TECHNISCHE UNIVERSITÄT WIEN
Vienna University of Technology

Optimising tool for the Supercritical Organic Rankine Cycle and Steam Rankine cycle

A 30 credit unit diploma project under supervision of

Ao. Univ. Prof. Dipl.-Ing. Dr. techn. Andreas WERNER

Institute for Energy Systems and Thermodynamics

This master thesis is completed at the Vienna University of Vienna

for the

Faculty of Chemical and Process Engineering

Michael Eilmer

e0525141 (E734)

Vienna, October 2012

Abstract

This work describes the development of an optimizing tool as help for the basic engineering of Organic Rankine and Steam Rankine cycles. Steam Rankine Cycles, or in this work called SRC's, are well proved for high temperature flue gas. Here the work gives a tool to calculate the basic variables for the design of such a facility. The so called ORC cycles are able to produce energy from low grade heat and can be a step forward in today's economic and environmental issues. For the ORC processes the improvement in technology leads to the possibility of using this cycle in the supercritical area. A possible lower entropy destruction results in higher power output as well as a higher thermal efficiency what is shown in the simulations at the end of this work. The tool is build up on an Open Source Programming language called PYTHON and gives choices between sub-, supercritical Organic Rankine, single pressure and double pressure Steam Rankine cycles. The various outputs, calculated from a given heat source data, can be used for further works in cost calculation or design of heat exchangers what is not part of this essay. This work also describes the usage of the optimising tool and errors that can occur.

Acknowledgements

This work would not have been possible without the help and supervision of my advisor Ao. Univ. Prof. Dipl.-Ing. Dr. techn.Andreas Werner. He gave me help and assistance if needed and supported me especially in the early stage of the modeling. During the whole work I felt like a respected colleague. Because of that the Institute for Energy Systems and Thermodynamics obtains this work and the optimizing tool for further research.

I would like to thank Martin Knoglinger for providing his thesis and his support in the early stage of the modelling. By choosing a proper programming language and understandable style he enabled me to build up on his work.

Thanks to my friend and team colleague Bernhard Handel-Mazzetti for proof reading parts of this essay.

I am deepest thankful to my parents Elisabeth and Thomas. Her financial support offered me to study at the Technical University of Vienna. My parents and brothers gave me emotional support during my whole education. My thanks are directed to this great family.

Finally my deepest appreciation is addressed to my girlfriend, Anna- Maria Haase. Her support, encouragement and love let me do this work.

INDEX

List of Tables

List of figures

List of equations

1. INTRODUCTION IN MODERN HEAT RECOVERY

Today's world primary energy need is at about 500EJ per year and is still rising (Hofbauer, 2010). The only times where there has been a decrease in energy need is during the financial and oil crisis's in the past (Figure 1.1 - (Orr)).

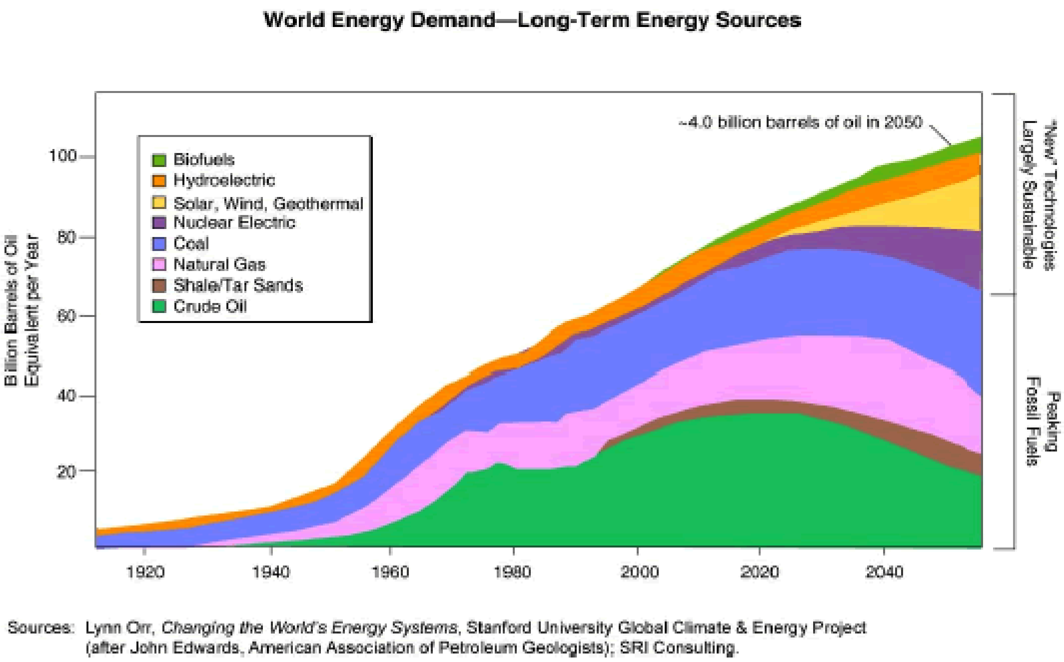


FIGURE 1.1 – WORLD ENERGY DEMAND

Several different types of energy producing plants are already in use depending on needed energy output, available energy resources and financial decisions. One often used process is the Rankine cycle, as realised in the Steam Rankine (SRC) and Organic Rankine cycle (ORC). The heat is produced by heating up air or a thermal working fluid with an energy source. This source can be thermal conversion of fuel, gas or coal, a sun collector, geothermal heat or exhaust gas of a motor. Figure 1.2 (Eurostat)shows the shares of energy production in the European Union

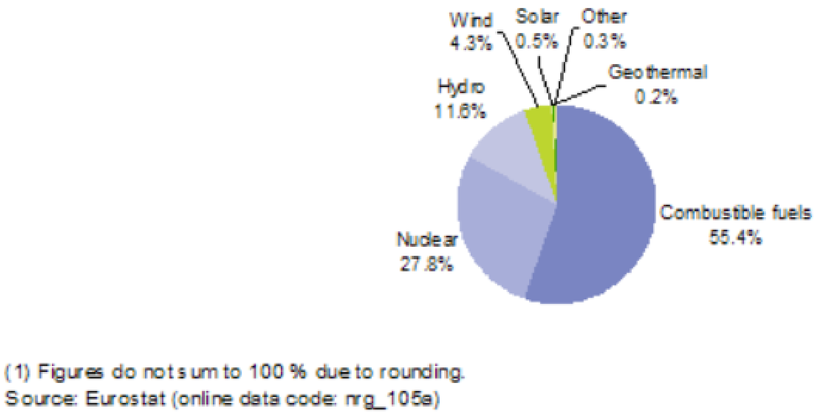


FIGURE 1.2 – ENEGRY PRODUCTION IN THE EURPEAN UNION

The basic Rankine cycle has the following steps:

- Isentropic expansion in turbine (1-2)
- Isobaric condensation (2-3)
- Isentropic compression (3-4)
- Isobaric heat transfer in heat exchanger (Economizer (4-5), evaporator (5-6), superheating (6-1))

Figure shows the T,s – diagram of an Rankine cycle and the basic flow sheet (Hofbauer, 2010):

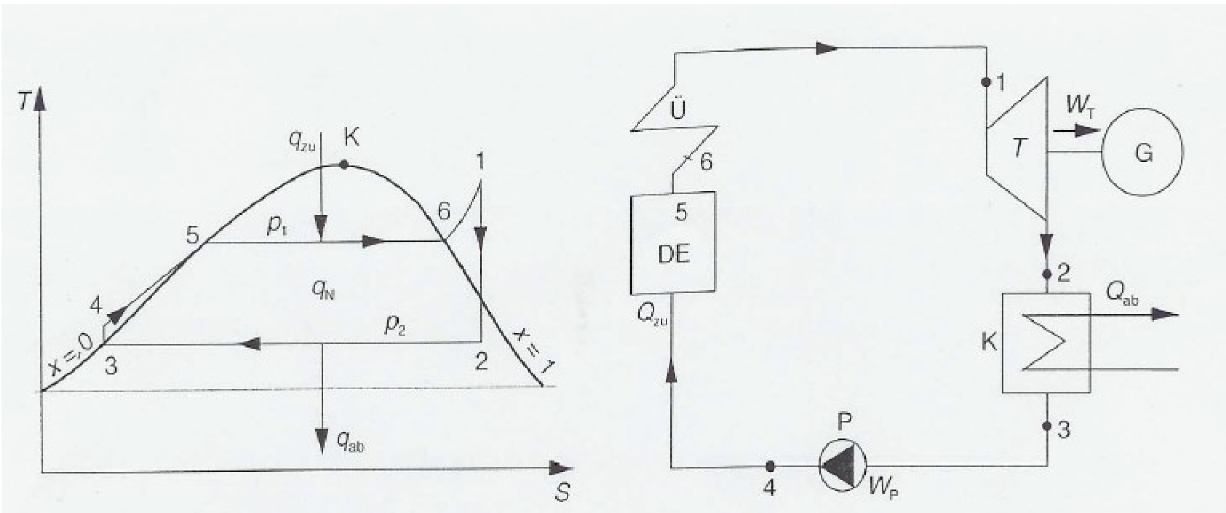


FIGURE 1.3 – T,S-DIAGRAM OF RANKINE CYCLE AND SIMPLE FLOWSHEET

The basic parts of a Rankine cycle are:

- Steam producer (DE: economizer, evaporator; Ü: superheater)
- Turbine (T) with Generator
- Condenser (K)
- Feedwater pump (P)

Differences, in the mainly installed Rankine cycles, are the working fluid used within the circuits. For the conventional Steam Rankine cycle, called SRC in this thesis, water is the working fluid and well probed. In the so called Organic Rankine cycle, ORC, an organic working fluid is used.

The ORC technology nowadays is used in several low heat power plants. One of its advantages is its ability to use heat on a lower temperature level than processes with water. Also the use in vehicles is a possible option.

Only 30% of an engine's fuel combustion energy is converted into useful work to move a vehicle and its accessory loads. The remainder is engine waste heat dissipated by the engine exhaust system, coolant system, and convection as well as radiation from the engine block. Nearly 40% of heat energy is wasted with the engine exhaust gas. If this portion of waste heat could be harnessed, energy efficiency would be significantly enhanced. Thereby vehicles all over the world could save lots of energy, which would also decrease global warming (Wang, Zhang, Ouyang, & Thao, 2011).

The use of conventional power plants with non renewable energy (fossil) is approved but with rising costs of oil, coal and gas and their impact on the environment modern power plants using renewable energy are appropriate options. In the past the efficiency of those power plants has not been satisfying but with the use of modern low heat recovery techniques, like ORC, this has improved. Nuclear power plants are a very good option concerning the energy, but the impact on the environment in case of a failure has to be taken in consideration. Table 1.1 shows the energetic efficiency, energy returned on energy invested (ERoEI) and the energetic amortisation time of different energy producing plants. Data had been found in the university work of Hermann Hofbauer (Hofbauer, 2010) and (Bull, 2010). Further references can be found in his Blog (Bull, 2010). The data used are average values and the size of the plant always has an impact on those. The efficiency of biomass, fossil and nuclear power plants is for example dependent on the energy producing facility. In this case a steam rankine cycle is used. The ERoEI is only suitable for renewable energy or if fuel is not taken in account, as fossil power plants always need a higher energy input compared to their output.

	Electrical efficiency [%]	**ERoEI [-]**	**amortisation [months]**
Coal	35-43	5.5	4
Nuclear	35-43	10.9	3
Hydroelectric	80-90	15-120	14
Wind power	30-45	25	7-16
Geothermal	15-25	20-30	
Solar thermal	13-20	9.9	70-100

TABLE 1.1 – ECONOMIC DATA OF ENERGY PRODUCING PLANTS

Geothermal power plants use the heat gradient between the earth core (3000-10.000°C) and surface temperature (up to 50°C). Because of the high temperature needed for energy production, mostly areas with a high temperature gradient are used for geothermal energy production. Power plants already using geothermal heat are "The Geysers, USA" and "Larderello, ITA". The city Rejkjavik is using geothermal heat to supply its heating system. There are several different techniques in gain geothermal or hydrothermal heat (shown in Figure 1.4 - (Green-Rock-Energy, 2009)). For power producing the "Hot-Dry-Rock" technique is commonly used (Figure 1.5 -- (Geothermal-Recources)). Here water is pumped under pressure into hot stone, usually Granit and Gneis, and then in a second pipe (approximately a hundred meters away) pumped into the plant where the heat it has adopted is used. The maximum temperatures are at about 100 to 200°C. In the work of Silke Köhler (Köhler, 2005) the comparison of the Kalina process and ORC in geothermal energy usage shows that the Kalina process performs better in lower and ORC in higher temperatures.

