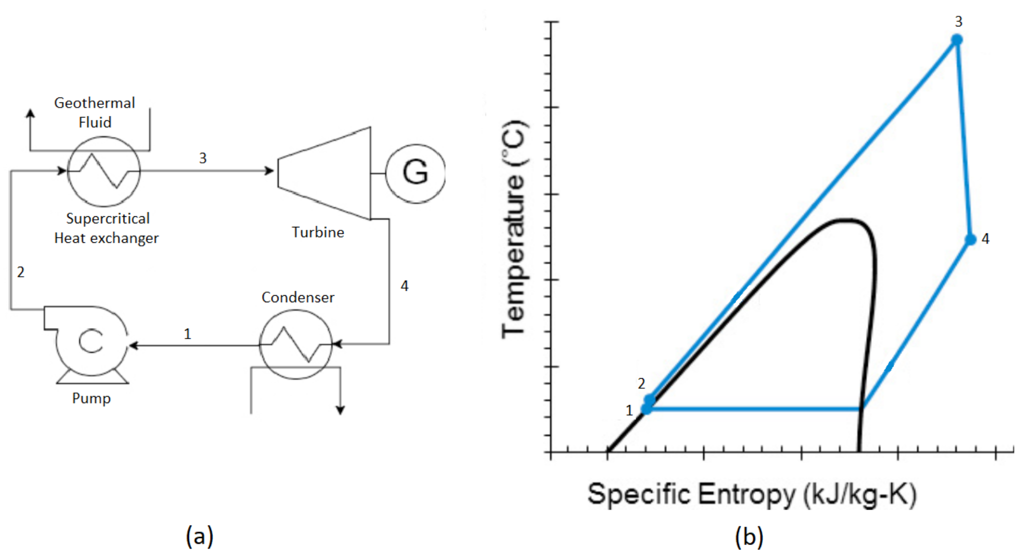

Figure A.5: Simplified lay-out of a regenerative super-critical ORC (a) and it's corresponding T-s-diagram (b)(Lai et al., 2010)

Appendix B

Optimization boundary conditions

B.1 Boundary conditions for the optimization of a basic subcritical ORC

1. P1: The lowest pressure in the cycle
   (a) P1 can't be lower than atmospheric pressure (10^5 Pa) to prevent in-leakage of ambient air.(Guillen and Zia, 2013)
   (b) The corresponding saturated liquid temperature can't be lower than 35°C, because it is cooled with ambient air.
2. P2: The highest pressure in the cycle
   (a) P2 can't be higher than the critical pressure of the working fluid.
   (b) P2 is always higher than P1.
3. T5: The highest temperature in the cycle (superheated vapour).
   (a) T5 can't be higher than the brine in going temperature T7.
   (b) T5 is always higher than or equal to T4.
   (c) The chosen T5 can not lead to a wet situation after the turbine.

B.2 Boundary conditions for the optimization of a regenerative subcritical ORC

1. P1: The lowest pressure in the cycle
   (a) P1 can't be lower than atmospheric pressure (10^5 Pa) to prevent in-leakage of ambient air.(Guillen and Zia, 2013)
   (b) The corresponding saturated liquid temperature can't be lower than 35°C, because it is cooled with ambient air.
2. P2: The highest pressure in the cycle
   (a) P2 can't be higher than the critical pressure of the working fluid.
   (b) P2 is always higher than P1.
3. T5: The highest temperature in the cycle (superheated vapour).
   (a) T5 can't be higher than the brine in-going temperature T7.

(b) T5 is always higher than or equal to T4.
(c) The chosen T5 can not lead to a wet situation after the turbine.

