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Solar thermal organic Rankine cycle (ORC)

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to obtain the academic degree of

Bachelor of Science in Engineering (BSc)

by

Ing. Patrick Schwarzbauer

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Thesis supervisor

Dr. Heather Dillon

I hereby declare that I prepared this work independently and without help from third parties, that I did not use sources other than the ones referenced and that I have indicated passages taken from those sources.

This thesis was not previously submitted in identical or similar form to any other examination board, nor was it published.

A handwritten signature in black ink, appearing to read 'Patrick Schwarzbauer'.

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Ing. Patrick Schwarzbauer
Portland (Oregon, USA), December 2017

Solar thermal organic Rankine cycle (ORC)

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ABSTRACT

Motivation: Using renewable sources of energy for generating electricity has become increasingly popular. Due to the effects of global warming and the climate change in general, people are more thoughtful with their decisions what kind of energy source they use for heating, electricity or even how their car is driven by. This work should demonstrate that renewable sources of energy, like solar power, are competitive in comparison to non-renewable sources, like oil or gas. Solar thermal energy generation using an Organic Rankine Cycle has applications where domestic hot water may be needed at the same time as power generation.

Results:

Based on the thermodynamic calculation the overall system efficiency is 7,361 %. The heat capacity of the plate heat exchanger (evaporator) is 13891 Watt, while the net power output of the turbine is 1022 Watt.

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1 INTRODUCTION

This thesis extends a 2015 senior design project of the University of Portland. The system is located on the green-roof of the Shiley School of Engineering. Main components include a flat plate thermal collector, an evacuated tube collector, and a converted turbine are already available and partly mounted. The former senior design team had some troubles with leakages in the water and the refrigerant cycle. The primary goal of this Bachelor thesis is to design a state of the art system to connect the piping in the refrigerant cycle to the hot water from the solar thermal collector. The piping and fittings should be pressure resistant, flexible in mounting and dismantling and as cheap as possible. A solar thermal organic Rankine cycle (ORC) is a thermodynamic process where a fluid is evaporated at a low temperature level by using solar energy. The superheated gas will drive a turbine and an electric generator to produce electricity by using heat energy. The pressure of the gas will decrease while passing the turbine. Afterwards the fluid goes into a heat exchanger where it's condensed, and then returns to the pump. "Compared with the steam Rankine cycle, the ORC is capable to realize more efficient expander at low power and show better thermodynamic performance at low temperature [1, page vii]." With solar thermal power generation, it is possible to meet the residential demand on heat and power. To increase the efficiency of an ORC process it is important to choose the optimal working fluid, which is way more difficult that it sounds in the first place.

The working fluid should match the thermophysical properties of the application as well as the chemical stability at the operating temperature. Typical working fluids are refrigerants (e.g. R123, R134a, R245fa) and hydrocarbons. Concentrating solar power (CSP) technologies such as parabolic trough, linear Fresnel reflector, heliostat tower and dish/engine systems are often used as solar collectors for higher temperature solar thermal systems [1, page 11]. The main focus of this project is the heat exchanger which evaporates the refrigerant. The technical plausibility of using a microchannel heat exchanger should be studied and evaluated. Schematics of the system, thermodynamic models (EES) and a working prototype should be provided at the end of this project.

2 THEORETICAL FOUNDATIONS

As mentioned in the introduction a solar thermal organic Rankine cycle (ORC) is a thermodynamic process where a fluid is evaporated at low temperature. For better understanding of the system and the theoretical background behind, it is necessary to design a schematic drawing. Figure 1 represents the schematic drawing of the system which is mounted on the green roof of the Shiley School of Engineering.

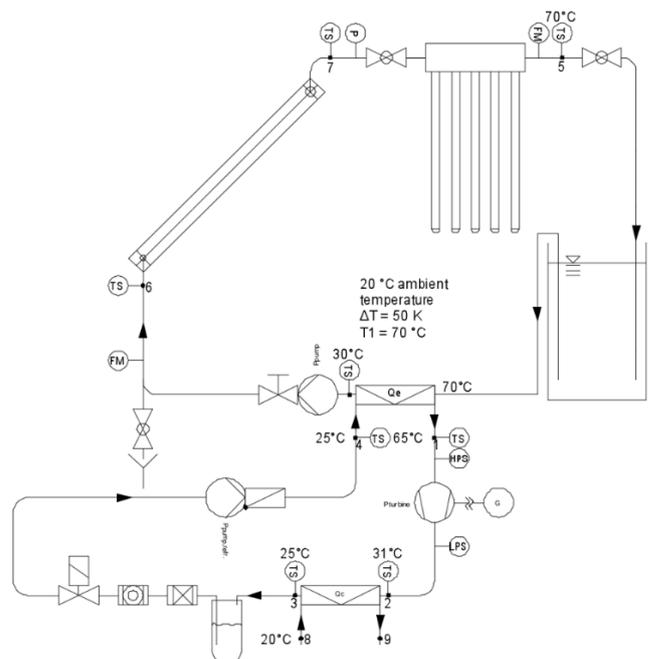


Figure 1. Schematic drawing ORC cycle and solar thermal collector.

The system consists of two cycles, the ORC cycle and the solar cycle. The refrigerant cycle is shown in the lower part of the schematic, see Figure 1. The upper part of the schematic drawing represents the solar water cycle. To keep the system as simple as possible, former senior design teams decided to keep the solar cycle open to ambient pressure and used water as the working fluid. The main components of the ORC cycle are shown in Figure 2 and Figure 3. The joining of the two systems using two different working fluids occurs at the heat exchanger labeled as Qe in Figure 1. The efficiency of this heat exchanger will determine the extent to which the ORC and solar thermal system are cost-effective at this small scale in the future.

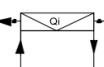
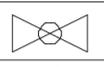
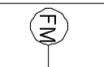
Design Table		
symbol	description	fluid
	Flat plate solar collector A=1,2 m ² Made by former senior design teams (University of Portland)	water
	Evacuated tube solar collector A=1,6 m ² Type: Made by former senior design teams (University of Portland)	
	Plastic water tank V= 30 liter	
	Counterflow heat exchangers for evaporating and condensing the refrigerant	
	Circulation pump. Type: Grundfos UP 26-96 F 60Hz Data sheet see Appendix A	
	Regulating valve for adjusting the flowrate of the solar cycle.	
	Standard ball valve.	
	Temperature sensors	
	Pressure gauge.	
	Flow meter.	

Figure 2. Symbols for solar water cycle.

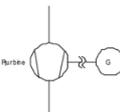
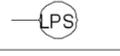
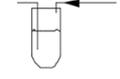
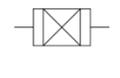
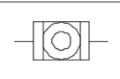
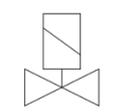
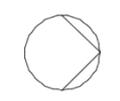
Design Table		
symbol	description	fluid
	Scroll compressor used as a turbine. Type: BITZER ECH209Y-02G	refrigerant R134a
	High pressure sensor.	
	Low pressure sensor.	
	Liquid refrigerant collector. Safety equipment for the pump. The pump should only carry liquid refrigerant to prevent damage.	
	Filter to protect the turbine and the pump against damage caused by rest material during construction.	
	Sight glass to check the phase of the fluid. It also indicates if there is water in the refrigerant cycle.	
	Solenoid valve for controlling the system.	
	Refrigerant pump to increase the pressure.	
	Non-return valve.	

Figure 3. Symbols for refrigerant cycle.

2.1 Thermodynamic cycle

Organic Rankine cycles are comparable to a steam Clausius-Rankine cycle. The main difference between both cycles is the type of fluid operating in the system. Water is typically used in a Clausius-Rankine cycle. Due to better thermodynamic performance at low temperature, refrigerants are used in organic Rankine cycles.

2.1.1 Clausius-Rankine cycle

The Clausius-Rankine cycle is a thermodynamic cycle which is named after the German physicist Rudolf Julius Emanuel Clausius and the Scottish engineer William John Macquorn Rankine. By using components like pumps, turbines, and heat exchangers it is possible to realize such a thermodynamic cycle. Even though it is not possible to reach the efficiency of a Carnot cycle, which is not explained in this thesis, the Clausius-Rankine cycle is a very important technical achievement. It describes the thermodynamic process which occurs in steam power plants. For better technical understanding of the cycle itself it is helpful to create a schematic drawing, therefore see Figure 4. The thermodynamic cycle can be shown in several ways, see Figure 5 and Figure 6. While Figure 5 represents a temperature-entropy diagram (T-s diagram), Figure 6 shows an enthalpy-entropy diagram (h-s diagram). Sometimes it is useful to use p-v diagrams. This typical diagram plots the change in pressure p with respect

to volume v . They are used to estimate the net work performed by a thermodynamic cycle. The area enclosed by the p - v curve represents the net work of the thermodynamic system [2].

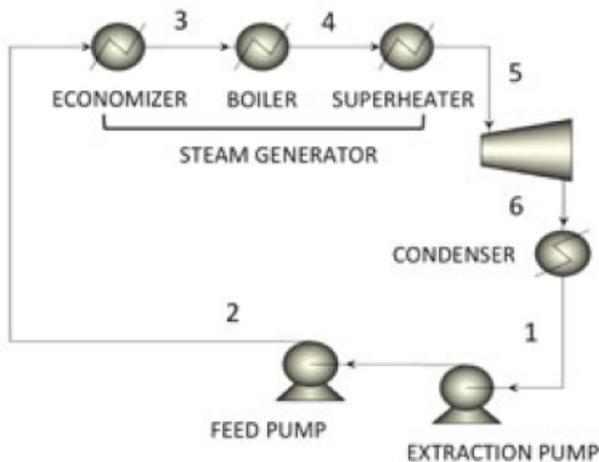


Figure 4. Schematic drawing of a typical Clausius-Rankine cycle [3, page 26].

In case of a Clausius-Rankine cycle, water is used as a working fluid. Water as a technical fluid offers several benefits, it's cheap in comparison to refrigerants, high specific heat capacity c_p ($c_p=4,18 \text{ J}/(\text{kg}\cdot\text{K})$) [3] and very resourceful. Besides all the benefits, using water as a fluid in thermodynamic cycles, also leads to some design limitations. The limitations are mostly due to the thermophysical properties of the fluid.

The thermodynamic cycle basically consists of several sections which are combined to one cycle [3]:

- 1-2: isentropic compression, $s=\text{const.}$ (one or more pumps)
- 2-3: isobaric heat supply to evaporation temperature, $p=\text{const.}$
- 3-4: isobaric evaporation, $p=\text{const.}$
- 4-5: isobaric overheating of the saturated steam to maximum cycle temperature, $p=\text{const.}$
- 5-6: isentropic expansion through steam turbine, $s=\text{const.}$
- 6-1: isobaric condensation to start conditions, $p=\text{const.}$

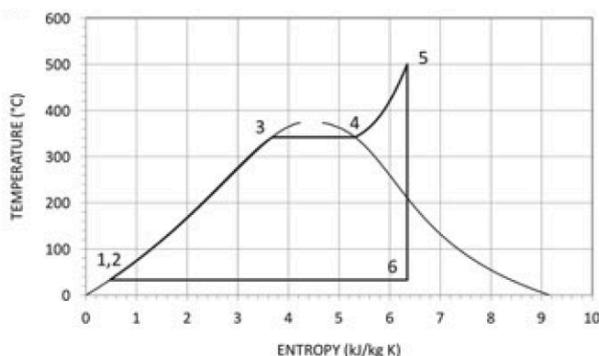


Figure 5. T-s diagram of a Clausius-Rankine cycle [3].

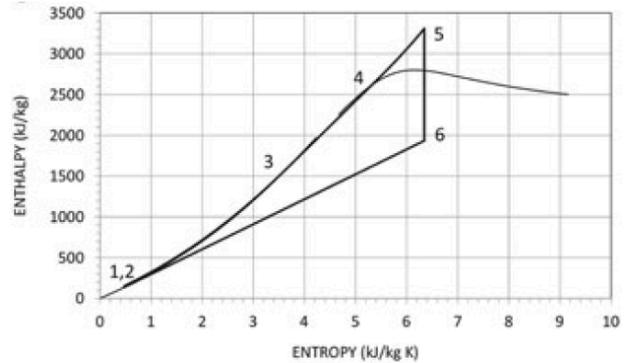


Figure 6. h-s diagram of a Clausius-Rankine cycle [3].

The efficiency of a Clausius-Rankine cycle can be calculated by using Equation 1.

$$\eta_{CR} = (\Delta w_P + \Delta w_T) / \Delta q_e \quad (1)$$

The specific compression work of the pump Δw_P is relatively small in comparison to the expansion work of the steam turbine Δw_T . Therefore, in many cases it is usual to neglect Δw_P . The heat supply Δq_e takes place from point 2 to 5, see Figure 4 and Figure 5. The equation can also be expressed by terms of enthalpy. Effective efficiencies of a real Clausius-Rankine include friction losses and pressure drops through heat exchangers are approximately 47%, see Figure 7 [3]. By increasing the evaporation pressure and temperature (3-5, see Figure 4) and decreasing the condensation pressure and temperature (6-1, see Figure 4) it is possible to increase the efficiency of a Clausius-Rankine cycle. However, there are some design limitations due to thermal stress of the material, which are not discussed in this thesis.

2.1.2 Organic-Rankine cycle

In general, an Organic-Rankine cycle (ORC) follows the same thermodynamic steps as a usual Clausius-Rankine cycle. The main difference between these two thermodynamic cycles is simply the temperature level where the working fluid evaporates and the working fluid itself. As mentioned in the previous chapter, see Figure 4, the evaporation temperature of a steam Rankine cycle is very high (3-5), up to 500°C and more, while the evaporation temperature of an ORC is very low (90-300°C) [4]. The components used in an ORC system are somewhat simpler than the components used in a steam Clausius-Rankine cycle. The three-phase evaporation can take place in one heat exchanger (HEX) instead of three, see Figure 4 [4]. The thermal stress of the material is much lower than in a steam Rankine-cycle. As a matter of fact, realizing an ORC system is cheaper than a steam Rankine system, however, the efficiency of an ORC is lower than in steam Rankine-cycles. The efficiency of an ORC system is typically 16%, which is very low in comparison to a steam Rankine cycle [4, Page 184]. Usually, ORC systems are used for industrial interests to recover the heat by using a renewable source of energy [3]. This paper should

discuss the plausibility of an ORC for small scale applications, like public housing.