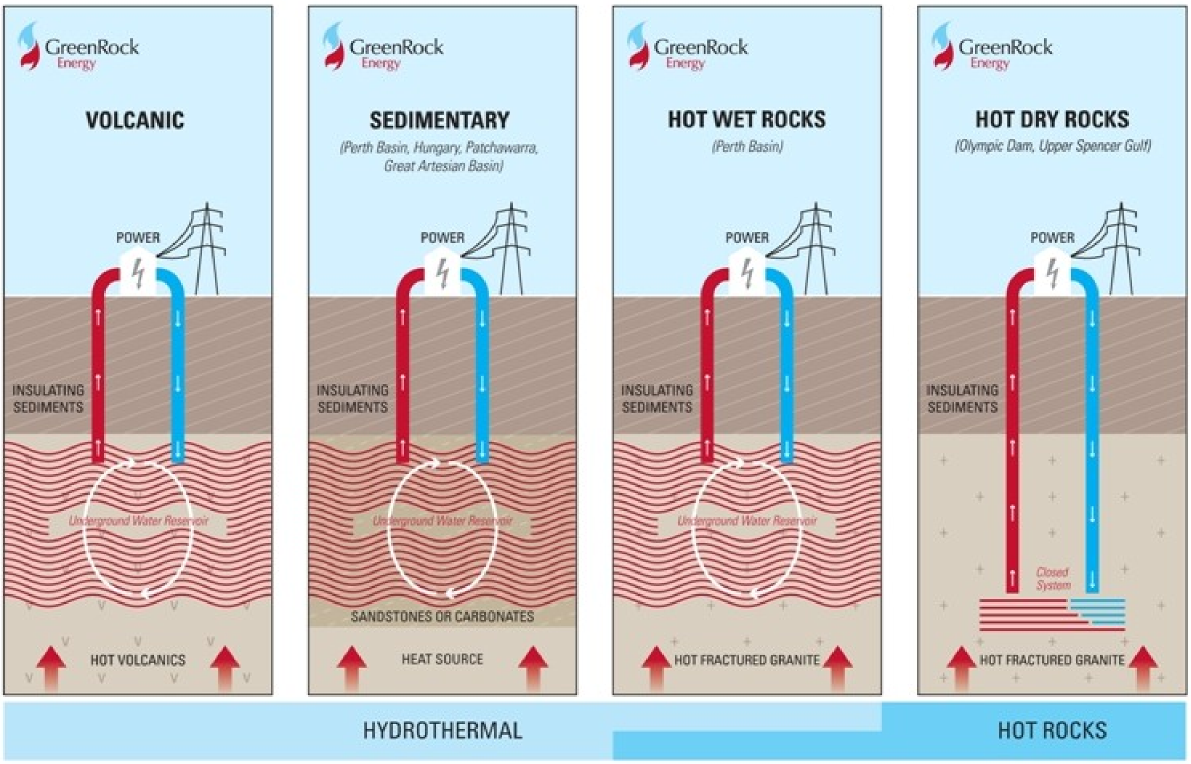


FIGURE 1.4 – GEOTHERMAL HEAT TYPES

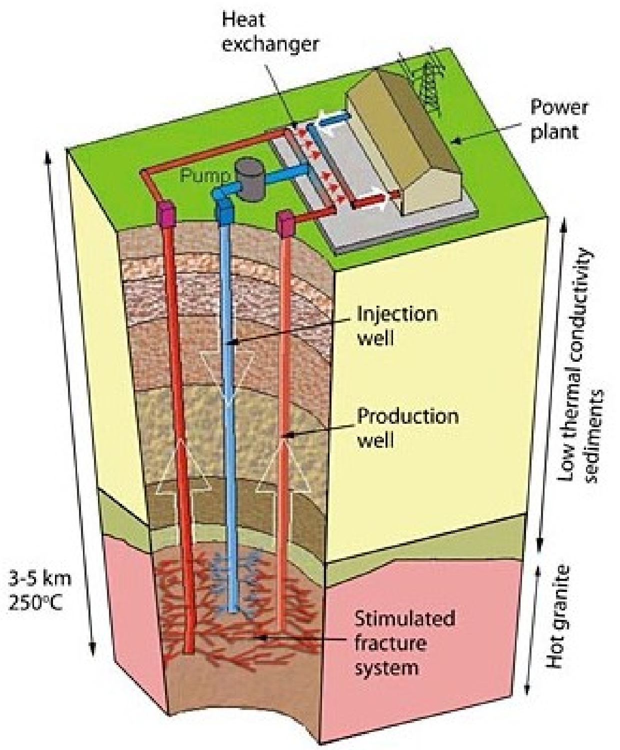


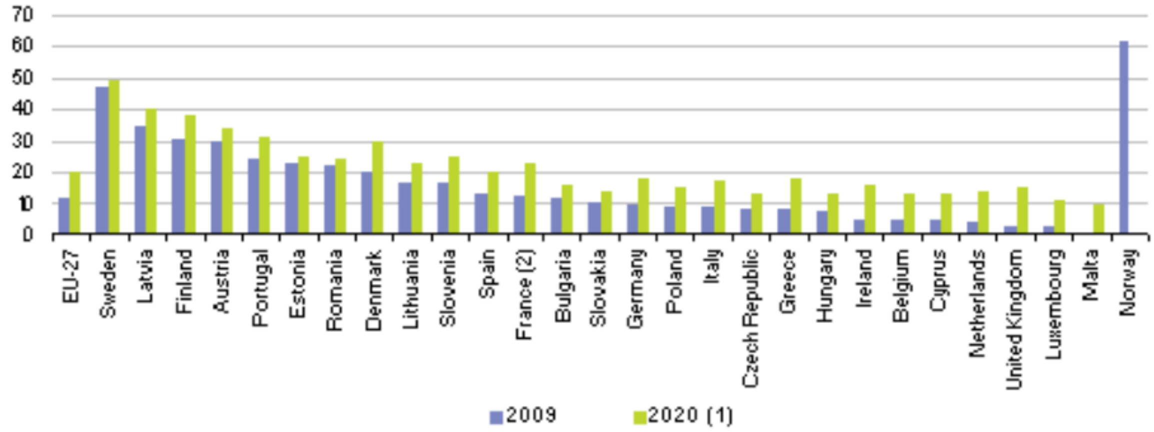
FIGURE 1.5 – HOT DRY ROCK POWER PLANT

Biomass can also be used to produce methane gas which is then refined and fed into a city gas network or burned in a gas motor. The remaining heat is used in a process like the ORC. The advantage of biomass/biogas is the neutral carbon-dioxide balance, in case of sustainable use, as the emitted CO_2 is bound by growing plants. Negative aspects of this process are the need of big areas for growing plants, high amount of chloride in fast growing plants and the use of fertilizer. If biomass is burned the temperatures are higher than usually in ORC plants but if no conventional SRC is suitable a cascade of ORC's can be implemented (Fischer, 2010).