4. T12: The vapour temperature at the cold side of the regenerator
   (a) T12 can't be lower than T1+T_{pp}.

B.3 Boundary conditions for the optimization of a supercritical ORC

1. P1: The lowest pressure in the cycle
   (a) P1 can't be lower than atmospheric pressure (10^5 Pa) to prevent in-leakage of ambient air.(Guillen and Zia, 2013)
   (b) The corresponding saturated liquid temperature can't be lower than 35°C, because it is cooled with ambient air.
2. P2: The highest pressure in the cycle
   (a) P2 can't be lower than the critical pressure of the working fluid that is used. The system will not be supercritical if this would be the case.
   (b) P2 is always higher than P1.
3. T5: The highest temperature in the cycle (superheated vapour).
   (a) T5 can't be lower than the critical temperature of the working fluid that is used. The system will not be supercritical if this would be the case.
   (b) T5 can't be higher than the brine in going temperature T7..
   (c) The chosen T5 can not lead to a wet situation after the turbine.

B.4 Boundary conditions for the optimization of a regenerative supercritical ORC

1. P1: The lowest pressure in the cycle
   (a) P1 can't be lower than atmospheric pressure (10^5 Pa) to prevent in-leakage of ambient air.(Guillen and Zia, 2013)
   (b) The corresponding saturated liquid temperature can't be lower than 35°C, because it is cooled with ambient air.
2. P2: The highest pressure in the cycle
   (a) P2 can't be lower than the critical pressure of the working fluid that is used. The system will not be supercritical if this would be the case.
   (b) P2 is always higher than P1.
3. T5: The highest temperature in the cycle (superheated vapour).
   (a) T5 can't be lower than the critical temperature of the working fluid that is used. The system will not be supercritical if this would be the case.
   (b) T5 can't be higher than the brine in going temperature T7..
   (c) The chosen T5 can not lead to a wet situation after the turbine.
4. T12: The vapour temperature at the cold side of the regenerator
   (a) T12 can't be lower than T1+T_{pp}.

Appendix C

Potential Working fluid properties

Table C.1: Relevant fluid properties for all potential working fluids

	Working fluid	Boiling P. (K)	T crit. (K)	Max. Op. T T (K)	P crit (MPa)	Mol. weight (g/mol)	Type
1	1-Butene	266.7	419.29	525	4.01	56.11	Dry
2	Acetone	329.2	508	550	4.70	58.08	Wet
3	Ammonia	239.82	405.4	700	11.33	17.03	Wet
4	Carbonyl Sulfide	223.15	304.13	650	7.38	60.07	Wet
5	Cyclohexane	353.85	553.65	700	4.08	84.16	Dry
6	Cyclopentane	322.45	512.1	550	4.57	70.01	Dry
7	Dichloroethane	356.7	561.2	1000	5.38	98.96	Dry
8	DimethylCarbonate	363.15	557	600	4.9	90.08	Dry
9	Dimethyl Ether	249.0	401.0	525	5.4	46.07	Wet
10	Ethylene oxide	283.8	468.92	1000	7.33	44.05	Wet
11	HFE143m	-	377.92	420	3.64	100.04	isentr.
12	Isobutane	249.0	407.7	575	3.65	58.12	Dry
13	Isohexane	334.0	497.8	550	3.04	86.18	Dry
14	Isopentane	301.1	461.0	500	3.38	72.15	Dry
15	Neopentane	282.6	433.8	550	3.20	72.15	Dry
16	Novec649	322.15	442.15	500	1.88	316	Dry
17	Propylene	225.46	365.57	575	4.66	42.08	Wet
18	R11	296.87	471.06	625	4.39	137.37	isentr.
19	R113	320.75	487.21	525	3.39	187.40	Dry
20	R114	276.9	418.85	507	3.39	170.92	Dry
21	R115	235.2	353.1	550	3.13	154.46	Dry
22	R12	243.35	385.12	525	4.14	120.91	wet
23	R1233zd(E)	291.15	439.6	550	3.62	130.49	dry
24	R1234yf	243.15	367.85	410	3.38	114	dry
25	R1234ze(E)	254.15	382.52	420	3.64	114.04	isentr.
26	R1234ze(Z)	282.15	423.27	430	3.53	114.04	isentr.
27	R124	262.15	395.425	470	3.62	136.4	isentr.
28	R125	224.65	339.17	500	3.61	120.02	isentr.
29	R134a	246.85	374.21	455	4.06	103.3	isentr.
30	R13I1	-	396.44	420	3.95	195.91	isentr.