Some applications for Organic-Rankine cycle are [4]:

- Biomass combined heat and power
- Geothermal energy
- Solar power plants
- Waste heat recovery

However, this research paper focuses on solar thermal applications only. The ORC cycle is shown in Figure 7. Solar radiation (1000 W/m²) heats up a heat transfer fluid (100-200°C), in this specific case the working fluid is R134a. The fluid then passes a heat exchanger, where the earned heat can be transferred to the ORC. This heat exchanger is called vaporizer or evaporator. The heat transfer fluid will be pumped back to the solar panels and the process starts again. The working fluid (organic fluid) of the ORC evaporates passing the vaporizer. The gas should be superheated to avoid erosion in the turbine. Heat energy can be converted into technical work and then into electricity. The turbine expands the working fluid isentropic (ideal) and begins to rotate. The rotation of the turbine drives a generator to produce electricity. The working fluid leaves the turbine with lower pressure. Afterwards the fluid will be condensed in a heat exchanger (e.g. air-cooled condenser, plate heat exchanger, tube heat exchanger). The liquid working fluid of the ORC will be pumped back to the vaporizer, and the cycle continues. There is also a type of system where the working fluid will be directly evaporated and superheated by passing the solar collectors. The unique part of this thesis is, that the turbine is a standard scroll compressor which is modified to act like a turbine. Bitzer, a German compressor manufacturer, is very interested in this project and donated the compressor for research purposes. The gas expands from a high-pressure level to a low-pressure level. As mentioned before, it is important that the gas at the exit of the turbine is still superheated or directly on the saturated vapor line. The quality of the steam “x” shouldn’t be less than one (x=1) to protect the turbine from erosion caused by liquid drops in the gas. The high velocity of the gas-liquid mixture will lead to major damage of the turbine. Both heat exchangers, evaporator and condenser, consists of two or three phase evaporation and condensation of the working fluid. The number of phases in each heat exchanger depends on following aspects:

- superheated or saturated gas (x=1) at the inlet of the condenser
- supercooled or saturated liquid (x=0) at the exit of the condenser
- supercooled or saturated liquid (x=0) at the inlet of the evaporator

The heat capacity of the superheated gas is low in comparison to the capacity of the saturated gas. This can be easily calculated by multiplying the mass flow rate with the enthalpy difference of each phase. For designing such systems, logp-h diagrams are very helpful. Capacities for evaporation and condensations can be calculated quickly

by using such diagrams. The liquefied refrigerant exiting the condenser will be collected in a liquid refrigerant collector, for schematic symbols and drawings see Figure 3 and Figure 7.

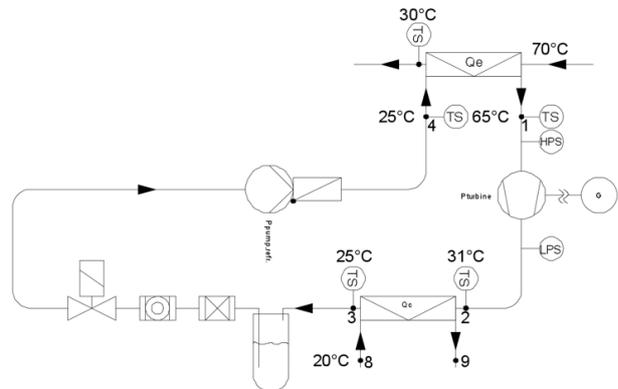


Figure 7. Schematic drawing of the ORC cycle.

The thermodynamic cycle is shown in Figure 8 and Figure 9. Both diagrams are generated with an Engineering Equation Solver (EES).

This software contains all important thermophysical properties of all kind of fluids as well as thermodynamic equations for solving difficult applications. After programming the code, it is easy to change values like temperatures and mass flow rates to analyze the system.

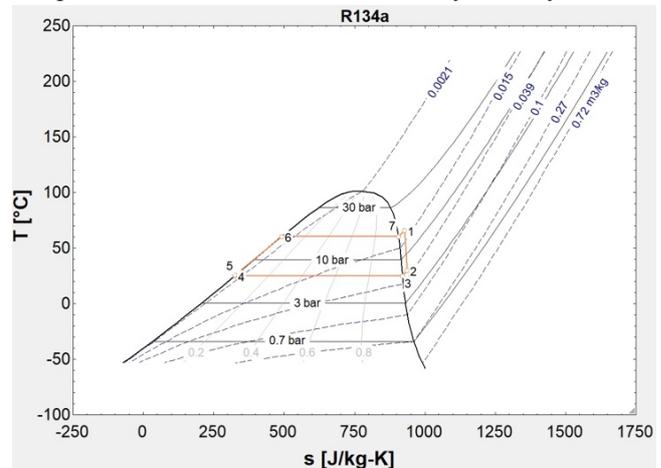


Figure 8. T-s diagram of the ORC cycle.

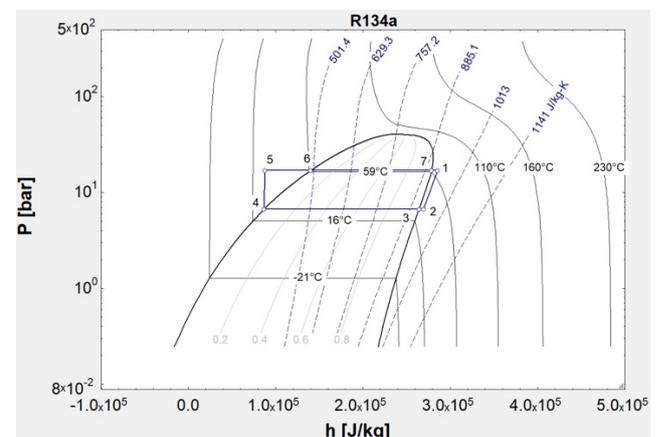


Figure 9. Logp-h diagram of the ORC cycle.

As mentioned earlier, an ORC cycle is similar to a steam Clausius-Rankine cycle, therefore compare Figure 5 and Figure 8. The temperatures of an ORC are very low in comparison to a regular steam system, see Figure 5. To realize an Organic-Rankine cycle several steps are necessary:

- 1-2: real expansion in turbine, $\eta_T = 0,80$ [3, Page 160]
- 2-3: isobaric heat dissipation in condenser, $p = \text{const.}$
- 3-4: isobaric condensation, $p = \text{const.}$
- 4-5: real compression in pump, $\eta_P = 0,75$ [3, Page 160]
- 5-6: isobaric heat supply, $p = \text{const.}$
- 6-7: isobaric evaporation, $p = \text{const.}$
- 7-1: isobaric heat supply, $p = \text{const.}$

Due to a very small enthalpy difference in section 4-5, the specific work Δw_p of the pump is mostly neglected. In small scale applications, the efficiency and performance of the pump is more important than in bigger applications, like big power plants. The state change from 4 to 5 happens with real conditions instead of isentropic conditions, see Figure 10.

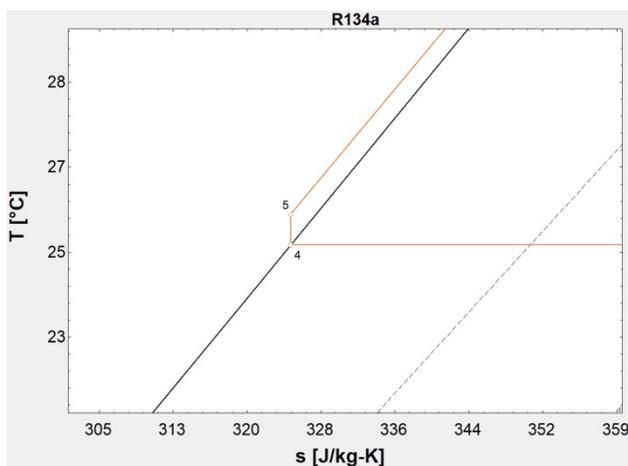


Figure 10. T-s diagram of the ORC cycle (State 4-5).

Furthermore, the design and the type of the evaporator Q_e should be analyzed. Due to low flow rates on the ORC side of the heat exchanger, the use of microchannel heat exchangers becomes more and more interesting. Microchannel heat exchangers are very small and have high heat transfer coefficients. Those heat exchangers are most likely used in cars as part of the A/C system.

2.1.3 Ideal cycle versus real cycle

Components like heat exchangers, valves, non-return valves, turbines, pumps, etc. occurs pressure drops. To keep the cycle most efficient it is important to reduce the pressure losses through such components. For pumps and turbines, the ideal thermodynamic process is isentropic. For real applications, this assumption isn't accurate enough, especially if the power and size of such components is increasing. Therefore, in many cases it is useful to calculate the pressure drop to keep the cycle

economically friendly. For heat exchangers, it is typical to consider isobaric conditions, which means no pressure drop of the fluid while passing the heat exchanger. In real life applications heat exchangers create pressure drops due to changes in direction of the fluid, and changes in velocity, see Figure 11. The black lines in Figure 11 represent the pressure drops in the heat exchangers. The pressure drop can be reduced by increasing the temperature difference through the heat exchanger, that leads to a decrease of the flow rate.

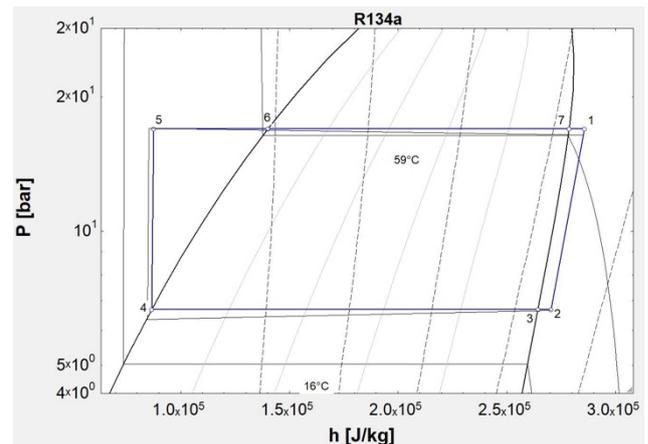


Figure 11. Logp-h diagram of the ORC cycle with pressure drops.

Even piping occurs pressure drops. Normally the pressure drops can be neglected if the length of the piping isn't significantly long.

3 THE REFRIGERANT

There are many different refrigerants which are capable for organic Rankine cycles. Table 1 shows different types of refrigerants including some fluid properties. Depending on the temperature which is available, different refrigerants are capable. This thesis is focusing on 1,1,1,2-Tetrafluoroethane (R134a). It's an inert gas used as a medium or high temperature refrigerant. Typical applications are [5]:

- Domestic refrigeration
- Commercial refrigeration
- Commercial refrigeration: Plug-ins & Vending machines
- Industrial refrigeration
- Transport refrigeration
- Residential and light air conditioning
- Industrial/commercial air conditioning DX chillers
- Industrial/commercial centrifugal compressors
- Mobile air conditioning

Table 2. ASHRAE Safety Classifications of Common Refrigerants [8]

Refrigerant Number	Toxicity Grouping	Flammability Number
R-11	A	1
R-12	A	1
R-22	A	1
R-123	B	1
R-124	A	1
R-125	A	1
R-134a	A	1
R-401A	A	1
R-406A	A	2
R-500	A	1
R-502	A	1
R-503	No rating listed	No rating listed
R-507A	A	1
R-717	B	2
R-744	A	1

4 COMPONENTS OF AN SOLAR ORC SYSTEM

In order to realize a thermodynamic cycle, several components are necessary. Main components like the turbine, flat plate collector, evacuated tube collector, solar circulation pump, ball valves of the solar cycle and flow meters, which are mounted in the solar cycle, are already incorporated. Also, an electrical control panel which is able to collect data like temperatures, pressures, flow rates, solar radiation and the rpm of the turbine is already available. For better understanding which components are necessary for a solar Organic-Rankine cycle see Figure 1, Figure 2 and Figure 3. A big issue of previous design teams was that the piping and the connection of each component wasn't designed and installed properly. As a result, many leaks were detected. In order to solve this problem, stainless steel pipes and Swageloks are used to prevent leakages. Swageloks are a metal to metal connection. Those fittings are able to handle high temperatures as well as high pressures and prevent leakages due to the metal to metal sealing.

4.1 Solar cycle

The incoming solar radiation (1000 W/m²) of the sun heats up a heat transfer fluid (100-200°C). The fluid then passes a heat exchanger, where the earned heat can be transferred to the ORC. This heat exchanger is called evaporator or vaporizer. The heat transfer fluid will be pumped back to the solar panels and the process starts again.

4.1.1 Solar collectors

Previous design teams built a flat plate and an evacuated tube collector, both collectors are connected in series to increase the outlet temperature of the heat transfer fluid, see Figure 1. Usually, the heat transfer fluid is water or a water/glycol mixture to prevent freezing. In this case the heat transfer fluid is just water. The design temperature difference of the inlet and outlet temperature of the collectors is 50 K. In order to reach this goal, both solar collectors should be orientated south. The angle of the both collectors is typically 30° to 45°. To reach a higher

efficiency of the evacuated tube collector, a flat mirror was placed underneath the tubes. The mirror reflects the incoming solar radiation back to the tubes to increase the efficiency of the solar collector, which leads to a higher overall temperature of the fluid and increases the temperature difference, see Figure 14. The flat plate solar collector is shown in Figure 13.

**Figure 13.** Flat plate solar collector.**Figure 14.** Evacuated tube solar collector.

Both solar collectors are mounted on the green roof of the Shiley School of Engineering (University of Portland).

4.1.2 Solar circulation pump

The solar circulation pump circulates the heat transfer fluid through the solar cycle. It is designed to reach the flow rate as well as to overcome the pressure losses of the system. The pump curve is shown in Appendix A. The pump is a standard circulation pump from Grundfos Type: UPS 26-96 F), see Figure 15. Especially for systems using renewable sources of energy, like the solar ORC, it is necessary to install high efficiency pumps in order to reach a high overall efficiency. The best way to select a pump which fits all criteria is to contact the supplier directly. Usually they have online software to select pumps. The operating point should be in the last third of the pump curve in order to reach a high efficiency.



Figure 15. Solar circulation pump.

4.1.3 Control panel

As mentioned, the control panel is able to control the ORC system, see Figure 16 and Figure 17. It collects data like temperatures, pressures, flow rates, solar radiation, rpm of the turbine and shuts off the system if necessary. The collected data is saved on a standard SD card, see Figure 18. The data logger was programmed with Arduino.



Figure 16. Control Panel.

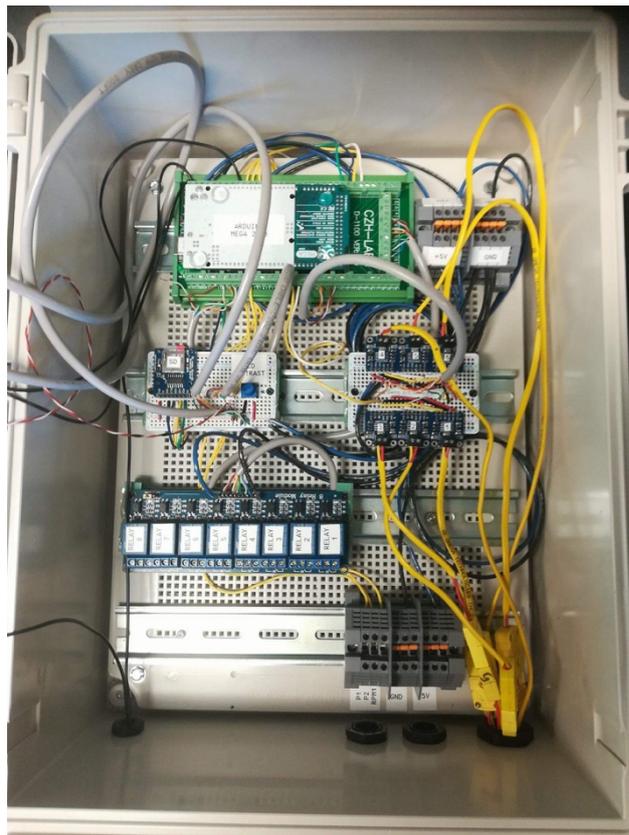


Figure 17. Control Panel.