As a third source of renewable energy the sun is often used. The sun is the main energy distributor in our solar system. After passing the atmosphere, an overall average of 219.000.000 Billion kWh reach the surface of our continents. This equals 2.500 times the yearly energy demand (Hofbauer, 2010). There are two types of solar power plants that can be used, concentrating and non-concentrating. For concentrating plants the working temperatures are 500-1200°C and for non-concentrating 100-180°C. In the process a fluid, depending on the working temperature either oil, salt or water, is running through a collector or pipes and the sun radiation, reflected by mirrors, is heating it up. The use of ORC in those plants is a proper approach. Further information can be found in Ling and Gang - Optimization of low temperature solar-thermal electric generation with Organic Rankine Cycle in different areas (Ling & Gang, 2012).

The optimal use of the remaining heat from energy or heat production is often a side target when planning a plant. In the past, the heat remaining after the main production step was cooled in water or air coolers and thereby lost. This was done because of inefficient processes in low-temperature heat recovery as well as the additional installation costs of more efficient solutions. Nowadays those costs are becoming lower, the efficiency is rising and there are bonuses in terms of environmental programs. Another influence is the so called internalisation of external costs. Looking on this issue from an environmental side, a better usage of heat means less fuel input for the same output and therefore a more sustainable use of resources. This not only means a decrease of demand for fossil, non-renewable resources but also minimizes the size of power plants.

Environmental responsibility is a big issue nowadays. The European Union plans to decrease energy use, as well as carbon-dioxide output for reaching their Kyoto goals, while increasing the amount of renewable energy (see Figure 1.6 - (Eurostat)).



(1) Indicative targets for 2020; not available for Norway.
(2) Excluding French overseas departments and territories.
Source: Eurostat (online data code: t2020_31)

FIGURE 1.6 – SHARE OF RENEWABLES IN GROSS FINAL ENERGY CONSUMPTION, 2009 IN THE EU

The targets are called "20-20-20" and aimed at 2020. As a result the European Union wants to promote an economy that respects the planet's resources, implement a low carbon system, improve its energy independence and strengthen security of energy supply. Part of this plan besides better energy transport efficiency, financial support for reducing energy consumption of buildings and improving energy performance is improving the energy transformation in facilities. It also plans to develop, in cooperation with industry professionals, guidelines for good practices designed both for existing facilities as well as for energy suppliers and distributors. The promotion of cogeneration will also be fostered and connections with decentralised generation centres will be encouraged (Union, Summaries of EU legislation - Energy efficiency). The plan promotes taxes as a powerful tool. As the boost of renewable energy is a core target, it is addressed in the so called "Global Energy Efficiency and Renewable Energy Fund" helping to mobilise private investments in energy efficiency and renewable energy projects. Papers and directives can be found at the pages of the European Concoucil (Concoucil, 2012) and Energy department (Union, European Union/Energy). Figure 1.7 - (Eurostat) shows the electricity generated from renewable energy recourses and the participation of different sources.

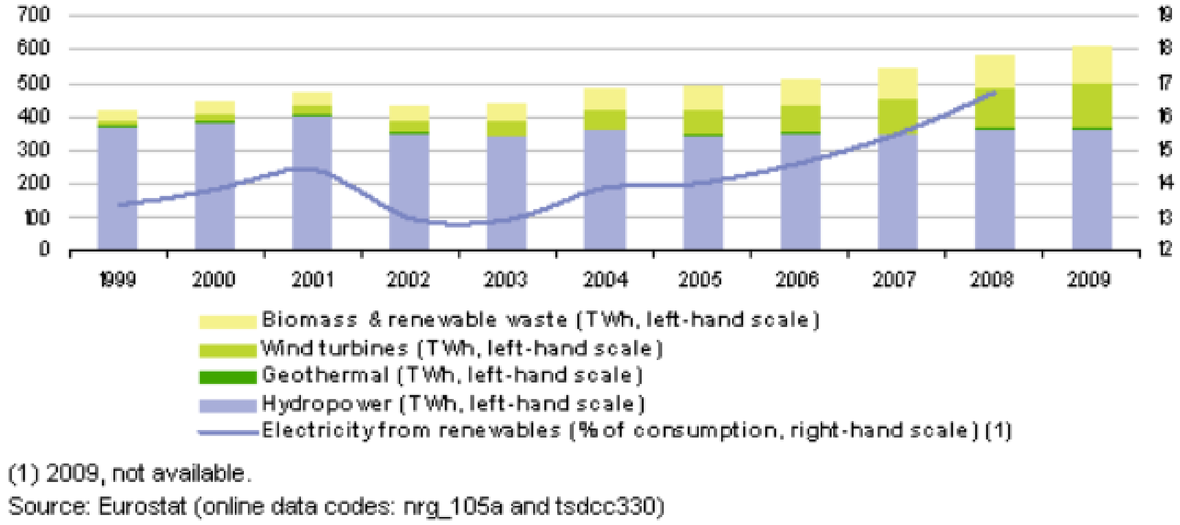


FIGURE 1.7 - ELECTRICITY GENERATED FROM RENEWABLE ENERGY SOURCES, EU-27, 1999-2009

As illustrated high energetic efficiency and ideal use of resources are the main targets in today's planning and operation of energy or heat plants. Better efficiency brings financial bonuses, in terms of lower taxes and start-up investments, as well as a lower demand of energy input. The boost of renewable energy affords a higher usage of low temperature heat and therefore improved processes, like Kalina or ORC for conversion.

2. Scope and target of thesis

The main objective of this work was to develop an optimising tool for **O**rganic **R**ankine and **S**team **R**ankine **C**ycles, as well as adding the functionality of the tool programmed by Martin Knoglinger. The main focus was on the supercritical ORC cycle, as a new perspective in waste heat recovery. Fluegas, in certain compositions, was supposed as main heat source. A thermal oil circuit for the ORC cycle is commonly used to operate the fluegas heat exchanger under atmospheric pressure. Today's low temperature ORC's also use thermal oil as main heat source and this feature was added too.