	Working fluid	Boiling P. (K)	T crit. (K)	Max. Op. T T (K)	P crit (MPa)	Mol. weight (g/mol)	Type
31	R141b	305.5	477.5	500	4.21	116.94	isentr.
32	R142b	263.55	410.26	470	4.06	100.49	isentr.
33	R143a	225.25	345.86	650	3.76	84.04	Wet
34	R152A	248.15	386.41	500	4.52	66.05	Wet
35	R161	236.05	375.25	450	5.01	48.06	Wet
36	R21	282.07	451.48	473	5.18	102.92	Wet
37	R218	309.85	345.02	440	2.64	188.02	Dry
38	R22	232.35	369.30	550	4.99	86.47	Wet
39	R227EA	256.80	374.90	475	2.92	170.03	Dry
40	R236EA	271.80	412.44	412	3.42	152.04	Dry
41	R236FA	271.71	398.07	400	3.20	152.04	Dry
42	R245ca	298.15	447.57	450	3.94	134.05	Dry
43	R245fa	288.4	427.01	440	3.65	134.05	Dry
44	R32	221.0	351.26	435	5.78	52.02	Wet
45	R365MFC	274.55	460	500	3.266	148.07	Dry
46	R40	249.3	416.3	730	6.67	50.49	Wet
47	R404A	221.35	345.27	500	3.73	97.6	Dry
48	R407C	229.55	359.34	500	4.63	86.2	Wet
49	R410A	224.6	344.49	500	4.90	72.6	Wet
50	R507A	220.35	343.77	500	3.70	98.9	Wet
51	RC318	267.15	388.38	623	2.78	200.03	Dry
52	SES36	308.75	450.70	725	2.85	184.53	Dry
53	Toluene	383.8	591.75	700	4.13	92.14	Dry
54	Water	373.15	647.10	2000	22.06	18.02	Wet
55	Cis-2-Butene	276.85	435.75	525	4.23	56.11	Dry
56	n-Butane	272.15	425.12	575	3.80	58.12	Dry
57	n-Heptane	371.6	540.13	600	2.74	100.21	Dry
58	n-Hexane	341.2	507.82	600	3.03	86.18	Dry
59	n-Pentane	309.2	469.7	600	3.37	72.15	Dry
60	n-Propane	231.15	369.89	650	4.25	44.10	Wet
61	Trans-2-Butene	274.50	425.61	625	4.03	56.11	Dry

Appendix D

Working Fluid Performance

Working fluid	Basic	Regenerative	Basic Critical	Regen. Critical
1-Butene	3.958	4.379	4.483	4.664
Acetone	2.309	2.367	-	-
Ammonia	3.715	3.758	2.827	2.880
Carbonyl Sulfide	3.321	3.690	4.080	4.240
Cyclohexane	1.473	1.528	-	-
Cyclopentane	2.694	2.757	-	-
Dichloroethane	1.331	1.368	-	-
Dimethyl Carbonate	1.177	1.219	-	-
Dimethyl Ether	3.764	4.108	4.445	4.552
Ethylene oxide	3.231	3.302	-	-
HFE143m	2.641	3.322	3.092	3.679
Isobutane	3.505	4.234	4.520	4.923
Isohexane	2.324	2.410	-	-
Isopentane	3.758	3.872	-	-
Neopentane	3.810	4.534	4.380	4.907
Novec649	2.616	3.081	2.672	3.163
Propylene	2.499	3.195	4.548	5.032
R11	3.344	3.407	-	-
R113	2.822	2.903	-	-
R114	3.396	4.056	3.661	4.172
R115	1.516	2.230	2.668	3.224
R12	2.985	3.495	3.681	4.004
R1233zd(E)	3.999	4.192	4.035	4.208
R1234yf	2.188	2.909	2.732	3.371
R1234ze(E)	2.707	3.401	3.073	3.692
R1234ze(Z)	3.828	4.180	-	-
R124	3.022	3.683	3.618	4.065
R125	1.083	1.680	2.724	3.247
R134a	2.580	3.217	3.446	4.058
R141b	3.409	3.479	-	-
R142b	3.661	4.059	4.033	4.271
R143a	1.542	2.280	3.504	4.022
R152A	3.311	3.788	4.221	4.436
R161	3.006	3.559	4.089	4.483
R21	3.286	3.347	-	-
R218	1.087	1.849	1.981	2.806
R22	2.332	3.243	3.732	4.109
R227EA	2.141	2.907	2.859	3.462