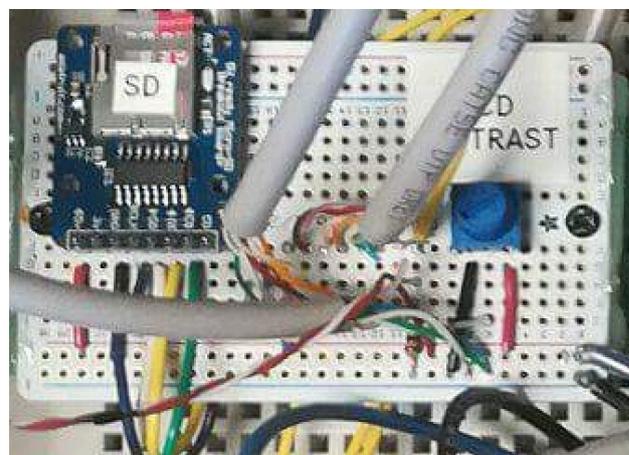


Figure 18. SD card for data collection.

4.2 Organic-Rankine Cycle

To generate electricity, it is necessary to evaporate the refrigerant to run the turbine. The turbine itself is a scroll compressor which is converted into a turbine. In order to do that some of the inner parts of the turbine are inverted. The working fluid (R134a) of the ORC evaporates passing the vaporizer. The gas should be superheated to avoid erosion in the turbine. Heat energy can be converted into technical work and then into electricity. The turbine expands the working fluid isentropic (ideal) and began to rotate. The rotation of the turbine drives a generator to produce electricity. The working fluid leaves the turbine with lower pressure. Afterwards the fluid will be

condensed in a heat exchanger (e.g. air-cooled condenser, plate heat exchanger, tube heat exchanger). The liquid working fluid of the ORC will be pumped back to the vaporizer and the cycle continues.

4.2.1 Turbine

As mentioned, the turbine is an inverted scroll compressor from Bitzer (Type: ECH209Y-02G), see Figure 19. The superheated gas (R134a) drives the turbine to generate electricity. The turbine was donated from Bitzer for research studies. Previous students than turned the compressor into a turbine. Due to the fact that the turbine was already available, several design decisions have to be considered in order to design the ORC. The flow rate of the ORC was limited by the minimum capable flow rate of the turbine. Therefore, the minimum flow rate of the turbine was set to 3 m³/h. On the other hand, that means it is possible that the available area of the collectors is too small. As a result, the collector area has to be increased.

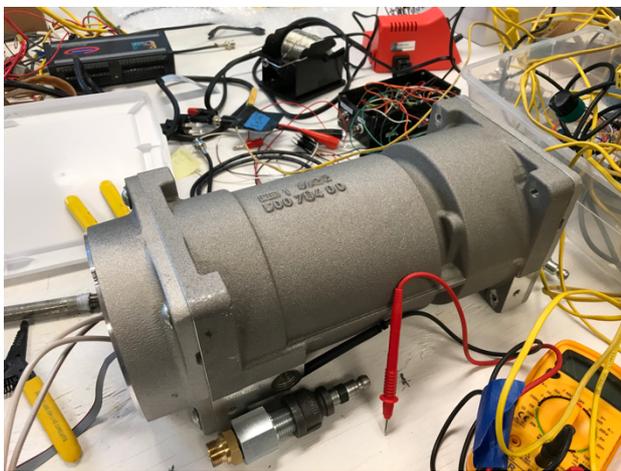


Figure 19. Turbine Bitzer.

4.2.2 Heat exchangers

To operate an ORC, it is necessary to evaporate and condensate the refrigerant for continuing the thermodynamic cycle. To avoid erosion inside the turbine the refrigerant should leave the evaporator in a superheated condition and vice versa it is necessary to make sure the refrigerant leaves the condenser as a liquid. Many different heat exchangers are available. In order to keep the system as compact as possible, plate heat exchangers or microscale heat exchangers can be used, see Figure 20.



Figure 20. Compact plate heat exchanger [9].

Usually, copper brazed plate heat exchangers or stainless-steel plate heat exchangers are used for refrigerant applications. For bigger systems, it is common to use two or more heat exchangers as an evaporator or condenser. The reason is to separate each phase in the heat exchangers to keep the whole system more compact and economical friendly. Also, it is easier to design each heat exchanger separately and to make sure that the refrigerant at the end of the evaporator is superheated. Also, the condenser can be split up in order to make sure the leaving refrigerant is liquid to protect the refrigerant pump from cavitation.

4.2.3 Refrigerant pump

The task of the refrigerant pump is to increase the pressure of the circulating fluid. It is also important to consider all pressure losses experienced in the system. The material of the pump as well as the sealing inside the pump must be compatible with the refrigerant. Standard sealing used in water pumps cannot be used for refrigerants. For large scale ORC systems, centrifugal pumps are used. A small-scale ORC system operates with a much lower flow rate than bigger systems, while the pressure increase is for both systems is the same. In order to meet all requirements, reciprocating positive replacement pumps can be used. Those pumps don't supply continuous flow, therefore it is necessary to install a small storage reservoir right after the pump. Plunger pumps can be used for small scale ORC systems, see Figure 21.



Figure 21. Plunger pump [10].

5 HEAT EXCHANGER DESIGN

5.1 Types of heat exchangers

Heat exchangers in general are used for transferring heat from a hot fluid to a cold fluid. They can be designed as parallel flow and counterflow as well as cross-flow and are classified to flow arrangement and type of construction [11], see Figure 23 and Figure 24. There are several heat exchanger types available [11]:

- Plate heat exchanger, see Figure 20
- Shell-and tube heat exchanger, see Figure 22
- Plate and shell heat exchanger
- Plate fin heat exchanger
- Microscale heat exchanger

The most common used heat exchangers for such applications are plate heat exchangers and shell-and-tube heat exchangers. In order to keep the heat exchanger as compact as possible counterflow heat exchangers can be used. In case of an evaporator operating in an ORC system it is important that the heat exchanger is able to change the phase of the refrigerant. The evaporator consists of three phases:

- preheating the refrigerant (state: 5-6)
- boiling the refrigerant (state: 6-7)
- superheating the gas (state: 7-1)

For better understanding of the phases see Figure 8 and Figure 9. One of the most common used heat exchanger is the shell-and-tube heat exchanger, see Figure 22. Those heat exchangers are easy to clean and they are also used in high pressure applications.

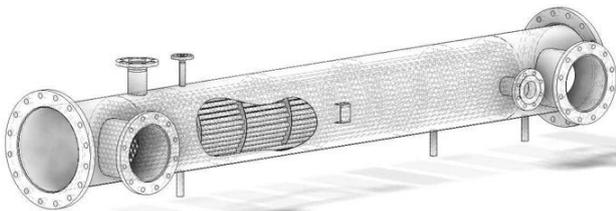


Figure 22. Shell-and-tube heat exchanger [12].

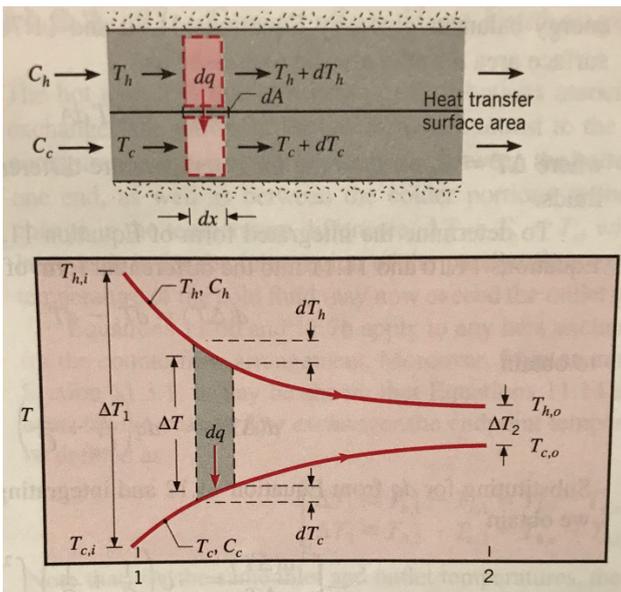


Figure 23. Temperature distributions for a parallel-flow heat exchanger [11].

The condenser instead consists of two phases. Like the evaporator the condenser should be able to cool the incoming superheated gas to condensing temperature as well as fully condensing the refrigerant. The steam quality of the exiting vapor should be $x = 0$. Sometimes it is common to subcool the exiting liquid to prevent the pump from cavitation. If a liquid refrigerant cooler is used, see Figure 3, it is not necessary to subcool the refrigerant.

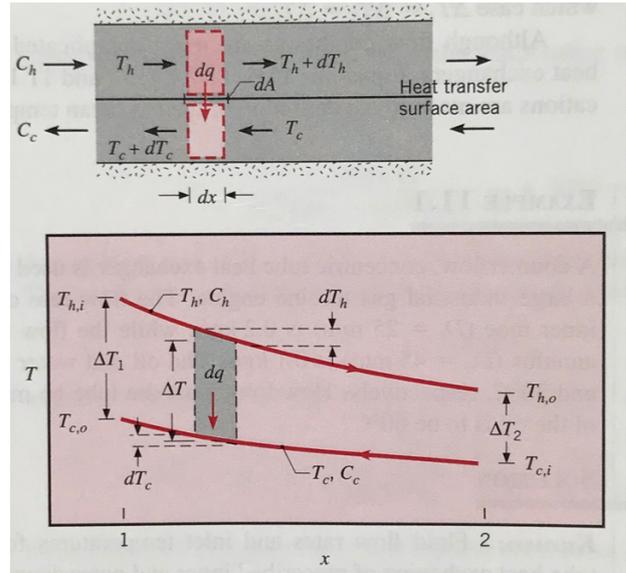


Figure 24. Temperature distributions for a counterflow heat exchanger [11].

5.1.1 Microscale heat exchangers

Microscale heat exchangers are basically like plate heat exchangers, see Figure 25. The main difference is the size of the hydraulic diameter. New technologies for microscale heat exchangers have hydraulic diameters in the range $Dh \leq 100 \mu m$ [11]. The main interest in the research of microscale heat exchangers are high heat transfer coefficients. The heat transfer coefficient is dramatic increasing if the hydraulic diameter decreases [11]. Flow rates of such heat exchangers are typically very low in comparison to other types. In case of an ORC system they do not suffice the design requirements. Another concern is the fact that the microscale heat exchanger should be able change the phase of the refrigerant. The specific volume will increase during evaporation. The size of the channels is too small to allow the fluid to expand. Another reason why this specific type of heat exchanger is not capable for an Organic-Rankine cycle is, that the heat capacity of the evaporator is approximately 14 kW in order to generate 1 kW electrical power.

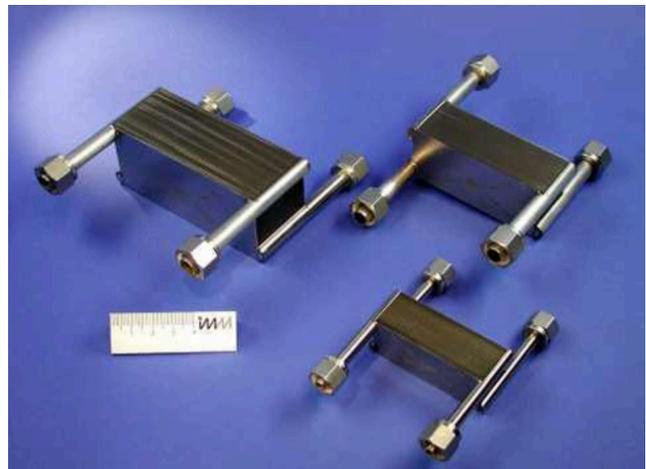


Figure 25. Counterflow microscale heat exchangers [12].

5.2 Theoretical foundations

There are two different ways to do the thermodynamic calculation on a HEX, the log mean temperature difference method (LMTD) or the NTU-method. If only the inlet temperatures are known, the use of the LMTD method requires a cumbersome iterative procedure [11]. Therefore, the use of the NTU-method would be more efficient. Since all temperatures and pressures are known, the LMTD calculation will be explained in detail. Another important factor of heat exchanger analysis is the overall heat transfer coefficient, see Equation 2. Sometimes it is difficult to predict the overall heat coefficient, especially when there is a change in phase of the fluid. It is highly recommended to check the result for the overall heat exchanger with representative values, therefore see Table 3. To approach realistic values the so-called fouling factor R_f is implemented in the equation. The value of the fouling factor strongly depends on the operating temperature, fluid velocity, and length of service of the heat exchanger [11]. In this calculation, the fouling factor will be neglected to keep the calculation simple, for representative values see Table 4.

$$\frac{1}{UA} = \frac{1}{h_i A_i} + \frac{R''_{f,i}}{A_i} + \ln\left(\frac{D_o}{D_i}\right) + \frac{R''_{f,o}}{A_o} + \frac{1}{h_o A_o} \quad (2)$$

The formulaic character for the overall heat coefficient is U. The formula also contains the Area A_i and A_o , both heat transfer coefficients h_i and h_o , fouling factors $R''_{f,i}$ and $R''_{f,o}$ as well as the diameters of the cylindrical wall. The calculation is based on cylindrical cross areas, where the subscript i means inside the tube and o means outside the tube. For other geometries it necessary to calculate the hydraulic diameter of the cross-section.

Table 3. Representative values of the overall heat coefficient U [11].

Fluid Combination	U [W/(m ² *K)]
Water to water	850 - 1700
Water to oil	110 - 350
Steam condenser (water in tubes)	1000 - 6000
Ammonia condenser (water in tubes)	800 - 1400
Finned-tube heat exchanger (water in tubes, air in cross flow)	25 - 50

Table 4. Representative fouling factors [11].

Fluid	R''_f [(m ² *K/W)]
Seawater and treated boiler feedwater (below 50 °C)	0,0001
Seawater and treated boiler feedwater (above 50 °C)	0,0002
River water (below 50 °C)	0,0002 - 0,001
Fuel oil	0,0009
Refrigerant liquids	0,0002

5.2.1 The Log Mean Temperature Difference (LMTD)

For heat exchanger analysis, it is essential to define temperatures, pressures as well as mass flow rates. The more values are known the better and easier the calculation is going to be. Another important value to predict the performance of a heat exchanger is enthalpy h. With the mass flow rate, which is constant, and the enthalpy difference at the inlet and outlet, the heat transfer rate Q can be calculated, see Equation 3. The specific heat capacity c_p at constant pressure is typically 4,180 [kJ/(kg*K)] for liquid water [11]. The inlet and outlet temperature of the hot fluid are calculated in Kelvin, in order to satisfy the equation. The mass flow rate m_h [kg/s] of the solar cycle was defined by previous senior design teams to reach the highest outlet temperature of the solar collectors.

$$Q = m_h * (h_i - h_o) = c_{p_h} * m_h * (T_{h,i} - T_{h,o}) \quad (3)$$

The heat transfer rate is related on the hot fluid, due to the laws of thermodynamics heat can only be transferred from a hot medium to a cold medium and not vice versa. For the ORC system, the hot fluid is the incoming solar water which evaporates and superheats the refrigerant before it enters the turbine. In case of condensing the refrigerant, the hot water is tap water of the City of Portland. The knowledge of the minimum and maximum temperature of the tap water is important for calculating the condenser as well as the overall performance of the ORC itself. Some systems use finned-tube heat exchangers, where the refrigerant is in the tubes and the air is in cross flow, see Figure 26. Due to a low overall heat transfer coefficient of such heat exchangers, the area has to be higher than the area of plate heat exchangers or other types.



Figure 26. Fin-tube heat exchanger [9].