As PYTHON was already used by previous work of Martin Knoglinger, and seen as a proper choice for development, it has been also used in this thesis. PYTHON (Python Software Foundation) is an easy understanding programming language, is free to use in private and commercial programs and makes the tool expendable. For adding future features, like cost calculations, the tool provides heat transfer data. The tool gives a first-look estimation about how much power in terms of electricity can be produced.

Main targets in all added optimisations:

- Find maximum power output of cycle
- High thermal efficiency

3.1 Sub- and supercritical ORC

The subcritical ORC process and several features, like temperature-entropy and other diagrams had already been implemented by Martin Knoglinger (Knoglinger, 2011). The aim was to add some missing bonus features like printing and report functions. As discussed in his work the maximum pressure, given by security issues, is 20bars and proper for today's ORC plants.

For the supercritical ORC the main objective was to find an optimisation to use in PYTHON, leading to maximum results in meaning of power output. The use of the same organic fluids like used in the subcritical ORC was seen as a proper approach. Propane can also be used in future ORC facilities and so added to the program. Because of the differences in the dew behaviour a different optimisation start for propane was suggested. During the expansion in the turbine the formation of droplets, leading to blade erosion, should not appear and was respected by optimising procedures.

Thermodynamic boundaries chosen for the cycles are:

- ORC subcritical:	maximum pressure:	200kPa	20bar
	minimum pressure:	1kPa	0,01bar
- ORC supercritical:	maximum pressure:	fluid limits (see Table 2.1)	
	minimum pressure:	1kPa	0,01bar

Maximum pressures of the supercritical cycle can be reached but then a different cycle fluid with higher critical points is suggested.

Fluid	P_{max} [MPa]
Pentane	100
Isopentane	1000
Isobutane	35
Octamethyl-trisiloxane MDM	30
Toluene	500
Cyclohexane	80
Cyclopentane	200
Propane	1000

TABLE 2.1 – MAXIMUM PRESSURE OF SUPERCRITICAL ORC CYCLE

3.2 STEAM RANKINE CYCLE

The SRC is an often used cycle in the industry for energy production. There are several different types of cycles and everyone with his advantages. Main target was to add the single and dual pressure cycle to the program and at first proper cycles for a first-view calculation needed to be found. As heat source fluegas is usually used and has been implemented in the optimising tool. Due to leakage in pipes and other process parts, gas especially Oxygen, is solved in the cycle water and needs to be eliminated. Therefore deaeration is implemented. Nowadays the combined use of facilities as electric power and heat supply for district heating is often used, leading to higher efficiency. To add this feature was also suggested. Blade erosion is also an issue in the SRC but because of the bigger turbines a maximum moisture content of 10% was seen as a proper limit.

Thermodynamic boundaries chosen for the cycles are:

maximum pressure:	can be chosen at input	suggested: 13000kPa	130bar
minimum pressure:		1kPa	0,01bar

3.3 OVERVIEW WORKING FLUIDS FOR ORC

The main difference between an ORC and a SRC is the use of an organic substance as working fluid. Organic fluids have lower critical points and boiling temperatures than water, making it suitable for low grade heat recovery. When choosing a working fluid other aspects like environmental suitability, health and safety aspects as well as costs should be taken in account. There are many studies already done relating to ORC and were mentioned in the pre-work of Martin Knoglinger (Knoglinger, 2011) on page 1. The article in the Elsevier magazine (Wang, Zhang, Ouyang, & Thao, 2011) shows the performance of different working fluids operating in specific regions was analyzed using a thermodynamic model built in MATLAB together with REFPROP.

The study of Borsukewicz-Gozdur (Borsukiewicz-Gozdur, 2010), on advantages of supercritical ORC cycles, shows that a higher output and efficiency compared to a subcritical ORC can be achieved. The safety issues, especially the higher pressure in combination with potential flammable and deleterious fluids, should be taken in account but the better performance is needed to suit to today's environmental needs. For the supercritical ORC process the investigation of the Kalsruher Institute of technology (Vetter, Wiemer, & Kuhn, 2011) shows that a high power output and a high thermal efficiency in electricity producing power plants is achieved with propane and isopentane. They mention propane as a proper working fluid with a low impact on the environment.

Table 2.2 shows critical temperatures, boiling points and molar mass of organic fluids used in this work. These data was taken from REFPROP, a thermo physical fluid database, provided by the National Institute of Standards and Technology (Lemmon E. H., 2010). Same fluids are used in the preceding work of Martin Knoglinger (Knoglinger, 2011). In his work the advantages of each fluid for the subcritical ORC process is explained.

	CAS number	**Molar mass [kg/kmol]**	**Tcrit [K]**	**pcrit [kPa]**	**Equation of States (EoS)**	**max. Temp. by EoS [K]**
Isopentane	78-78-4	72.149	460.39	3369.6	(Polt, 1973)	589
Isobutane	75-28-5	58.122	407.81	3629	(Buecker, 2006)	575
Octamethyl-trisiloxane	107-51-7	236.53	564.09	1415	(Colonna, 2011)	673
Toluene	108-88-3	92.138	591.75	4126.3	(Lemmon E. W., 2006)	700
Cyclohexane	110-82-7	84.161	553.64	4075	(Penoncello, 1995)	700
Cyclopentane	287-92-3	70.133	511.69	4515	(Gedanitz, 2008)	600
Pentane	109-66-0	72.149	469.7	3370	(Span, 2003)	600
Propane	74-98-6	44,069	369,89	4251,2	(Lemmon, McLinden, & Wagner, 2009)	650
Water	7732-18-5	18.015	647.1	22064	(Wagner, 2002)	2000

TABLE 2.2 – CRITICAL PROPERTIES OF WORKING FLUIDS

One advantage of an ORC fluid is the high molecular mass and therefore small sized units leading to overall lower installation costs. The use of a heat transfer oil circuit between the heat source and the ORC fluid is common and leads to extra costs. Today's low temperature energy facilities, like geothermal and solar thermal plants, don't necessarily need that anymore.

Crucial for the optimisation is the slope of the dew line in the Temperature-Entropy (T,s) diagram. Therefore substances can be classified in two groups, wet and dry fluids. Wet fluids have a negative and dry fluids a positive inclination of the dew line. Organic fluids can have positive, negative or almost isentropic inclinations. The steam curve of water is negative in the T,s-diagram. For this work mainly dry organic fluids were used what brings advantages in optimisation. For example drop erosion on the turbine blade is crucial in power plants. Due to the positive inclination superheating, used in conventional steam cycle processes to allow expansion to the superheated steam area and eliminating droplets, is not needed in the chosen ORC cycle. The lower maximum temperature can lead to a higher turbine output as explained later.