Working fluid	Basic	Regenerative	Basic Critical	Regen. Critical
R236EA	3.201	3.839	-	-
R236FA	2.878	3.384	-	-
R245ca	3.792	3.904	3.994	4.251
R245fa	3.755	4.231	3.812	4.295
R32	2.201	2.826	3.353	3.807
R365MFC	3.425	3.533	-	-
R40	3.621	3.680	3.634	3.685
R404A	1.441	2.094	3.217	3.731
R407C	2.023	2.718	3.387	3.857
R410A	1.662	2.318	3.475	4.009
R507A	1.377	2.089	3.180	3.694
RC318	2.358	3.165	2.818	3.471
SES36	3.484	3.656	-	-
Toluene	0.704	0.737	-	-
Water	0.619	0.763	-	-
Cis-2-Butene	4.233	4.431	4.194	4.252
n-Butane	3.947	4.479	4.494	4.748
n-Heptane	0.997	1.051	-	-
n-Hexane	1.954	2.029	-	-
n-Pentane	3.622	3.719	-	-
n-Propane	2.611	3.304	4.540	5.067
Trans-2-Butene	4.139	4.412	4.430	4.548

Appendix E

Pearsons Correlation Coeffecient

The strength of this linear relation is tested using the Pearson correlation coefficient. For the super-critical systems the Pearson correlation coefficients or "Pearsons r's" are determined using equation (E.1):

$$r = \frac{1}{n-1}\sum_{i=1}^{n}\left(\frac{x_i-\bar{x}}{s_x}\right)\left(\frac{y_i-\bar{x}}{s_y}\right) \tag{E.1}$$

with:
$n =$ The amount of trial working fluids
$x_i =$ Molar mass of working fluid i
$\bar{x} =$ the average value of the molar mass of all working fluids
$s_x =$ the standard deviation of the molar mass' of all working fluids
$y_i =$ The net work done by the system with working fluid i
$\bar{y} =$ the average value of work done by the system for all working fluids
$s_i =$ the standard deviation of work done by the system for all working fluids

The resulting values for Pearsons correlation coefficient for a linear relationship between the molecular mass and net work ouput on the molecular mass interval [40,200] can be seen in table E.1.

Table E.1: Pearsons r

	Linear relation
Super-critical ORC	-0.68
Regenerative super-critical ORC	-0.71

A relation with a correlation coefficient with an absolute value higher than 0.7 is considered as strong (LaerdStatistics, 2018). In this case, Pearsons r is equal or higher than or close to 0.7, which means that the correlation is (moderately) strong in both the non-regenerative and regenerative situation. The minus sign in front of the coefficient simply states that the relationship is a negative linear relationship.

Appendix F

ORC properties for co-generation

Here, all thermodynamic information can be found describing each ORC co-generation system.

F.1 HTCC co-generation for situation A

Maximum heat production

power produced/consumed:
Net Electric Power = 1.593 MW_e
Thermal Power = 26.455 MW_{th}
Pump Power required = 0.170 MW_e
Fan Power required = 0 MW_e

Brine Properties:
Brine mass flow = 50 kg/s
Brine ingoing Temperature = 200 °C
ORC brine outgoing T = 102.9 °C
Brine injection temperature = 70 °C

ORC properties:
Working fluid: n-butane
Working fluid mass flow= 66.78 kg/s
Condensation temperature = 95 °C
Evaporation pressure = 34.16 bar
Degree of superheating = 4.02 °C

Restricted heat production

power produced/consumed:
Net Electric Power = 1.591 MW_e
Thermal Power = 19.5 MW_{th}
Pump Power required = 0.170 MW_e
Fan Power required = 0.001 MW_e

Brine Properties:
Brine mass flow = 50 kg/s
Brine ingoing Temperature = 200 °C
ORC brine outgoing T = 102.9 °C
Brine injection temperature = 102.9 °C