With the knowledge of the predicted heat capacity Q of the heat exchanger, the total area A can be calculated by using Equation 4.

$$Q = U * A * \Delta\vartheta m \tag{4}$$

As a first step, a rough value for the area A can be calculated by using Table 3. For more accurate values it is necessary to calculate the overall heat capacity for each phase of the heat exchanger. Since the heat exchanger consists of three different phases, three values for U are needed, see Figure 27. The mean logarithmic temperature $\Delta\vartheta m$ of the heat exchanger can be determined by using Equation 5 and Figure 23 for counterflow heat exchangers.

$$\Delta\vartheta m = (\Delta T_1 - \Delta T_2) / \ln(\Delta T_1 / \Delta T_2) \tag{5}$$

In terms of temperature differences, Equation 6 and 7 can be used.

$$\Delta T_1 = T_{h,i} - T_{c,o} \tag{6}$$

$$\Delta T_2 = T_{h,o} - T_{c,i} \tag{7}$$

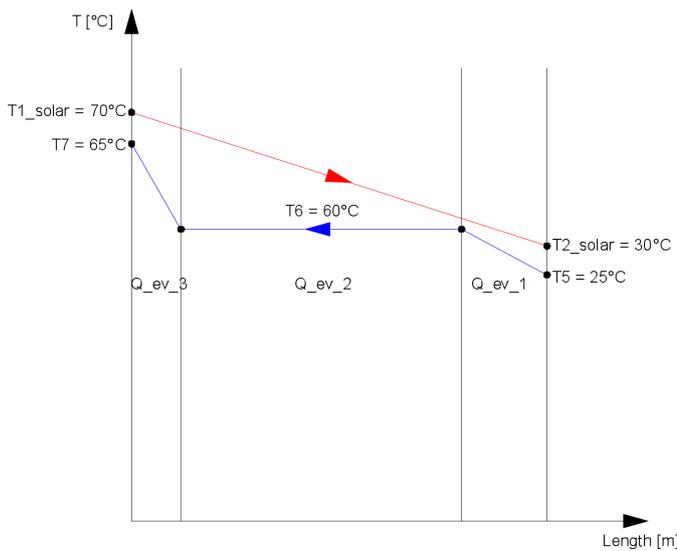


Figure 27. Temperature distributions for a counterflow heat exchanger.

5.3 Heat exchanger design with Matlab

The thermodynamic cycle was calculated in EES, see Appendix B for code. To design the heat exchangers a few assumptions were made:

- Isobaric heat transfer through all three stages
- Laminar flow
- Ambient pressure for solar cycle ($p_{amb} = 1013$ mbar)
- No pressure drops in piping or valves
- Pressures, temperatures and flow rates as well as mass flow rates are known from the senior design team projects.

- The overall convection coefficient U for boiling is assumed to be $U = 2000$ W/(m²*K).

As mentioned the heat exchanger is operating as an evaporator, that means that the incoming liquid (R134a) is preheated to evaporation temperature (60 °C). Afterwards the refrigerant evaporates until the vapor quality is $x = 1$. To prevent erosion of turbine components, the saturated gas should be superheated. In this case the outlet temperature of the superheated gas is 65 °C. The thermodynamic analysis of the heat exchanger is calculated in Matlab. Because of the known pressure and temperatures, the LMTD method seems to fit perfectly. To calculate the HEX, it is necessary to generate an excel sheet including following data:

- Pressure [p]=bar
- Temperature [T]=K
- Specific enthalpy [h]=J/kg

It is helpful to create pros and cons lists as well as a table with design criteria in order to decide the right type of heat exchanger, Table 5 lists a few advantages and disadvantages of plate heat exchangers. For design criteria see Table 6.

Table 5. Pros and cons of plate heat exchangers.

Pros	Cons
Operate 0.1 to 2.5 MPa Operate -40 to 260 C Effectiveness up to 93%	Cannot accommodate high pressures, temperatures and high pressure, temperature differentials
Size T = 0.5 to 1.2 mm W = 20 to 1200 mm L = 0.6 to 5 m	Mainly used for liquid to liquid heat exchange
High turbulence due to plates reduces fouling effect	Not suitable for erosive duties or fluids containing fibrous materials
Surface are 1/2 to 1/3 that of shell tube exchanger	Viscous fluids lead to maldistribution problems when cooling
Cost, space reductions to shell and tube	Max pressure 2.5 MPA but usually 1 MPA Max temperature 260 C but usually 150 C
Leakage occurs only if hole is developed	Pressure drop across exchanger is higher than shell and tube
Held up volumes of fluid is small	
High thermal performance in counter flow arrangements make temperature approaches up to 1 C	
Flow induced vibration, noise, thermal stresses, and entry issues do not exist	
Special plates are available for 2 phase condensation.	
Flow velocities and plate lengths are small, so pressure drop is generally acceptable	

Table 6. Design criteria for evaporator.

#	Criterion	Priority	Description
1	Safety	Essential	Must hold pressure of working fluid.
2	Size	Essential	Must fit on the Shiley roof and through a doorway, so the heat exchanger must be compact.
3	Operational	Essential	Must superheat the working fluid 5K over saturated temperature to avoid erosion.
4	Durability	High	Fouling issues need to be negligible for reliable use of the evaporator in future research and labs.
5	Durability of ORC	High	A evaporator that can operate in harsh weather is wanted for future operations on the Shiley roof.
6	Economy	High	Maximum of \$300 is preferred for budget considerations.

The first columns of the code consist of basic information and values, like incoming solar collector temperature $T1_solar$, outlet solar collector temperature $T2_solar$, specific heat capacity of water cp_w as well as the mass flow rate $mdot_solar$. With this information, it is possible

to calculate the generated heat capacity Q_{solar} [W] of the solar collectors, see Equation 8.

$$Q_{\text{solar}} = \dot{m}_{\text{solar}} \cdot c_{p,w} \cdot (T_{2,\text{solar}} - T_{1,\text{solar}}) \quad (8)$$

With the knowledge of the generated heat capacity from Equation 8 it is possible to start the calculation of the refrigerant cycle. Like before it is necessary to define all known information of the heat exchanger, in this case the outlet temperature of the evaporator $T_{2,\text{ev}}$ and the evaporation pressure p_{ev} . The next step is to implement a code to interpolate the values for enthalpy which are generated with EES and exported to an excel sheet called "PropertiesR134a.xlsx". To import the data from excel we can use the Matlab command:

```
>>Table_h = xlsread('PropertiesR134a.xlsx')
```

To realize an Organic-Rankine cycle several steps are necessary:

- 1-2: real expansion in turbine, $\eta_T = 0,80$ [3, page 26]
- 2-3: isobaric heat dissipation in condenser, $p = \text{const}$.
- 3-4: isobaric condensation, $p = \text{const}$.
- 4-5: real compression in pump, $\eta_P = 0,75$ [3, page 26]
- 5-6: isobaric heat supply, $p = \text{const}$.
- 6-7: isobaric evaporation, $p = \text{const}$.
- 7-1: isobaric heat supply, $p = \text{const}$.

To find the enthalpy of condition 1, which represents the superheated gas (R134a) following code can be used:

```
>>T2_ev_deg=65;
>>T2_ev=T2_ev_deg+273.15;
>>p_ev=17*10.^5;
%Import excel data for Enthalpy
>>Table_h = xlsread('PropertiesR134a.xlsx')
>>find p_ev=17
>>find T2_ev_deg=65
>>sl = 2; %size of the square input matrix
>>t1 = Table_h(:,2); %Capture the table headers
>>t = t1(1:length(t1)); %fix the first NaN value
>>t = repmat(t,1,sl-1); %make the headers a matrix that repeats
>>t = t(:)'; %creates a vector out of matrix
>>p1 = Table_h(:,1); %Capture the labels for each row
>>p = p1(1:length(p1))';%fix the first NaN value
>>p = repmat(p,1,sl-1); %make the headers a matrix that repeats
>>h = Table_h(:,3); %Just the enthalpy values
%turn the matrix of h values into a long vector
>>h = h(:)'; %creates a vector out of matrix
```

% Now we have 3 long vectors that contain pressure, temperature and enthalpy. to find the closest point to our target values we can use minimization of residuals.

```
>>p_find=17; %pressure of superheated gas
```

```
>>T_find=65; %temperature of superheated gas
>>r=(p-repmat(p_find,1,length(p))).^2+(t-repmat(T_find,1,length(t))).^2;
```

% Find the smallest residual

```
>> [m,i] = min(r);
```

%the index, i, now represents the point closest to the value we are trying to look up.

```
>>h1 = interp1([p(i) p(i+1)],[h(i) h(i+1)],p_find,'linear')
```

This procedure can be done for every condition of the ORC cycle, see Table 7.

Table 7. Key thermodynamic states during evaporation.

State #	State Description	h J/kg	P bar	ρ kg/m ³	s J/(kg*K)	T °C	v m ³ /kg	x
5	Inlet of Evaporator	87443	17	1210	324.9	25.7	0.0008263	
6	Begin evaporation	140062	17	1051	491.3	60.43	0.0009518	0
7	Evaporation	278570	17	88.51	906.4	60.43	0.0113	1
1	Outlet evaporator	285665	17	84.12	927.6	65.7	0.01189	

Within all the values for enthalpy, all three heat capacities for all phases of the HEX can be calculated. The command `repmat` has following definition:

"B = repmat(A,n) returns an array containing n copies of A in the row and column dimensions. The size of B is size(A)*n when A is a matrix" [13].

For interpolating between values, the command `interp1` can be used with following description:

"vq = interp1(x,v,xq) returns interpolated values of a 1-D function at specific query points using linear interpolation. Vector x contains the sample points, and v contains the corresponding values, v(x). Vector xq contains the coordinates of the query points.

If you have multiple sets of data that are sampled at the same point coordinates, then you can pass v as an array. Each column of array v contains a different set of 1-D sample values" [13].

Now it is possible to calculate the dimensions, numbers of gaps, areas of each phase as well as the Reynold numbers, Nusselt numbers and the overall convection coefficients U for each phase, see Appendix B for thermodynamic calculation. To simplify the code, an assumption for the overall convection coefficient of the boiling phase is made. During this phase change a lot of knowledge of heat transfer is necessary. To make sure the area is big enough to evaporate the refrigerant the convection coefficient is defined very low ($U_2 = 2000$ [W/(m²*K)]). As a design decision, the exterior dimensions are quadratic ($L = 0.2$ m). This results in several solutions for number of gaps as well as the gap width "a" for each phase, see Appendix B. For superheating the gas, laminar flow is assumed. This assumption is true for water but not for the superheated refrigerant. Therefore, it is necessary to calculate a new Nusselt number for turbulent flow, see Equation 9 [11, page 485].

$$NuD_{rf_3} = \left[\frac{(f/8) \cdot (ReD_{rf_3} - 1000) \cdot Prm_{rf_3}}{(1 + 12.7 \cdot (f/8)^{0.5}) \cdot (Prm_{rf_3}^{2/3} - 1)} \right] \tag{9}$$

With the new Nusselt number a new area $A3_{new}$, gap width $a3_{new}$ and a new overall heat coefficient $U3_{new}$ can be calculated. The last step is to sum up all the areas, numbers of gates as well as the gap widths. The calculation of the heat exchanger leads to following solution:

- Total heat capacity of the evaporator $Q_{ev} = 13891 \text{ W}$
- Total area $A_{total} = 1,8427 \text{ m}^2$
- Exterior dimensions (quadratic) $L = 0,2 \text{ m}$
- Total width $B_{total} = 0,6177 \text{ m}$

To reduce the width, the number of gaps can be increased, see Figure 28. The exterior dimensions of the heat exchanger seem very big in comparison to the heat capacity Q_{ev} . The reason of this phenomenon is the boiling phase of the heat exchanger. The assumption of the convection coefficient is very low, see Table 3 to compare. There can also be errors in Nusselt numbers and fluid properties like density, viscosities or thermal conductivity. The results of the calculation are shown in Table 8 and Table 9 for English units.

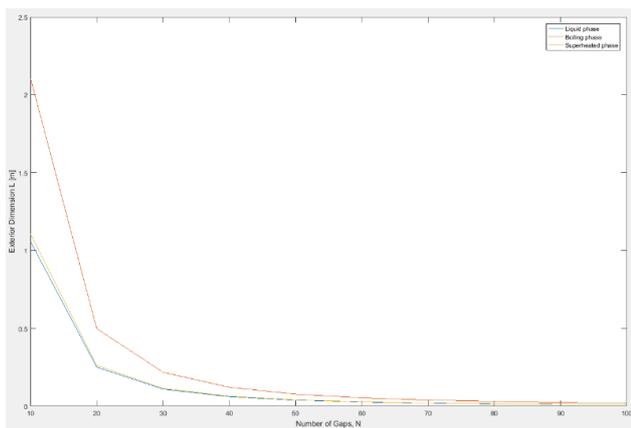


Figure 28. Dimensions of the plate heat exchanger.

Table 8. Evaporator conditions (SI units).

Working Fluid	R134a	Heating Fluid	Water
In Temp.	25.7 C	In Temp.	70 C
Evaporation Temp.	60.43 C	Out Temp.	30 C
Out Temp.	65.7 C		
In Pressure	17 bar (abs)		
Out Pressure	17 – 0.344 bar		
Flow	0.0701 kg/s	Flow	0.0063 kg/s
Max Pressure Drop	344 mbar	Max Pressure Drop	690 mbar

Table 9. Evaporator conditions (English units).

Working Fluid	R 134a	Heating Fluid	Water
In Temp.	78.3 F	In Temp.	158 F
Evaporation Temp.	140.7 F	Out Temp.	86 F
Out Temp.	150.3 F		
In Pressure	250 psia		
Out Pressure	250 – 5 psia		
Flow	0.155 lbm/s 558 lbm/h	Flow	0.01389 lbm/s 50 lbm/h
Max Pressure Drop	5 psi	Max Pressure Drop	10 psi

6 IMPROVEMENTS AND RESULTS

Typical solar systems are usually pressurized systems to prevent freezing of the fluid, corrosion, evaporation of the water, and to guarantee a safe operation of the system. Currently the system is open to the ambient, therefore people have to check the water level in the tank to make sure the solar circulation pump is supplied with enough water. Another decrease in efficiency of the system is the fact that air can be inside the collectors. Trapped air in a heat exchanging device acts like insulation, which leads to a lower heat capacity as well as a lower outlet temperature of the solar collectors. Now the system is not able to operate automatically. If an automatically controlled system is desired it is highly recommended to design a pressurized solar cycle, see Figure 29 for an example of a schematic drawing for pressurized systems.

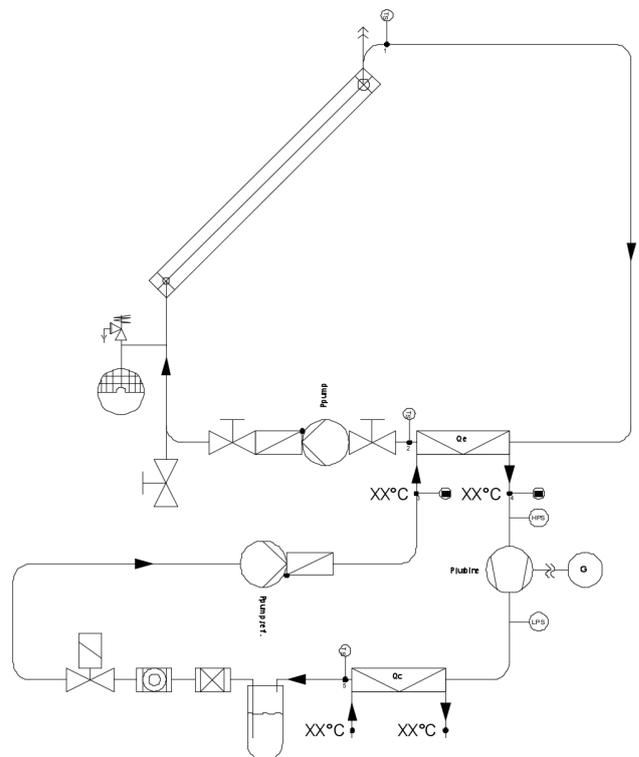


Figure 29. Pressurized solar Organic-Rankine cycle.