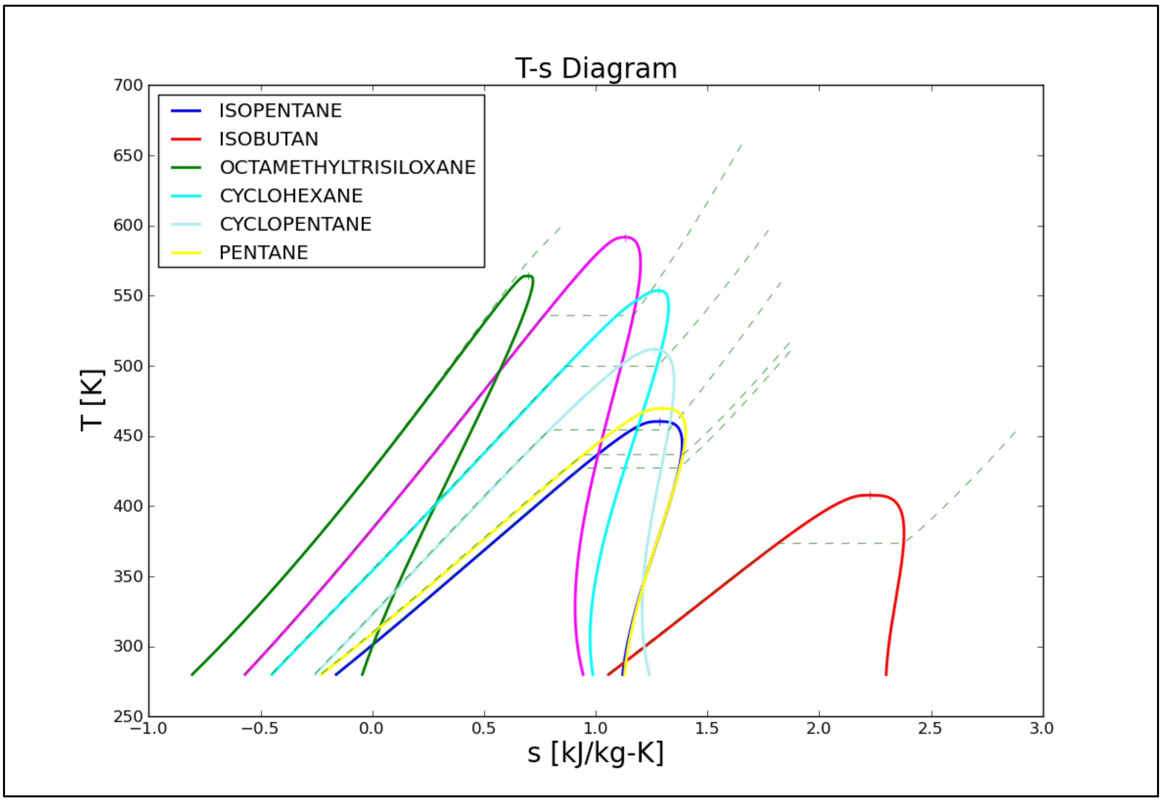


FIGURE 2.1 – T,S-DIAGRAM OF DIFFERENT WORKING FLUIDS

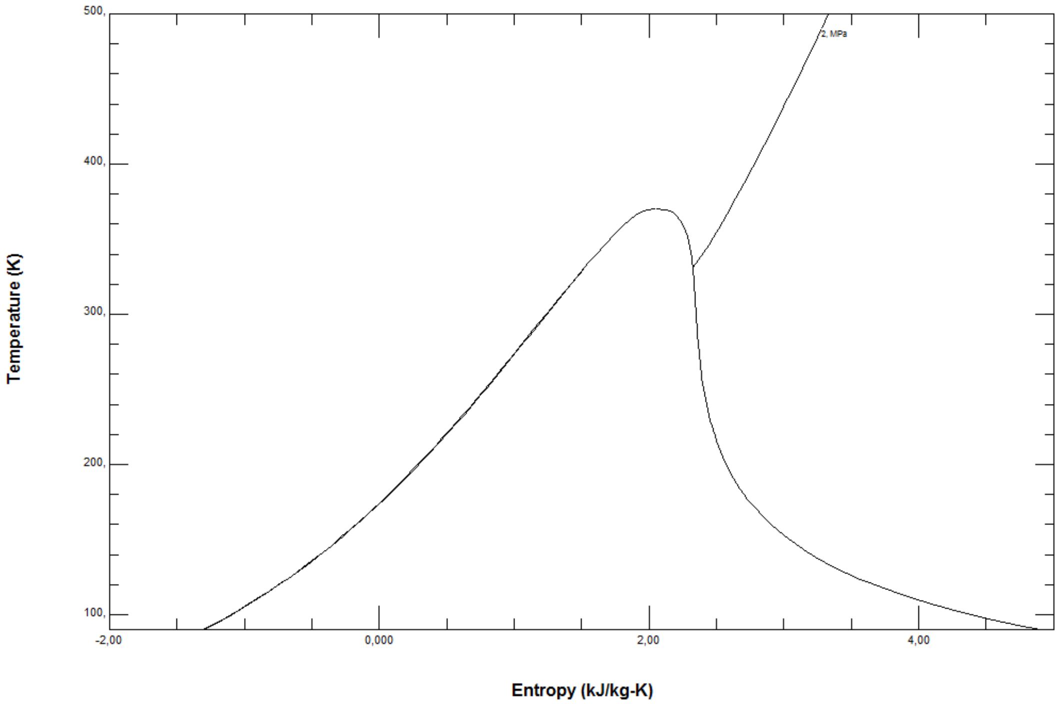


FIGURE 2.2 – T,S DIAGRAM OF PROPANE

Figure 2.1 and Figure 2.2 show the T,s-curves of working fluids used in this work (from REFPROP). Additionally, it should be mentioned that the organic fluids used do have a smaller latent heat in comparison to water. To get an optimum result the optimisation inputs should be varied and especially different fluids should be checked. Because REFPROP is used as the fluid database and only a short code is needed, every fluid included in REFPROP can be implemented into the program.