ORC properties:
Working fluid: n-butane
Working fluid mass flow= 66.78 kg/s
Condensation temperature = 95 °C
Evaporation pressure = 34.416 bar
Degree of superheating = 4.02 °C

F.2 HTCC co-generation for situation B

Maximum heat production

power produced/consumed:
Net Electric Power = 1.137 MW_e
Thermal Power = 25.814 MW_{th}
Pump Power required = 0.130 MW_e
Fan Power required = 0 MW_e

Brine Properties:
Brine mass flow = 60 kg/s
Brine ingoing Temperature = 175 °C
ORC brine outgoing T = 116.24 °C
Brine injection temperature = 70 °C

ORC properties:
Working fluid: n-butane
Working fluid mass flow= 51.044 kg/s
Condensation temperature = 95 °C
Evaporation pressure = 34.416 bar
Degree of superheating = 1 °C

Restricted heat production

Power produced/consumed:
Net Electric Power = 1.137 MW_e
Thermal Power = 19.5 MW_{th}
Pump Power required = 0.170 MW_e
Fan Power required = 0 MW_e

Brine Properties:
Brine mass flow = 60 kg/s
Brine ingoing Temperature = 175 °C
ORC brine outgoing T = 116.24 °C
Brine injection temperature = 95.06 °C

ORC properties:
Working fluid: n-butane
Working fluid mass flow= 51.044 kg/s
Condensation temperature = 95 °C
Evaporation pressure = 34.16 bar
Degree of superheating = 1 °C

F.3 AHC co-generation for situation A

Maximum heat production

power produced/consumed:
Net Electric Power = 0.923 MW_e
Thermal Power = 27.011 MW_{th}
Pump Power required = 0.0638 MW_e
Fan Power required = 0 MW_e

Brine Properties:
Brine mass flow = 50 kg/s
Brine ingoing Temperature = 200 °C
ORC brine outgoing T = 161.05 °C
Brine injection temperature = 70 °C

ORC properties:
Working fluid: n-butane
Working fluid mass flow= 19.57 kg/s
Condensation temperature = 70 °C
Evaporation pressure = 34.16 bar
Degree of superheating = 17.6 °C

Restricted heat production

Power produced/consumed:
Net Electric Power = 1.705 MW_e
Thermal Power = 19.5 MW_{th}
Pump Power required = 0.157 MW_e
Fan Power required = 0.0284 MW_e

Brine Properties:
Brine mass flow = 50 kg/s
Brine ingoing Temperature = 200 °C
ORC brine outgoing T = 102.5 °C
Brine injection temperature = 90.5 °C

ORC properties:
Working fluid: n-butane
Working fluid mass flow= 58.27 kg/s
Condensation temperature = 90.5 °C
Evaporation pressure = 34.16 bar
Degree of superheating = 12.16 °C

F.4 AHC co-generation for situation B

Maximum heat production

power produced/consumed:
Net Electric Power = 0.886 MW_e
Thermal Power = 25.98 MW_{th}
Pump Power required = 0.0613 MW_e
Fan Power required = 0 MW_e

Brine Properties:
Brine mass flow = 60 kg/s
Brine ingoing Temperature = 175°C
ORC brine outgoing T = 143.30 °C
Brine injection temperature = 70 °C

ORC properties:
Working fluid: n-butane
Working fluid mass flow= 18.79 kg/s
Condensation temperature = 70 °C
Evaporation pressure = 34.16 bar
Degree of superheating = 17.6 °C

Restricted heat production

Power produced/consumed:
Net Electric Power = 1.483 MW_e
Thermal Power = 19.5 MW_{th}
Pump Power required = 0.149 MW_e
Fan Power required = 0.0300 MW_e

Brine Properties:
Brine mass flow = 60 kg/s
Brine ingoing Temperature = 175 °C
ORC brine outgoing T = 109.35 °C
Brine injection temperature = 82.65 °C

ORC properties:
Working fluid: n-butane
Working fluid mass flow= 51.04 kg/s
Condensation temperature = 83.15 °C
Evaporation pressure = 34.16 bar
Degree of superheating = 1 °C