To guarantee a safe operation of the solar system it is necessary to install a safety valve as well as an expansion vessel. The expansion vessel can compensate in case of volume expansion. It also determines the pressure in the solar cycle. Another task of the expansion vessel is if no electricity is needed and the solar radiation is still high, the liquid can evaporate in the collectors. The solar cycle is then in a so-called stagnation phase. In order to make sure that the system can operate when needed, the expansion vessel is able to cool and liquefy the steam. Usually, the amount of water after condensing will be same in the system. In case of leakages, an automatic refilling system can be installed. The use of solenoid valve and a pressure transmitter will provide the right amount of water if needed. Typical, two ball valves and a strainer before the

solenoid valve are implemented to prevent the solenoid valve for damage. The ball valves are in case for maintaining the system. The control panel should be able to convert the incoming data from the pressure transmitter and generate an output signal to the solenoid valve. The efficiency of ORC systems in general is very low, approximately 15% system efficiency. Due to the fact that solar radiation is free, such systems definitely have future potential, especially for waste heat applications. The thermodynamic calculation of the system with EES reached an overall efficiency of 7,361 %. To increase the efficiency, it is necessary to install more solar collectors. The best orientation for the collectors should be analyzed, as well as the right angle. Piping and bigger tanks should be insulated to decrease heat losses. To prevent leakage of the ORC special pipe fittings and connections are needed (Swagelok). Another important aspect to consider is pipe-laying. The refrigerant carries small amounts of oil through the system. It is necessary to make sure the oil can return to the turbine to prevent bearing damage and a proper function of the turbine. The results of the thermodynamic calculation are shown in Table 10. The whole code is attached in Appendix B. As shown in Table 10 the heat capacities for the evaporator and the condenser are very high in comparison to the electric output of the turbine. These results demonstrate the complexity of renewable sources of energy in order to be economical arguable. The annuity of such systems is way longer than common systems to produce electricity. But as mentioned before, if waste heat is available such systems can support the existing system to decrease the operating costs. If several renewable sources are available the existing system, like gas turbines for example, can be displaced step by step. Another train of thought is to evaporate the refrigerant directly in the collectors. Therefore, no solar pump is needed and the system efficiency can be increased.

Table 10. Results of the thermodynamic calculation with EES.

Results of the solar Organic-Rankine system		
Description	Value	Unit
Mass flow rate refrigerant	0,07008	kg/s
Mass flow rate of the solar cycle	0,0063	kg/s
Heat capacity Evaporator	13891	W
Heat capacity Condenser	12868	W
Net power of the turbine	1022	W
Pump power	59,83	W
Outlet temperature of solar collectors (water)	70	°C

Inlet temperature of solar collectors (water)	30,5	°C
Inlet temperature of the evaporator (R134a)	25	°C
Outlet temperature of the evaporator (R134a)	65	°C
Inlet temperature of the condenser (R134a)	30	°C
Inlet temperature of the condenser (R134a)	30	°C
Inlet temperature of the condenser (water)	20	°C
Outlet temperature of the condenser (water)	6,07	°C
System efficiency η	7,361	%

7 ACKNOWLEDGEMENTS

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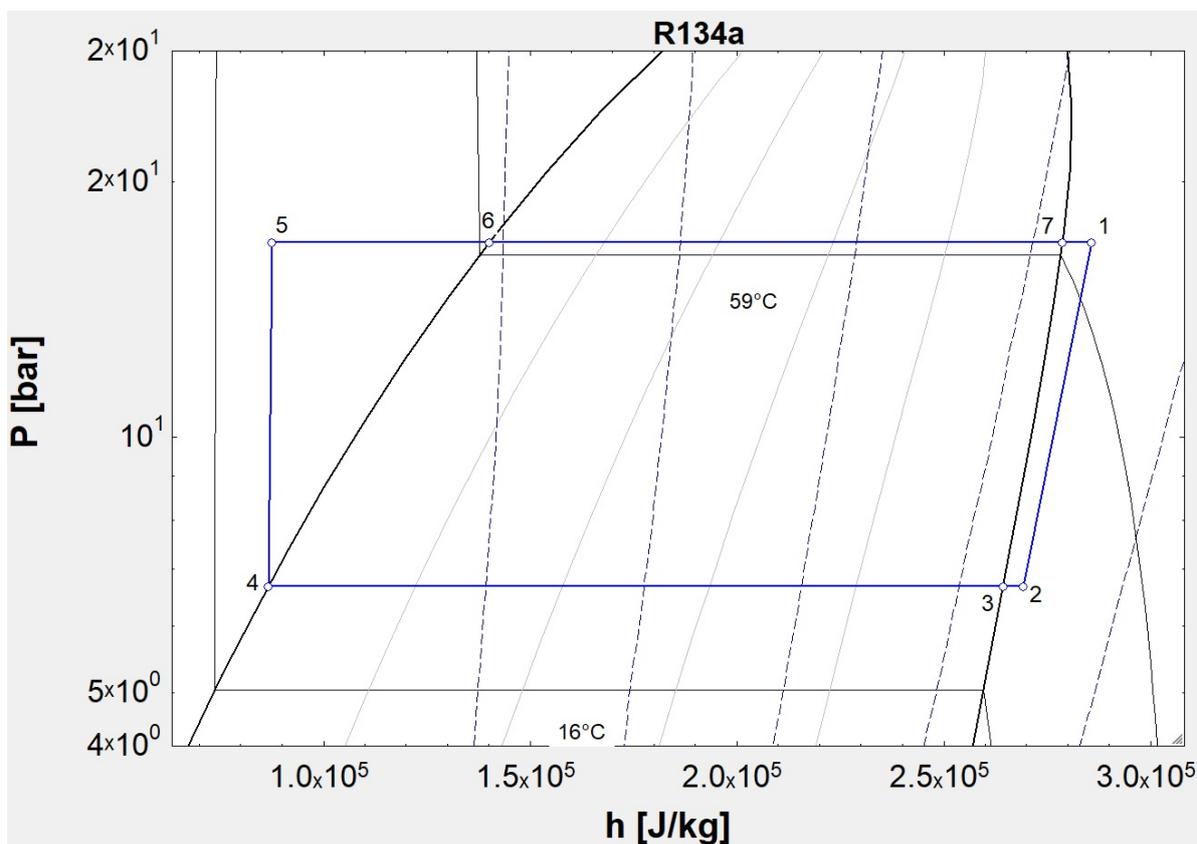
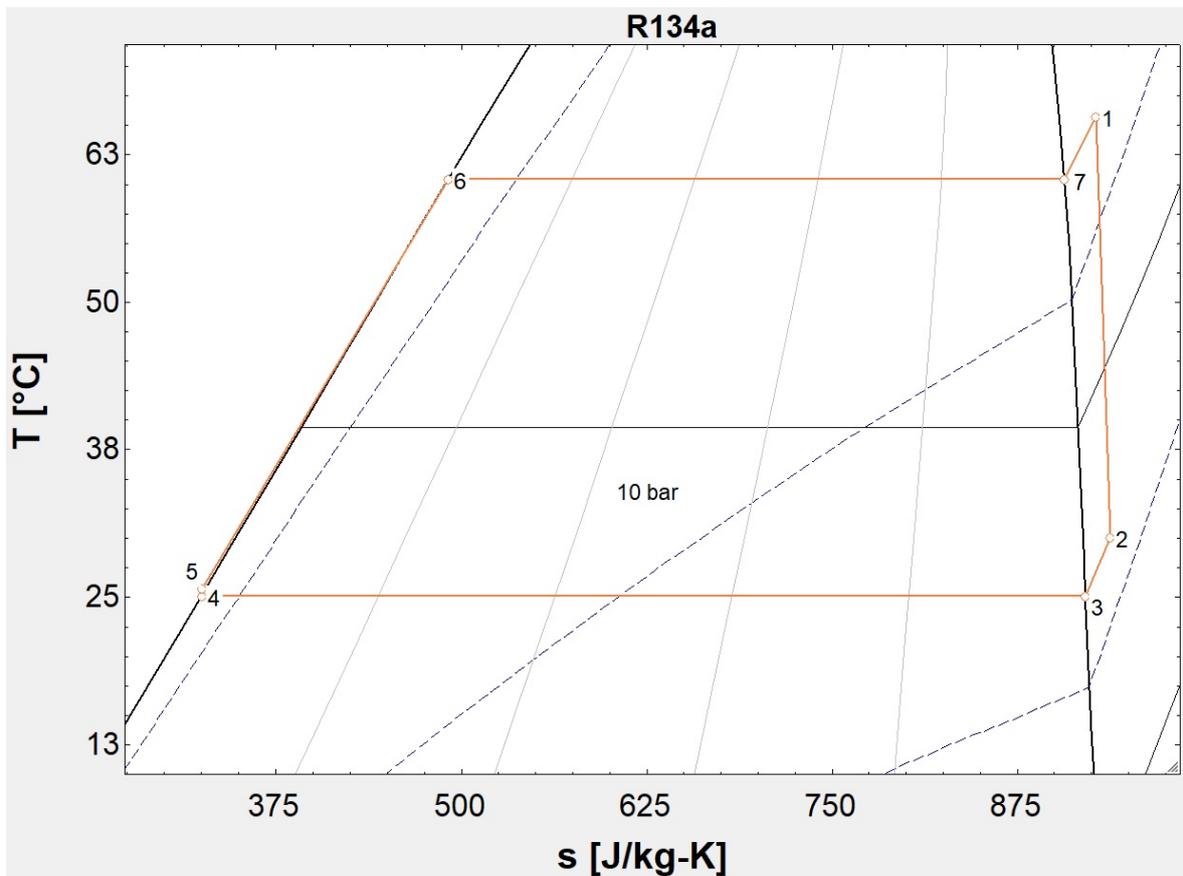
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REFERENCES

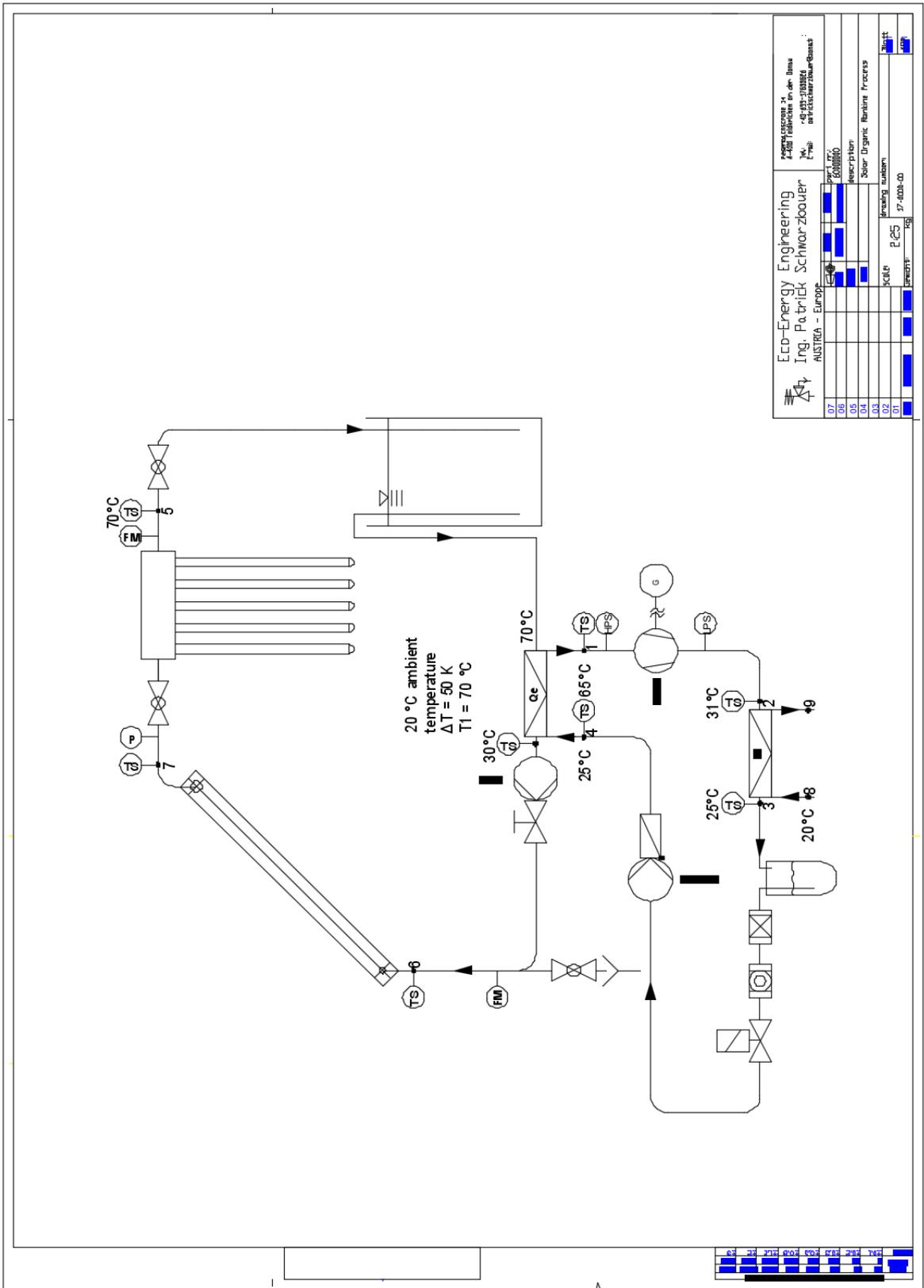
- [1] Jing Li: Structural Optimization and Experimental Investigation of the Organic Rankine Cycle for Solar Thermal Power Generation, China, Springer Theses, 2015.
- [2] Wikipedia The Free Encyclopedia.
Homepage: https://en.wikipedia.org/wiki/Pressure-volume_diagram
Date: 10/10/2017
- [3] Costante Mario Invernizzi: Closed Power Cycles, London, Springer, 2013.
- [4] Sylvain Quoilin, Martijn Van Den Broek, Sebastien Declaye, Pierre Dewallef, Vincent Lemort. Techno-economic survey of Organic Rankine Cycle (ORC) systems. Elsevier 2013.
- [5] The Linde Group, Linde AG: Klosterhofstraße 1, 80331 Munich, Germany.
Homepage:
http://www.linde-gas.com/de/products_and_supply/refrigerants/hfc_refrigerants/r134a/index.html
Date: 12/03/2017
- [6] Taschenbuch für Heizung + Klimatechnik; Recknagel, Sprenger, Schramek; 2013/2014;
76. Auflage; Oldenbourg Industrieverlag GmbH
- [7] The Engineering ToolBox.
Homepage:
https://www.engineeringtoolbox.com/Refrigerants-Environment-Properties-d_1220.html
Date: 12/05/2017
- [8] American Society of Heating, Refrigerating and Air-Conditioning Engineers. ASHRAE Handbook. Atlanta, GA: The Society, 1998.
- [9] Alfa Laval
Homepage: <http://www.alfalaval.de>
Date: 03/19/2017
- [10] David Meyer, Choon Wong, Frithjof Engel and Dr. Susan Krumdieck. DESIGN AND BUILD OF A 1 KILOWATT ORGANIC RANKINE CYCLE POWER GENERATOR. 35th New Zealand Geothermal Workshop: 2013 Proceedings 17 – 20 November 2013, Rotorua, New Zealand.
- [11] Incropera, DeWitt, Bergman, Lavine; Introduction to Heat Transfer; John Wiley & Sons, Inc. 2007.
- [12] Luwinox
<https://www.luwinox.de>
Date: 03/19/2017
- [12] Homepage Wikidot: Micro Scale Heat Exchangers
Homepage:
me1065.wikidot.com/micro-scale-heat-exchangers
Date: 10/12/2017
- [13] Homepage Mathworks.
Homepage: <https://www.mathworks.com>
Date: 10/12/2017

APPENDIX A

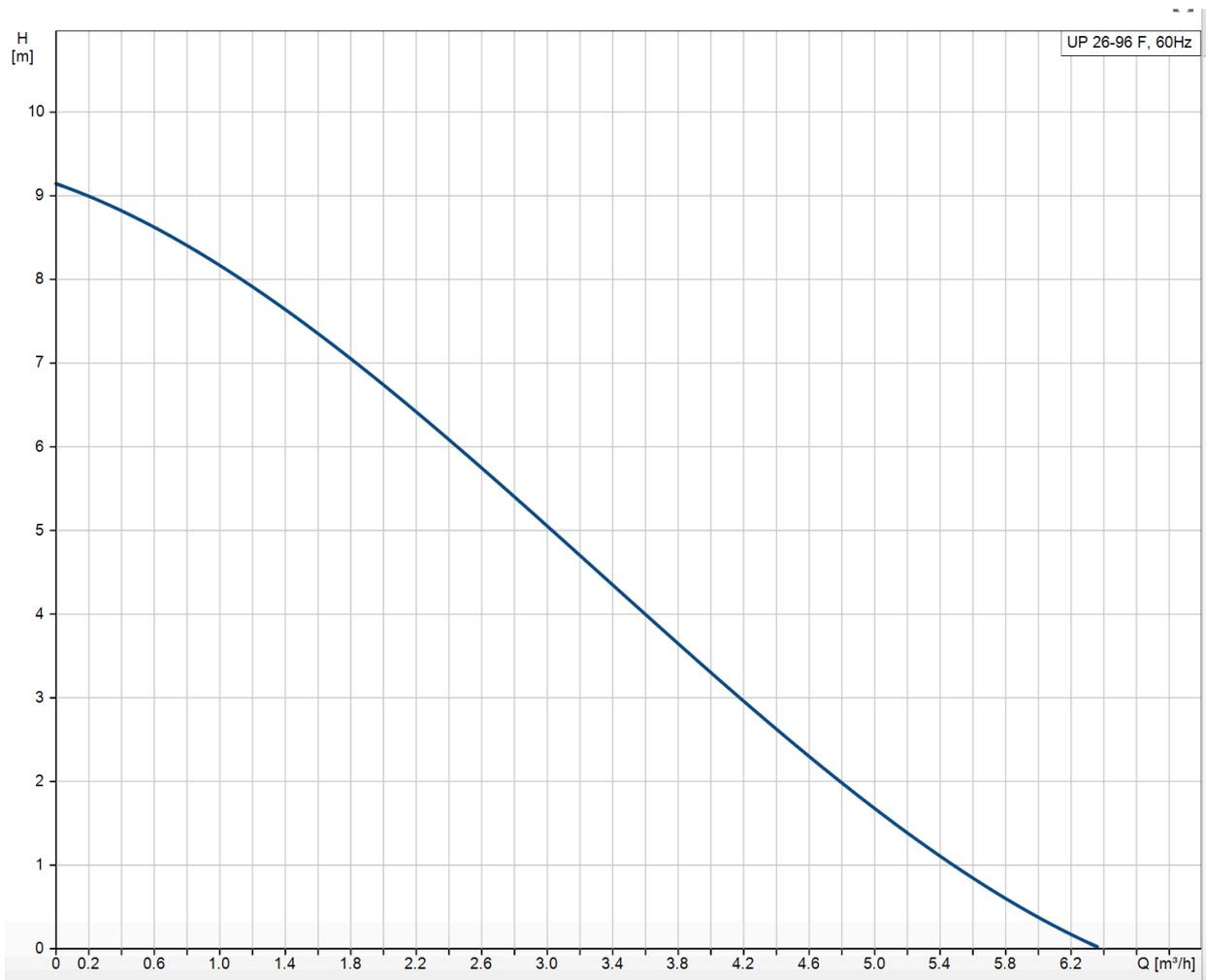
1. Cycle diagrams generated with EES.



2. Schematic drawing of the ORC system



3. Pump curve for solar circulation pump (Type: UP 26-96 F)



4. Refrigerant Environmental Data



Refrigerants Environmental Data. Ozone Depletion and Global Warming Potential.

Linde strongly advocates the use of refrigerant gases with zero ozone depletion potential and fully supports the phase out of HCFC's. Linde also advocates the use of lower global warming potential gases when they contribute to an overall reduction in the carbon footprint of refrigeration systems.

This table provides up to date internationally recognised data on the ozone depletion potential and global warming potential of the gases, as well as an easy reference summary of their impact.

Product information (sorted by Product Type and Name)

Type	Product R- Number	ODP ¹		GWP ²	
CFC	12	1	High	10900	High
	502	0,33	High	4657	High
HCFC	22	0,055	Medium	1810	Medium
	123	0,060	Medium	77	Low
	401A	0,033	Medium	1182	Medium
	401B	0,036	Medium	1288	Medium
	402A	0,019	Medium	2788	High
	402B	0,030	Medium	2416	Medium
	408A	0,024	Medium	3152	High
	409A	0,046	Medium	1909	Medium
HFC	23	0	Zero	14800	High
	32	0	Zero	675	Medium
	134a	0	Zero	1430	Medium
	404A	0	Zero	3922	High
	407A	0	Zero	2107	Medium
	407C	0	Zero	1774	Medium
	407F	0	Zero	2088	Medium
	417A	0	Zero	2346	Medium
	422A	0	Zero	3143	High
	422D	0	Zero	2729	High
	423A	0	Zero	2280	Medium
	424A	0	Zero	2440	Medium
	427A	0	Zero	2138	Medium
	428A	0	Zero	3607	High
	434A	0	Zero	3245	High
	437A	0	Zero	1805	Medium
	438A	0	Zero	2265	Medium
	442A	0	Zero	1888	Medium
	507A	0	Zero	3985	High
	508B	0	Zero	13396	High
M089	0	Zero	3805	High	
HFO	1234yf	0	Zero	4	Low
	1234ze	0	Zero	6	Low
Natural/Not in Kind	170	0	Zero	6	Low
	290	0	Zero	3	Low
	600a	0	Zero	3	Low
	717	0	Zero	0	Zero
	744	0	Zero	1	Low
	1150	0	Zero	4	Low
	1270	0	Zero	2	Low

ODP band	Montreal Protocol Impact	GWP band	EU F-Gas 2 Impact³
□ Zero	No restriction	■ Less than 150	Low No controls
■ Medium	Subject to consumption phase down	■ 150-2500	Medium Some supply restrictions and new equipment use bans
■ High	100% global production & consumption ban	■ Greater than 2500	High Substantial supply and use restrictions and new equipment bans

¹ Ozone Depletion Potential, UNEP (2006). R11=1, ² Global Warming Potential (100 year), IPCC 4th Assessment Report, 2007. CO₂ = 1, ³ Regulations under negotiation. Most likely scenario at time of publication.

Product information (sorted by environmental impact)

Type	Product R- Number	ODP ¹		GWP ²	
Natural	717	0	Zero	0	Zero
Natural	744	0	Zero	1	Low
Natural	1270	0	Zero	2	Low
Natural	290	0	Zero	3	Low
Natural	600a	0	Zero	3	Low
Natural	1150	0	Zero	4	Low
HFO	1234yf	0	Zero	4	Low
HFO	1234ze	0	Zero	6	Low
Natural	170	0	Zero	6	Low
HFC	32	0	Zero	675	Medium
HFC	134a	0	Zero	1430	Medium
HFC	407C	0	Zero	1774	Medium
HFC	437A	0	Zero	1805	Medium
HFC	407F	0	Zero	1825	Medium
HFC	442A	0	Zero	1888	Medium
HFC	410A	0	Zero	2088	Medium
HFC	407A	0	Zero	2107	Medium
HFC	427A	0	Zero	2138	Medium
HFC	438A	0	Zero	2265	Medium
HFC	423A	0	Zero	2280	Medium
HFC	417A	0	Zero	2346	Medium
HFC	424A	0	Zero	2440	Medium
HFC	422D	0	Zero	2729	High
HFC	422A	0	Zero	3143	High
HFC	434A	0	Zero	3245	High
HFC	428A	0	Zero	3607	High
HFC	M089	0	Zero	3805	High
HFC	404A	0	Zero	3922	High
HFC	507A	0	Zero	3985	High
HFC	508B	0	Zero	13396	High
HFC	23	0	Zero	14800	High
HCFC	123	0,060	Medium	77	Low
HCFC	402B	0,030	Medium	2416	Medium
HCFC	401A	0,033	Medium	1182	Medium
HCFC	401B	0,036	Medium	1288	Medium
HCFC	409A	0,046	Medium	1909	Medium
HCFC	22	0,055	Medium	1810	Medium
HCFC	402A	0,019	Medium	2788	High
HCFC	408A	0,024	Medium	3152	High
CFC	502	0,33	High	4657	High
CFC	12	1	High	10900	High

ODP band **Montreal Protocol Impact**

□ Zero No restriction

■ Medium Subject to consumption phase down

■ High 100% global production & consumption ban

GWP band

■ Less than 150 Low

■ 150-2500 Medium

■ Greater than 2500 High

EU F-Gas 2 Impact³

No controls

Some supply restrictions and new equipment use bans

Substantial supply and use restrictions and new equipment bans

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 Phone +49.89.7446-2339, Fax +49.89.7446-2071, refrigerants@linde-gas.com, www.linde-gas.com/refrigerants



¹ Ozone Depletion Potential, UNEP (2006). R11=1, ² Global Warming Potential (100 year), IPCC 4th Assessment Report, 2007. CO₂ = 1, ³ Regulations under negotiation. Most likely scenario at time of publication.

APPENDIX B

1. Heat exchanger calculation with Matlab.

```

%Patrick Schwarzbauer
%schwarzb20@up.edu
clear all
close all

%FINAL PROJECT: Calculation of a Heat
Exchanger (HEX) in MatLab

%Given information (Solar Thermal
Organic Rankine Cycle --> Capstone
project):
%Solar cycle (water)
T1_solar_deg=70;
%Inlet temperature Solar collector in
°C (Water)
T1_solar=70+273.15;
%Inlet temperature Solar collector in
K (Water)

deltaT_solar=40;
%Delta T in Solar cycle (K)
T2_solar_deg=T1_solar_deg-
deltaT_solar; %Outlet temperature
Solar collector in °C (Water)
T2_solar=T1_solar-deltaT_solar;
%Outlet temperature Solar collector in
K (Water)
cp_w=4180;
%specific heat capacity in J/(kgK),
assumption: constant due to low
temperature and no change in phase
mdot_solar=0.0063;
%Mass flow of the solar water cycle in
kg/s [PowerEnergy 2016-59098, 2016]

Q_solar=mdot_solar*cp_w*(T2_solar-
T1_solar) %Heat capacity solar cycle
in [J/s=W]
%-----
%-----
%-----

%ORC cycle (R134a)
%Calculation of enthalpy of the
superheated gas at the exit of the
%evaporator
T2_ev_deg=65;
%Outlet temperature Evaporator in °C
(R134a)
T2_ev=T2_ev_deg+273.15;
%Outlet temperature Evaporator in K
(R134a)
p_ev=17*10.^5;
%Evaporation pressure in [Pa] R134a:
assumption no pressure drop through
HEX

%Import excel data for Enthalpy
Table_h =
xlsread('PropertiesR134a.xlsx')
find p_ev=17
find T2_ev_deg=65

sl = 2;
%size of the square input matrix
t1 = Table_h(:,2);
%Capture the table headers
t = t1(1:length(t1));
%fix the first NaN value
t = repmat(t,1,sl-1);
%make the headers a matrix that
repeats
t = t(:)';
%creates a vector out of matrix

p1 = Table_h(:,1);
%Capture the labels for each row
p = p1(1:length(p1))';
%fix the first NaN value
p = repmat(p,1,sl-1);
%make the headers a matrix that
repeats

h = Table_h(:,3);
%h = Table_h(3:sl,2:sl); %Just the
enthalpy values

%h = reshape(h,1,length(h)^2); %turn
the matrix of h values into a long
vector
h= h(:)';
%creates a vector out of matrix

% Now we have 3 long vectors that
contain pressure, temperature and
% enthalpy. to find the closest point
to our target values we can use
% minimization of residuals

p_find=17;
%pressure of superheated gas
T_find=65;
%temperature of superheated gas
r = (p- repmat(p_find,1,length(p))).^2
+ (t- repmat(T_find,1,length(t))).^2;
% Find the smallest residual

[m,i] = min(r);
%the index, i, now represents the
point closest to the value we are
trying to look up.

h1 = interp1([p(i) p(i+1)], [h(i)
h(i+1)], p_find, 'linear')
%finds the value for enthalpy

```

```

%-----
%-----
%-----
%Calculation of enthalpy of the liquid
fluid at the inlet of the
%evaporator

t5 = Table_h(:,2);
%Capture the table headers
t = t5(1:length(t5));
%fix the first NaN value
t = repmat(t,1,sl-1);
%make the headers a matrix that
repeats
t = t(:)';
%creates a vector out of matrix

p5 = Table_h(:,1);
%Capture the labels for each row
p = p5(1:length(p5))';
%fix the first NaN value
p = repmat(p,1,sl-1);
%make the headers a matrix that
repeats

h = Table_h(:,3);
%h = Table_h(3:sl,2:sl); %Just the
enthalpy values

%h = reshape(h,1,length(h)^2); %turn
the matrix of h values into a long
vector
h= h(:)';
%creates a vector out of matrix

% Now we have 3 long vectors that
contain pressure, temperature and
% enthalpy. to find the closest point
to our target values we can use
% minimization of residuals

p_find=17;
%pressure of the liquid refrigerant
T_find=25;
%temperature of the liquid refrigerant
r = (p- repmat(p_find,1,length(p))).^2
+ (t- repmat(T_find,1,length(t))).^2;
% Find the smallest residual

[m,i] = min(r);
%the index, i, now represents the
point closest to the value we are
trying to look up.

h5 = interp1([p(i) p(i+1)], [h(i)
h(i+1)], p_find, 'linear')
%finds the value for enthalpy

%-----
%-----
%-----
%Calculation of enthalpy of the
preheated liquid (x=0)

t6 = Table_h(:,2);
%Capture the table headers
t = t6(1:length(t6));
%fix the first NaN value
t = repmat(t,1,sl-1);
%make the headers a matrix that
repeats
t = t(:)';
%creates a vector out of matrix

p6 = Table_h(:,1);
%Capture the labels for each row
p = p6(1:length(p6))';
%fix the first NaN value
p = repmat(p,1,sl-1);
%make the headers a matrix that
repeats

h = Table_h(:,3);
%h = Table_h(3:sl,2:sl); %Just the
enthalpy values

%h = reshape(h,1,length(h)^2); %turn
the matrix of h values into a long
vector
h= h(:)';
%creates a vector out of matrix

% Now we have 3 long vectors that
contain pressure, temperature and
% enthalpy. to find the closest point
to our target values we can use
% minimization of residuals

p_find=17;
%pressure of the preheated liquid
T_find=60;
%temperature of the preheated liquid
r = (p- repmat(p_find,1,length(p))).^2
+ (t- repmat(T_find,1,length(t))).^2;
% Find the smallest residual

[m,i] = min(r);
%the index, i, now represents the
point closest to the value we are
trying to look up.

h6 = interp1([p(i) p(i+1)], [h(i)
h(i+1)], p_find, 'linear') %finds the
value for enthalpy

%-----
%-----