F.5 DCC co-generation for situation A

Maximum heat production

power produced/consumed LTC/HTC:
Net Electric Power = 1.131/0.828 MW_e
Thermal Power = 13.757/12.337MW_{th}
Pump Power required = 0.102/0.088 MW_e
Fan Power required = 0/0 MW_e

Brine Properties LTC/HTC:
Brine mass flow = 24/26 kg/s
Brine ingoing Temperature = 200/200 °C
ORC brine outgoing T = 86.26/102.87 °C
Brine injection temperature = 70/70 °C

ORC properties LTC/HTC:
Working fluid: n-butane/n-butane
Working fluid mass flow= 33.38/34.73 kg/s
Condensation temperature = 77.97/95 °C
Evaporation pressure = 34.16/34.16 bar
Degree of superheating = 1.99/4.03 °C

DCC Restricted heat production

Power produced/consumed LTC/HTC:
Net Electric Power = 1.205/0.910 MW_e
Thermal Power = 5.462/14.022 MW_{th}
Pump Power required = 0.108/0.090 MW_e
Fan Power required = 0.074/0 MW_e

Brine Properties LTC/HTC:
Brine mass flow = 23.5/26.5 kg/s
Brine ingoing Temperature = 200/200 °C
ORC brine outgoing T = 79.16/102.87 °C
Brine injection temperature = 79.16/102.87 °C

ORC properties LTC/HTC:
Working fluid: n-butane/n-butane
Working fluid mass flow= 33.19/35.40 kg/s
Condensation temperature = 70.70/95 °C
Evaporation pressure = 34.16/34.16 bar
Degree of superheating = 1.29/4.03 °C

F.6 DCC co-generation for situation B

Maximum heat production

power produced/consumed LTC/HTC:
Net Electric Power = 0.811/0.592 MW_e
Thermal Power = 12.085/13.433 MW_{th}
Pump Power required = 0.075/0.067 MW_e
Fan Power required = 0/0 MW_e

Brine Properties LTC/HTC:
Brine mass flow = 28.74/31.26 kg/s
Brine ingoing Temperature = 175/175°C
ORC brine outgoing T = 106.39/116.24 °C
Brine injection temperature = 70/70 °C

ORC properties LTC/HTC:
Working fluid: n-butane/n-butane
Working fluid mass flow= 24.45/26.60 kg/s
Condensation temperature = 77.92/95 °C
Evaporation pressure = 34.16/34.16 bar
Degree of superheating = 1/1 °C

Restricted heat production

Power produced/consumed LTC/HTC:
Net Electric Power = 0.897/0.605 MW_e
Thermal Power = 13.75/5.73 MW_{th}
Pump Power required = 0.079/0.069 MW_e
Fan Power required = 0.897/0 MW_e

Brine Properties LTC/HTC:
Brine mass flow = 28/32 kg/s
Brine ingoing Temperature = 175/175 °C
ORC brine outgoing T = 100.74/116.24 °C
Brine injection temperature = 100.74/116.24 °C

ORC properties LTC/HTC:
Working fluid: n-butane/n-butane
Working fluid mass flow= 23.82/27.22 kg/s
Condensation temperature = 67.73/95 °C
Evaporation pressure = 34.16/34.16 bar
Degree of superheating = 1/1 °C

Appendix G

Matlab Code

All matlab files can be found by following the URL:
https://www.dropbox.com/sh/674dn0escbgw7rs/AABcS3xPUtDCgEubye4pxlTaa?dl=0

G.1 MatLab Code Thermodynamic Optimization

The matlab files required to run the optimization for the simple ORC systems can be found by following the URL:

https://www.dropbox.com/sh/zj6c5xksk75wh87/AACJCD1d544Nvf5rx8ZMsIdPa?dl=0

G.2 MatLab Code Optimization Co-Generation

The matlab files required for the optimization of the co-generation plants can be found by following the URL:

https://www.dropbox.com/sh/82948nc4wkbyjml/AABns94JkYTAdX8KWEF76XG8a?dl=0