```

```

-----
-----
%Calculation of each heat capacity (3-
phase)
h7=426600;
%Enthalpy of the saturated refrigerant
(x=1) [J/kg]

Q_ev = 13891      %(-Q_solar)
%Heat capacity of the Evaporator (ORC)
in W
mdot_rf=Q_ev/(h1-h5);
%calculation of the mass flow rate in
[kg/s]

Q_ev_1=mdot_rf*(h6-h5)
%Heat capacity of the Evaporator
(ORC): heat up to saturation
temperature in [W]
Q_ev_2=mdot_rf*(h7-h6)
%Heat capacity of the Evaporator
(ORC): Evaporation from x=0 to x=1 in
[W]
Q_ev_3=mdot_rf*(h1-h7)
%Heat capacity of the Evaporator
(ORC): superheated gas (saturation
temp. + 10K to make sure all the gas
is evaporated) in W
%-----
-----
-----
%Heat exchanger (HEX) calculation for
preheating (State 5 to 6):
T6_deg=60;
%saturation temperature of R134a in
[C]
T6=T6_deg+273.15;
%saturation temperature of R134a in
[K]

%Assumption for inlet refrigerant
temperature: T1_ev = 25 C, needed to
%calcualte mean specific isobar heat
capacity cp [cp]=J/(kg*K)

cp_rf_5=1414.1;
%specific isobar heat capacity cp
[cp]=J/(kg*K) of inlet refrigerant
cp_rf_6=1653.2;
%specific isobar heat capacity cp
[cp]=J/(kg*K) of exit refrigerant,
before evaporation begins
cpm_rf_1=(cp_rf_6+cp_rf_5)/2;

T5=T6-Q_ev_1/(mdot_rf*cpm_rf_1);
%inlet temperature R134a in [K]
T5_deg=T5-273.15;
%inlet temperature R134a [C]

%Assumption OK!
Q_solar_1=Q_ev_1;
%set both heat capacities equal to
calculate inlet temperature of solar
cycle (see Figure XX)
Tx=Q_solar_1/(mdot_solar*cp_w) + T5;
%inlet temperature of solar cycle for
preheating the refrigerant in [K]
Tx_deg=Tx-273.15;
%inlet temperature of solar cycle for
preheating the refrigerant in [C]

%Geometry: Compact plate HEX that
consists of a stack of thin metal
sheets,
%seperated by N gaps of width a. To
keep the calculation easy, a
%quadratical geometry was choosen. L
represents the exterior dimensions.

L=0:0.1:1;
%analyzing the length from 0 to 1m
N=0:10:100;
%analyzing the number of gaps

NuD=7.54;
%Nusselt number for each interior gap
(assumption: laminar flow) Table 8.1
[Incropera et al. 2007, Page 489]

k_ref_5=0.0803;
%thermal conductivity of R134a at
state 5, saturated liquid (25C) in
[W/(m*K)][Incropera et al. 2007, Page
858] ATTENTION THERE IS A MISTAKE IN
THIS BOOK! IT SAYS k*10^3 BUT IT
SHOULD SAY k*10^(-3)
k_ref_6=0.0675;
%thermal conductivity of R134a at
state 6, saturated liquid (60C) in
[W/(m*K)][Incropera et al. 2007, Page
858]
k_ref=(k_ref_5+k_ref_6)/2;
%mean value of thermal conductivity
(R134a)
k_solar=0.625;
%thermal conductivity of water in
[W/(m*K)][Incropera et al. 2007, Page
643]

delta1=(Tx-T6);
%temp. difference between solar cycle
inlet and ORC cycle outlet
delta2=(T2_solar-T5);
%temp. difference between solar cycle
outlet and ORC cycle inlet

```

```

deltathetam_1=(delta1-
delta2)/log(delta1/delta2); %mean
specfic heat capacity

h_ref_1 =
@(N,L) (NuD.*k_ref.*N)./(2.*L)
%heat transfer coefficient of
refrigerant
h_solar_1 =
@(N,L) (NuD.*k_solar.*N)./(2.*L)
%heat transfer coefficient of solar
water

%{
Problem with In-line function:
Q_ev_1_v=Q_ev_1*[1 1 1 1 1 1 1 1 1 1
1]
L = @(N)
(Q_ev_1_v)/(deltathetam_1.*(N-
1).*N).*((1/h_ref_1)+(1/h_solar_1))
I tried to create a vector same
dimension as N for solving the in-line
function L. But it doesn't work.
So to continue with the code I decide
to simplify the equation to numbers
and then create a simpler in-line
function.
%}

L= @(N) 95.1227./(N.*(N-1))
%In-line function L in [m]

plot(N,L(N))
%Plot numbers of gaps over the length
of the HEX
xlabel('Number of Gaps, N')
%Label axis
ylabel('Exterior Dimension L [m]')
%Label axis
hold on

%{
Design Decision:
To keep the HEX comapact I recommand a
exterior length of 20cm (0.2m), see
plot (N,L(N)). This leads to a number
of 22 gaps.
%}
N_1=22;
%Number of gaps due to design decision
Ext_length=L(N_1)
%Exterior length due to design
decision in [m]

al=Ext_length/N_1
%gap width in [m]
Dh=2*Ext_length;
%hydraulic diameter in [m]

rho_solar=1000;
%density water in [kg/m^3]
um_1_solar=2*mdot_solar/(rho_solar*Ext
_length.^2); %mean velocity in
each water-filled gap
mu_solar=725*10^(-6);
%dynamic viscosity of water [Pa*s]
ReD_solar_1 =
(rho_solar*um_1_solar*Dh)/mu_solar
%Reynolds number of water

rho_rf_1=1212.2;
%density R134a @ 25 C in [kg/m^3]
rho_rf_2=1054.54;
%density R134a @ 60 C in [kg/m^3]
rho_rf=(rho_rf_1+rho_rf_2)/2;
um_1_rf=2*mdot_rf/(rho_rf*Ext_length.^
2); %mean velocity in
each R134a-filled gap
mu_rf_1=200.61*10^(-6);
%dynamic viscosity of R134a @ 25 C
[Pa*s]
mu_rf_2=84.74*10^(-6);
%dynamic viscosity of R134a @ 60 C
[Pa*s]
mu_rf=(mu_rf_1+mu_rf_2)/2;
%mean dynamic viscosity of R134a
ReD_rf_1 = (rho_rf*um_1_rf*Dh)/mu_rf
%Reynolds number of liquid refrigerant
R134a
%ASSUMPTION OF LAMINAR FLOW IS
CORRECT!

A1=Ext_length.^2*(N_1-1)
%total heat transfer area for
preheating
U1=Q_ev_1/(A1*deltathetam_1)
%overall convection coefficient for
preheating phase in [W/(m^2*K)]

%-----
%-----
%-----
%Heat exchanger(HEX) calculation for
boiling (State 6 to 7):

U2=2000;
%ASSUMPTION: overall convection
coefficient for boiling phase in
[W/(m^2*K)]

Ty=Q_ev_2/(mdot_solar*cp_w)+Tx;
%inlet temperature of solar cycle for
boiling the refrigerant in [K]
Ty_deg=Ty-273.15;
%inlet temperature of solar cycle for
boiling the refrigerant in [C]

```

```

delta3=(Ty-T6);
%temp. difference between solar cycle
inlet and ORC cycle outlet
delta4=(Tx-T6);
%temp. difference between solar cycle
outlet and ORC cycle inlet

deltathetam_2=(delta3-
delta4)/log(delta3/delta4);
%mean specific heat capacity

A2=Q_ev_2/(U2*deltathetam_2)
%total heat transfer area for boiling
N_2=A2/(Ext_length.^2) + 1;
%Number of gaps calculated

N_2_chosen=2
%Number of gaps chosen
a2=Ext_length/N_2_chosen
%gap width in [m]

%-----
%-----
%-----
%Heat exchanger (HEX) calculation for
superheating (State 7 to 1):

L_3=0:0.1:1;
%analyzing the length from 0 to 1m
N_3=0:10:100;
%analyzing the number of gaps

cp_rf_7=1395.3;
%specific isobar heat capacity cp
[cp]=J/(kg*K) of inlet refrigerant @
17 bar and saturation temperature 60 C
cp_rf_1=1482;
%specific isobar heat capacity cp
[cp]=J/(kg*K) of exit refrigerant,
superheated gas (18 bar)
cpm_rf_3=(cp_rf_6+cp_rf_5)/2;

k_ref_7=19.14*10^(-3);
%cof R134a at state 7, saturated
vapour (60C) in [W/(m*K)] [Incropera et
al. 2007, Page 858] ATTENTION THERE IS
A MISTAKE IN THIS BOOK! IT SAYS k*10^3
BUT IT SHOULD SAY k*10^(-3)
k_ref_1=20.09*10^(-3);
%thermal conductivity of R134a at
state 1, superheated gas (65C) in
[W/(m*K)] [Incropera et al. 2007, Page
858]
k_ref_3=(k_ref_7+k_ref_1)/2;
%mean value of thermal conductivity
(R134a)

delta7=(T1_solar-T2_ev);
%temp. difference between solar cycle
inlet and ORC cycle outlet
deltall=(Ty-T6);
%temp. difference between solar cycle
outlet and ORC cycle inlet

deltathetam_3=(delta7-
deltall)/log(delta7/deltall);
%mean specific heat capacity

h_ref_3 =
@(N_3,L_3) (NuD.*k_ref_3.*N_3)./(2.*L_3
)
%heat transfer
coefficient of refrigerant
h_solar_3 =
@(N_3,L_3) (NuD.*k_solar.*N_3)./(2.*L_3
)
%heat transfer
coefficient of solar water

%{
Problem with In-line function:
Q_ev_3_v=Q_ev_3*[1 1 1 1 1 1 1 1 1 1
1]
L_3 = @(N_3)
(Q_ev_3_v)/(deltathetam_3.*(N_3-
1).*N_3).*((1/h_ref_3)+(1/h_solar_3))
I tried to create a vector same
dimension as N for solving the in-line
function L. But it doesn't work.
So to continue with the code I decide
to simplify the equation to numbers
and then create a simpler in-line
function.
%}

L_3= @(N_3) 189.5./(N_3.*(N_3-1))
%In-line function L in [m]

plot(N_3,L_3(N_3))
%Plot numbers of gaps over the length
of the HEX
xlabel('Number of Gaps, N')
%Label axis
ylabel('Exterior Dimension L [m]')
%Label axis
hold on

%{
Design Decision:
To keep the HEX compact I recommend a
exterior length of 20cm (0.2m), see
plot (N_3,L_3(N_3)). This leads to a
number of 31 gaps.
%}
N_3=31;
%Number of gaps due to design decision

a3=Ext_length/N_3
%gap width in [m]

```

```

rho_solar=1000;
%density water in [kg/m^3]
um_3_solar=2*mdot_solar/(rho_solar*Ext_length.^2);
%mean velocity in each water-filled gap

ReD_solar_3 =
(rho_solar*um_3_solar*Dh)/mu_solar
%Reynolds number of water

rho_rf_7=88.5183375;
%density R134a @ 60 C in [kg/m^3]
rho_rf_11=94.778433653846;
%density R134a @ 65 C in [kg/m^3]
rho_rf_3=(rho_rf_7+rho_rf_11)/2;
um_3_rf=2*mdot_rf/(rho_rf_3*Ext_length.^2);
%mean velocity in each R134a-filled gap
mu_rf_7=14.185875*10^(-6);
%dynamic viscosity of R134a @ 60 C [Pa*s]
mu_rf_11=14.382990384615*10^(-6);
%dynamic viscosity of R134a @ 65 C [Pa*s]
mu_rf_3=(mu_rf_7+mu_rf_11)/2;
%mean dynamic viscosity of R134a
ReD_rf_3 =
(rho_rf_3*um_3_rf*Dh)/mu_rf_3
%Reynolds number of liquid refrigerant R134a

A3=Ext_length.^2*(N_3-1)
%total heat transfer area for preheating
U3=Q_ev_3/(A3*deltathetam_3)
%overall convection coefficient for preheating phase in [W/(m^2*K)]

%ASSUMPTION OF LAMINAR FLOW IS INCORRECT! Calculate new Nusselt number for
%turbulent flow (R134a)!

%-----
%-----
%-----
%New calculation due to turbulent flow conditions

Pr_rf_7=1.0335;
Pr_rf_11=1.0527307692308;
Prm_rf_3=(Pr_rf_7+Pr_rf_11)/2;

%Assumption for friction factor f:
Material:Copper, Lead, Brass, Aluminum
%--> Absolute roughness coefficient
k=(0.001 - 0.002)*10^(-3) m
[https://www.engineeringtoolbox.com/su

rface-roughness-ventilation-ducts-d_209.html]
%Then use Moody diagram for friction factor based on ReD_rf_3 &
k=0.0015*10^(-3)!

f=0.035;
%friction factor choosen with Moody diagram
[https://www.engineeringtoolbox.com/moody-diagram-d_618.html]

NuD_rf_3=((f/8)*(ReD_rf_3-1000)*Prm_rf_3)/(1+12.7*(f/8).^0.5)*(Prm_rf_3^(2/3)-1)
%[PAGE 485 Eq. 8.62]

L_3_new=0:0.1:1;
%analyzing the length from 0 to 1m
N_3_new=0:10:100;
%analyzing the number of gaps

h_ref_3_new =
@(N_3_new,L_3_new) (NuD_rf_3.*k_ref_3.*N_3_new)./(2.*L_3_new)
%heat transfer coefficient of refrigerant
h_solar_3 =
@(N_3_new,L_3_new) (NuD.*k_solar.*N_3_new)./(2.*L_3_new)
%heat transfer coefficient of solar water

L_3_new= @(N_3_new)
99.5658./(N_3_new.*(N_3_new-1))
%In-line function L in [m]

plot(N_3_new,L_3_new(N_3_new))
%Plot numbers of gaps over the length of the HEX
xlabel('Number of Gaps, N')
%Label axis
ylabel('Exterior Dimension L [m]')
%Label axis
legend('Liquid phase','Boiling phase','Superheated phase') %adds legend

N_3_new=23;
%Number of gaps due to design decision (L=200mm)

a3_new=Ext_length/N_3_new
%gap width in [m]

A3_new=Ext_length.^2*(N_3_new-1)
%total heat transfer area for preheating

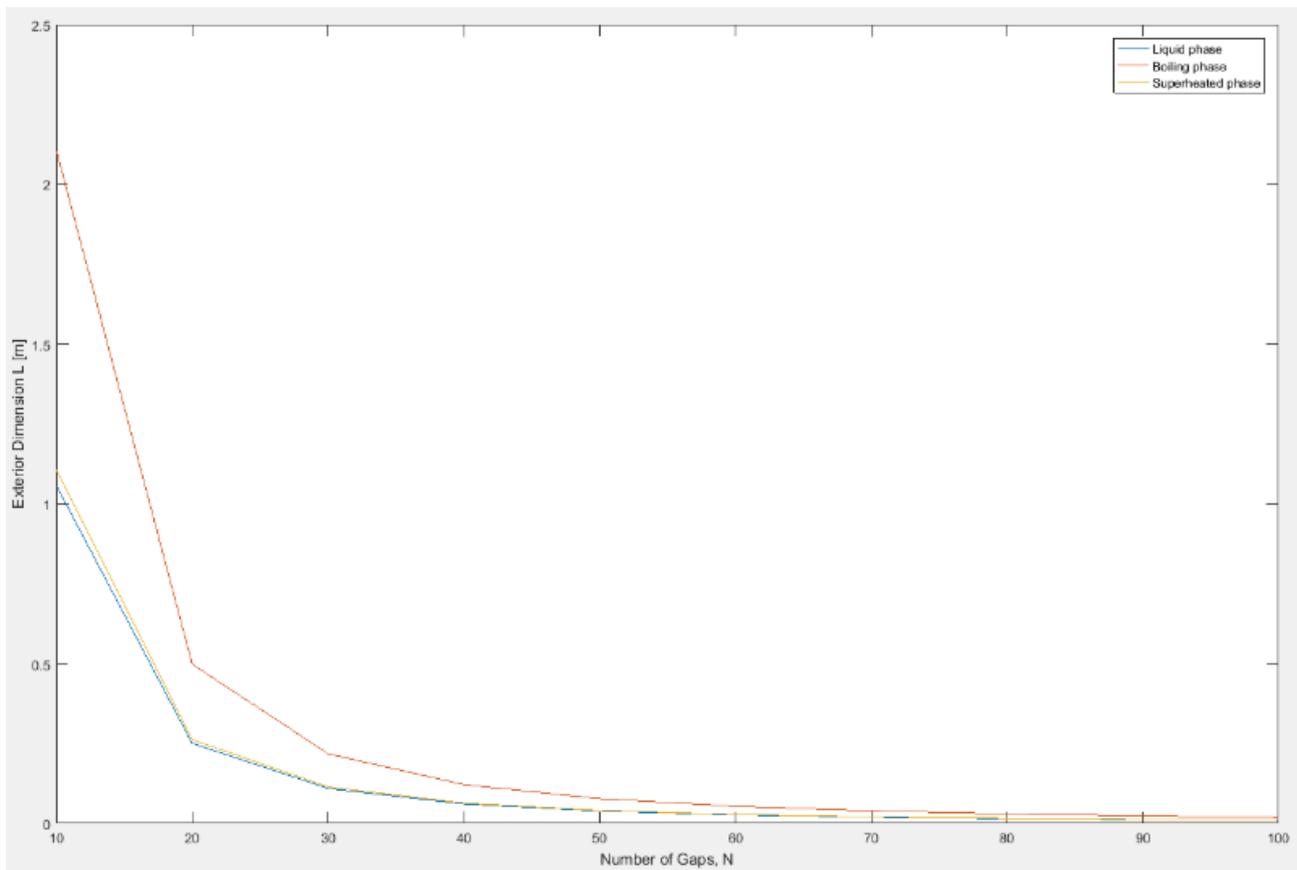
```

```

U3_new=Q_ev_3/(A3_new*deltathetam_3)
%overall convection coefficient for
preheating phase in [W/(m^2*K)]

N_total=N_1+N_2_choosen+N_3_new
%total value of gaps
A_total=A1+A2+A3_new
%total area in [m^2]
B_total=N_1*a1+N_2_choosen*a2+N_3_new*
a3_new          %total width in [m]
%-----
%-----
%-----
%-----
%-----
%-----

```



2. Thermodynamic cycle calculation with EES.

"{Bachelor Thesis 1: Solar Thermal Organic Rankine Cycle (ORC)
 Author: Ing. Patrick Schwarzbauer
 Eco-Energy Engineering (University of Applied Sciences Upper Austria)
 In cooperation with: University of Portland
 Supervisor: Dr. Heather Dillon (University of Portland)
 DI Rudolf Kraft (FH-Wels)

 For better understanding of the code and all the points mentioned in the code look up the schematic drawing of the system

General information:
 This EES code is programmed for Organic-Rankine cycles (short: ORC).
 Solar collectors are providing the heat for running the refrigerant cycle and the turbine.
 All units are in the metric system (SI).}"

"THERMODYNAMIC CALCULATION OF ALL POINTS IN THE SYSTEM:"

"Point 1: Superheated refrigerant entering the turbine"

T[1]=T[5]+40
 [C]

P[1]=P[5]

h[1]=enthalpy(R134a,T=T[1],P=P[1])

s[1]=entropy(R134a,T=T[1],P=P[1])

v[1]=volume(R134a,T=T[1],P=P[1])

rho[1]=density(R134a,T=T[1],P=P[1])

"hsat_liq[1] = enthalpy(R134a, x = 0, P=P[4])"
 "hsat_vap[1] = enthalpy(R134a, x=1, P=P[4])"
 "Tsat[1]=t_sat(R134a,P=P[1])"

"Point 2: After turbine / before condensation"

P[2] = 6.684
 [bar]

"Condensing Pressure, Temperature = 25C"

etaT =
 0.80

"Book: Closed Power Cycles [3]"

etaT = (h[1]-h[2])/(h[1]-
 hs[2])

ss[2]=s[1]

hs[2]=enthalpy(R134a,s=ss[2],P=P[2])
 s[2] =entropy(R134a,P = P[2],h=h[2])
 T[2]=temperature(R134a,P=P[2],h=h[2])

rho[2]=density(R134a,h=h[2],P=P[2])

"Tsat[2] =t_sat(R134a,P=P[2])"
 "hsat[2] = enthalpy(R134a, x = 1, P =
 P[2])"

"Point 3: Condensation to x=1"

P[3]=P[2]
 x[3]=1

h[3]=enthalpy(R134a,x=x[3],P=P[3])
 s[3]=entropy(R134a, h=h[3],
 P=P[3])

v[3]=volume(R134a,h=h[3],P=P[3])

rho[3]=density(R134a,h=h[3],P=P[3])
 T[3]=temperature(R134a,h=h[3],P=P[3])

"Point 4: after condensing (saturated liquid)"

$$P[4] = P[2]$$

$$x[4]=0$$

$$T[4] = \text{temperature}(R134a, P=P[4], x=x[4])$$

$$h[4]=\text{enthalpy}(R134a, x=x[4], P=P[4])$$

$$s[4]=\text{entropy}(R134a, x=x[4], P=P[4])$$

$$v[4]=\text{volume}(R134a, x=x[4], P=P[4])$$

$$\rho[4]=\text{density}(R134a, x=x[4], P=P[4])$$

"Point 5: Entering the Pump"

$$P[5] = 17 \text{ [bar]}$$

$$\eta_P = 0.75$$

"Book: Closed Power Cycles [3]"

$$\eta_P = (h[5]-h[4])/(h[5]-h[4])$$

$$s[5]=s[4]$$

$$h[5]=\text{enthalpy}(R134a, s=s[5], P=P[5])$$

$$s[5]=\text{entropy}(R134a, h=h[5], P=P[5])$$

$$v[5]=\text{volume}(R134a, h=h[5], P=P[5])$$

$$\rho[5]=\text{density}(R134a, h=h[5], P=P[5])$$

$$T[5] = \text{temperature}(R134a, h=h[5], P=P[5])$$

"Point 6: Heating up the refr. to evaporation pressure"

$$P[6]=P[5]$$

$$x[6]=0$$

$$h[6]=\text{enthalpy}(R134a, x=x[6], P=P[6])$$

$$s[6]=\text{entropy}(R134a, h=h[6], P=P[6])$$

$$v[6]=\text{volume}(R134a, h=h[6], P=P[6])$$

$$\rho[6]=\text{density}(R134a, h=h[6], P=P[6])$$

$$T[6]=\text{temperature}(R134a, h=h[6], P=P[6])$$

"Point 7: Evaporation to x=1"

$$P[7]=P[6]$$

$$x[7]=1$$

$$h[7]=\text{enthalpy}(R134a, x=x[7], P=P[7])$$

$$s[7]=\text{entropy}(R134a, h=h[7], P=P[7])$$

$$v[7]=\text{volume}(R134a, h=h[7], P=P[7])$$

$$\rho[7]=\text{density}(R134a, h=h[7], P=P[7])$$

$$T[7]=\text{temperature}(R134a, h=h[7], P=P[7])$$

"Point 8: superheating the gas to start condition [1]"

$$P[8]=P[1]$$

$$T[8]=T[1]$$

$$h[8]=\text{enthalpy}(R134a, T=T[8], P=P[8])$$

$$s[8]=\text{entropy}(R134a, h=h[8], P=P[8])$$

$$v[8]=\text{volume}(R134a, h=h[8], P=P[8])$$

```

rho[8]=density(R134a,h=h[8],P=P[8])

flow_rfc = 0.000833 [m^3/s]
mrfc =
rho[1]*flow_rfc

    "set based on minimum flow rate of bitzer turbine
(3 m3/h)"
Qrfc= mrfc*(h[1]-h[5])

Wt = mrfc*(h[2] - h[1])
Wp = mrfc*(h[5] - h[4])
Wnet = (-Wt)-Wp
eta = Wnet/Qrfc *100

"CONDENSATION"
"caclulate heat taken away from R134a during
condensation"
"during condensation and from superheated to
satuarted vapor"

"Q_cond = mrfc*(hsat[2] - h[3]) + mrfc*(h[2] -
hsat[2])"

Q_cond = mrfc*(h[4]-h[2])
Q_cond = Q_cw

Q_cw = m_cw*cp_cw*(T_cw2 - T_cw1)

"Cooling Water Specs"
T_cw1 =
20

    "14 C winter temperature / 20 C summer
temperature"
T_cwf = (T_cw1+T_cw2)/2

    "film temperature"
flow_cw = 0.0002208 [m^3/s]

    "3.5 gpm based turbine project or could be
10gpm based on Port. Water Bureau"
P_cw = 4.14

    "60 psi between 40-80 range"
cp_cw=cp(Water,T=T_cwf,P=P_cw)
"rho_cw = density(Water,T=T_cwf,P=P_cw)"
rho_cw = 1000 [kg/m^3]
m_cw = rho_cw*flow_cw

"EVAPORATION"
"caclulate heat taken away from R134a during
evaporation"

"during subcooled to saturated liquid, evaporation,
and from satuarted vapor to superheated"

"Q_evap = mrfc*(hsat_liq[1]-h[4]) +
mrfc*(hsat_vap[1]-hsat_liq[1]) + mrfc*(h[1] -
hsat_vap[1])"
Q_evap = mrfc*(h[1]-h[5])

"calculate heat from solar collector"
"2: Calculation of mass flow in the refrigerant cycle"

m_sc=0.0063
[kg/s]

    "Mass flow of the solar water cycle
[PowerEnergy 2016-59098, 2016]"
"constrained by research paper which found max
delta T at this flow rate of water"

T_sc1=30.5
[C]

    "Inlet temperature solar water cycle"
"{Tsc1_deg=converttemp(K;C;Tsc1)}"

    "Converting Kelvin in Celsius"

T_sc2=70
[C]

    "Outlet temperature solar water cycle"
"{Tsc2_deg=converttemp(K;C;Tsc2)}"

    "Converting Kelvin in Celsius"

T_scf=(T_sc1+T_sc2)/2

    "Mean temperauter of solar water cycle"
P_sc=1.01325
[bar]

    "Pressure of solar water cycle --> assumption:
ambient pressure due to open system 1013mbar"

cp_sc=cp(Water,T=T_scf,P=P_sc)

    "specific heat at constant
pressure"

```

"heat from solar collector"

$$Q_{sc} = m_{sc} \cdot c_{p,sc} \cdot (T_{sc2} - T_{sc1})$$

"theoretical equivalence of heat from solar collector and r134a enthalpies to find theoretical mass flow of r134a"

$$Q_{sc} = m_{rfc_prime} \cdot (h_{sat_liq}[1] - h[4]) + m_{rfc_prime} \cdot (h_{sat_vap}[1] - h_{sat_liq}[1]) + m_{rfc_prime} \cdot (h[1] - h_{sat_vap}[1])$$

"mrfc_prime=0.00625 kg/s is too low for operating conditions of turbines and pumps"

$$Q = k \cdot A \cdot \Delta T_{thetam}$$

$$\Delta T_1 = (T_{sc2} + 273.15) - (T[1] + 273.15)$$

$$\Delta T_2 = (T_{sc1} + 273.15) - (T[5] + 273.15)$$

$$\Delta T_{thetam} = (\Delta T_1 - \Delta T_2) / (\ln(\Delta T_1 / \Delta T_2))$$

$$k = 1200 \text{ [W/m}^2\text{*k]}$$

$$A = Q_{evap} / (k \cdot \Delta T_{thetam})$$

2.1 Table including properties of each state.

Sort	1 h_i	2 P_i [mixed]	3 ρ_i	4 s_i	5 T_i [mixed]	6 v_i	7 x_i	8 h_{s_i}	9 ss_i
[1]	285665	17	84.12	927.6	65.7	0.01189			
[2]	270220	6.684	31.39	940.3	30.99			266359	927.6
[3]	264244	6.684	32.5	920.5	25.13	0.03077	1		
[4]	86589	6.684	1206	324.9	25.13	0.000829	0		
[5]	87443	17	1210	324.9	25.7	0.0008263		87229	324.9
[6]	140062	17	1051	491.3	60.43	0.0009518	0		
[7]	278570	17	88.51	906.4	60.43	0.0113	1		
[8]	285665	17	84.12	927.6	65.7	0.01189			

2.2 Solutions

Unit Settings: SI C bar J mass deg

$$A = 2.549 \text{ [m}^2\text{]} \quad c_{p,cw} = 4184 \text{ [J/kg*K]} \quad c_{p,sc} = 4181 \text{ [J/kg*K]}$$

$$k = 1200 \text{ [W/m}^2\text{*k]} \quad m_{rfc} = 0.07008 \text{ [kg/s]} \quad m_{cw} = 0.2208 \text{ [kg/s]}$$

$$\rho_{cw} = 1000 \text{ [kg/m}^3\text{]} \quad T_{cw1} = 20 \text{ [C]} \quad T_{cw2} = 6.07 \text{ [C]}$$

$$\delta_1 = 4.296 \quad \delta_2 = 4.796 \quad \Delta T_{thetam} = 4.541$$

$$\eta = 7.361 \quad \text{etaP} = 0.75 \quad \text{etaT} = 0.8 \quad \text{flow}_{cw} = 0.0002208 \text{ [m}^3\text{/s]} \quad \text{flow}_{rfc} = 0.000833 \text{ [m}^3\text{/s]}$$

$$Q_{rfc} = 13891 \text{ [W]} \quad Q_{cond} = -12868 \text{ [W]} \quad Q_{cw} = -12868 \text{ [W]} \quad Q_{evap} = 13891 \text{ [W]} \quad Q_{sc} = 1041 \text{ [W]}$$

$$T_{scf} = 50.25 \text{ [C]} \quad W_{net} = 1022 \text{ [W]} \quad W_p = 59.83 \text{ [W]} \quad W_t = -1082 \text{ [W]}$$