3.4 SUMMARY OF PREVIOUS WORKS

As already mentioned before, this thesis is building up on previous works by Martin Knoglinger (Knoglinger, 2011) and Erich Opitz (Opitz, 2011). Mainly the program of Knoglinger was used as the base for the ORC and SRC optimisation tool. The work of Opitz was used as literature to get a first idea on how to optimize a SRC cycle. Because EES (F-Chart Software)(was used for finding maximum output and the available variety of variables to optimise is bigger than in the "ORC and SRC optimising tool", programmed in PYTHON, the decision was to build up a new SRC optimisation tool.

The work of Martin Knoglinger is an optimising tool for subcritical ORC cycles with a maximum pressure of 20bar as well as a parameter study of proper organic working fluids for waste heat recovery to use in those ORC cycles. The tool offers the same organic fluids as chosen for this work except propane. It gives the opportunity to define fluegas temperatures and mixtures as well as minimum and maximum temperature limits, pinch points and efficiency of turbine and pump. A pressure drop simulation can be done and all boundaries for the cooling system defined. The implementation of an IHE (Internal heat exchanger), as often used in today's power plants, is available. The optimisation is slightly different than the one used for the supercritical ORC and used a BRENT or FSOLVE calculation to find the cooling system and maximum pressure limits inside the given boundaries. The maximum power output is found by increasing the superheating temperature by 1°K per step and then searching for the maximum output in this array. As part of his work the dew point of the fluegas can be calculated by giving the program a mass or volume percentage of its contents, resulting in a minimum temperature limit for corrosion free cooling. The outputs of the program are not only temperature and pressure state points and limits. It also calculates massflowrates, kA values and entropy loss in the heat exchangers for further planning and cost calculations. An option to print an temperature-entropy and an enthalpy-temperature diagram is also available and should be completed by a heat-temperature diagram (Q,T). Possible improvements in the program are the variability of the maximum optimisation pressure. It optimises well on 20bars as a security limit, but nowadays the issues are solved by better materials and safety precautions. Because this tool should give a basic overview of a possible ORC plant there should be a printing function as well an overall report. Low temperature ORC cycles use different working fluids, like molten salt, liquid metal or thermo oil, instead of fluegas as heat source. To implement a wide range of working fluids would use lot of time but a possible implementation of only an oil circuit as heat source is a possibility. Because of the programming style he used an adaption of maximum pressure and installed fluids is not a long time work as long as it stays in the subcritical area.

The work of Erich Opitz, using EES and VBA, has a different approach. Mainly he described different SRC types and added a subcritical ORC cycle. Because EES supports the solving of formulas on all variables, his work is more the thermodynamic characterisation of an SRC and ORC cycle. The tool can maximise and minimise power output, pressures, massflowrates and thermal efficiency. Boundaries that should be varied can be chosen and limits set. This approach on equations makes the tool very versatile. There are much more inputs needed than in the tools programmed by PYTHON possibly leading to a more exact result but also to more calculation time and usage of resources. Output of several diagrams, kA values and destructed exergy is also available. An overall report is not implemented. For the ORC the same boundaries and assumptions for maximum pressure and the cycle itself as in the work of Knoglinger where taken. For the SRC cycle a maximum pressure of 120bars was used. Future work on this could be adding further fluids over REFPROP and making more types of cycles available. The minus while using EES is that it is not a cost free program and therefore not always available.

To get one optimising tool the decision to implement all in one PYTHON program was made. The easier accessibility of PYTHON and output abilities overcome the less variability and possibility of solving thermodynamic equations compared to EES.

3.5 PROCEEDING WORKS

Based on the works discussed before the optimising tool was built up and some new features were added. The optimising tool now supports ORC sub- and supercritical cycle and two different SRC cycles. For better overview and expendability the whole program folder was divided in packages for Designs, tools and the main programs.

The work for the ORC optimising tool was mainly on the supercritical cycle and therefore only some small extras were added to the subcritical calculations. Functions added to the ORC subcritical cycle are:

- Report function
- Printing function of diagrams
- Printing function of reports
- Printing function of Flowsheets with calculation result

The supercritical cycle type chosen is similar to the one used in the subcritical optimisation. Also an option with IHE was added. The big difference is that there is no need for an evaporator and superheater. Figure shows both cycles:

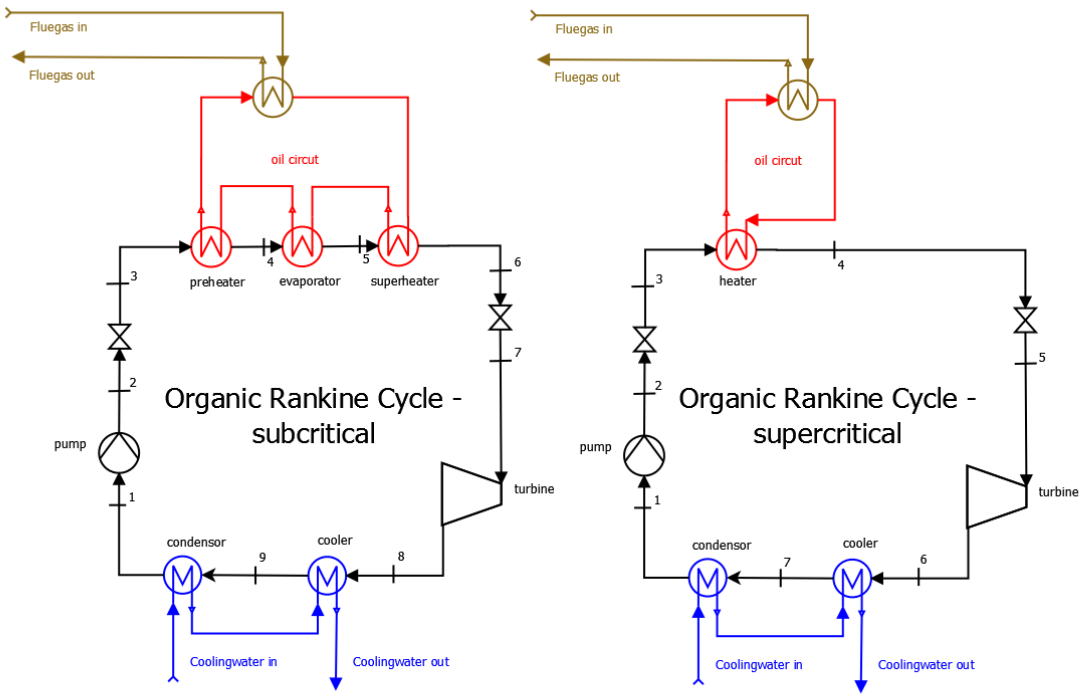


FIGURE 2.3 – FLOWSHEET OF THE SUB- AND SUPERCRITICAL ORC

Functions like calculation the flue dew-point, kA-values and destructed exergy were taken from the work of Martin Knoglinger (Knoglinger, 2011) and adapted. Tools for checking the correct pressure levels and ensuring that the minimum temperature limit in the heat exchanger stays observe were programmed. A database of fluid information, like s_{max} as explained in a later chapter, for faster calculations was written. The Design of in- and output windows stayed the same but some extra buttons were added. The report function was programmed, making possible to read all given and calculated data on one page. To make it available for future planning or presentations they can be printed. The code for printing an T,s or h,T –diagram was written to make it printable. Differences between the sub- and supercritical ORC makes a direct use not possible. In the end a numeration of state points on the diagrams was added. The output of a Q,T – diagram gives a good look on the Temperature differences in the heat exchanger. ORC supercritical optimisation only works inside certain fluegas temperature limits, depending on the organic working fluid used, so an option to use subcritical calculations, in case the supercritical fails, is also part of the ORC program.

The first plan was to convert the SRC EES program directly into PYTHON language and add the same features to the optimising tool. Because of the different interpretation and programming style, PYTHON needs a stepwise or object oriented programming of all calculations and output and EES/VBA the equations and boundaries which get automatically sorted and printed by EES, a direct use was not possible. The work was used as a first overview of what is important and which assumptions and boundaries should be taken. The SRC optimising tools builds up of that knowledge but for the optimising itself, especially the live steam temperatures and pressures different literature was used.

For the SRC optimising tool a single and a dual pressure cycle were implemented. Feedwater preheating was not programmed because it leads to a lower turbine output, what was the main target in this work. Feedwater preheating leads to a lower use of fuel but also to a sharp decrease of output (see Figure 2.4 - (Kehlhofer, Hannemann, Stirnimann, & Rukes, 2009)). Future work can add this feature for conventional plants and fuel saving issues. The thermal deaeration which is implemented has the same effect and is leading to a higher feed water temperature and therefore minimizing the power output. Deaeration is needed for the proper work of the plant to prevent corrosion because of oxygen in the feedwater. The used dual pressure SRC cycle is differing in its complexity from those in the EES tool and was chosen because of optimising issues. Figure 2.5 shows the differences in the dual pressure cycle. The basic difference is that the high pressure and low pressure circuit are totally separate. Only the feedwater tank is used together and therefore also deaeration.

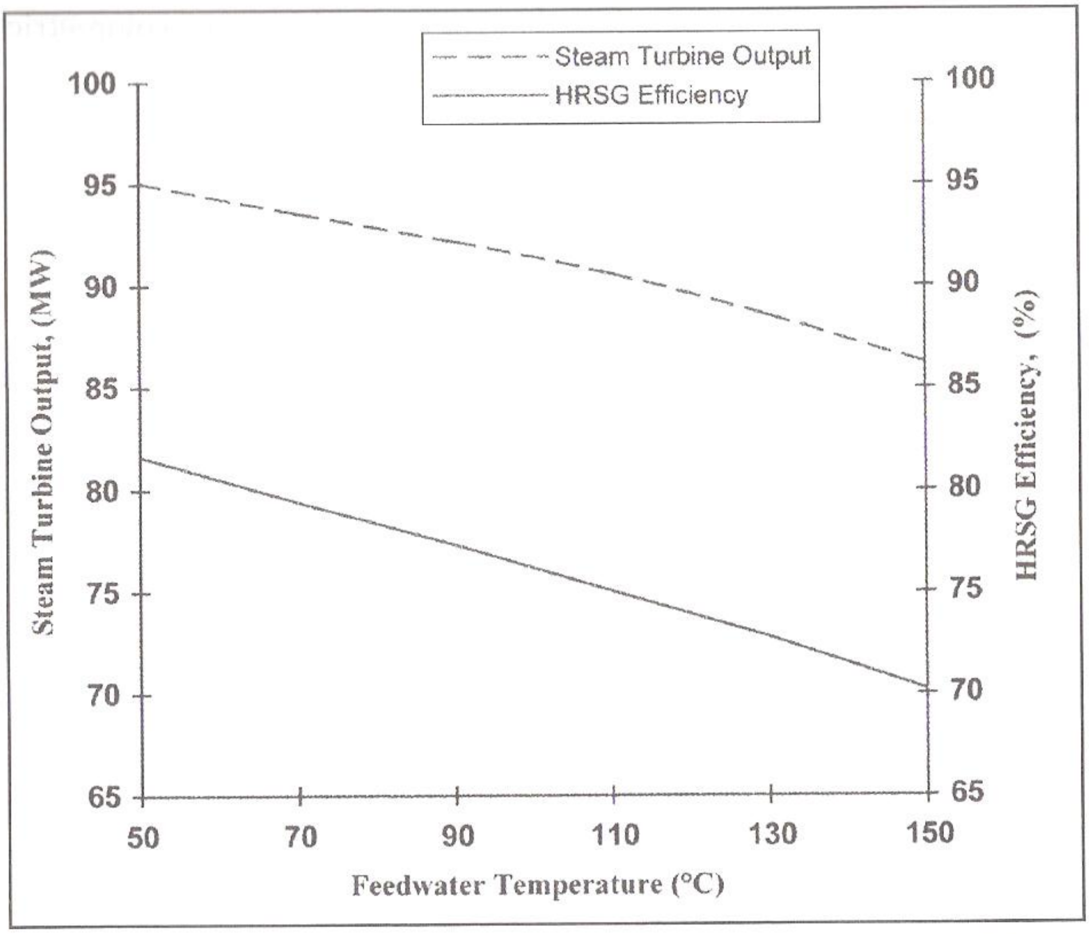


FIGURE 2.4 – DECREASE IN TURBINE OUTPUT BY HIGHER FEEDWATER TEMPERATURE

Functions implemented in the SRC cycles are:

- Output of results on a flow sheet
- T,s-; h,T- and Q,T diagrams
- Report function
- Printing function of diagrams
- Printing function of reports
- Printing function of Flowsheets with calculation result

The flowsheet for the high pressure and low pressure circuit in the dual pressure cycle with calculated results, the Report and all diagrams are on separate pages because of space issues. Output like kA Values, transferred heat in certain heat exchangers and destructed exergy is calculated as well and shown in the results.

Overall a optimising/engineering tool, with several in- and outputs, diagrams, report and printing features, for ORC sub- and supercritical as well as single SRC and dual pressure plants was set up.

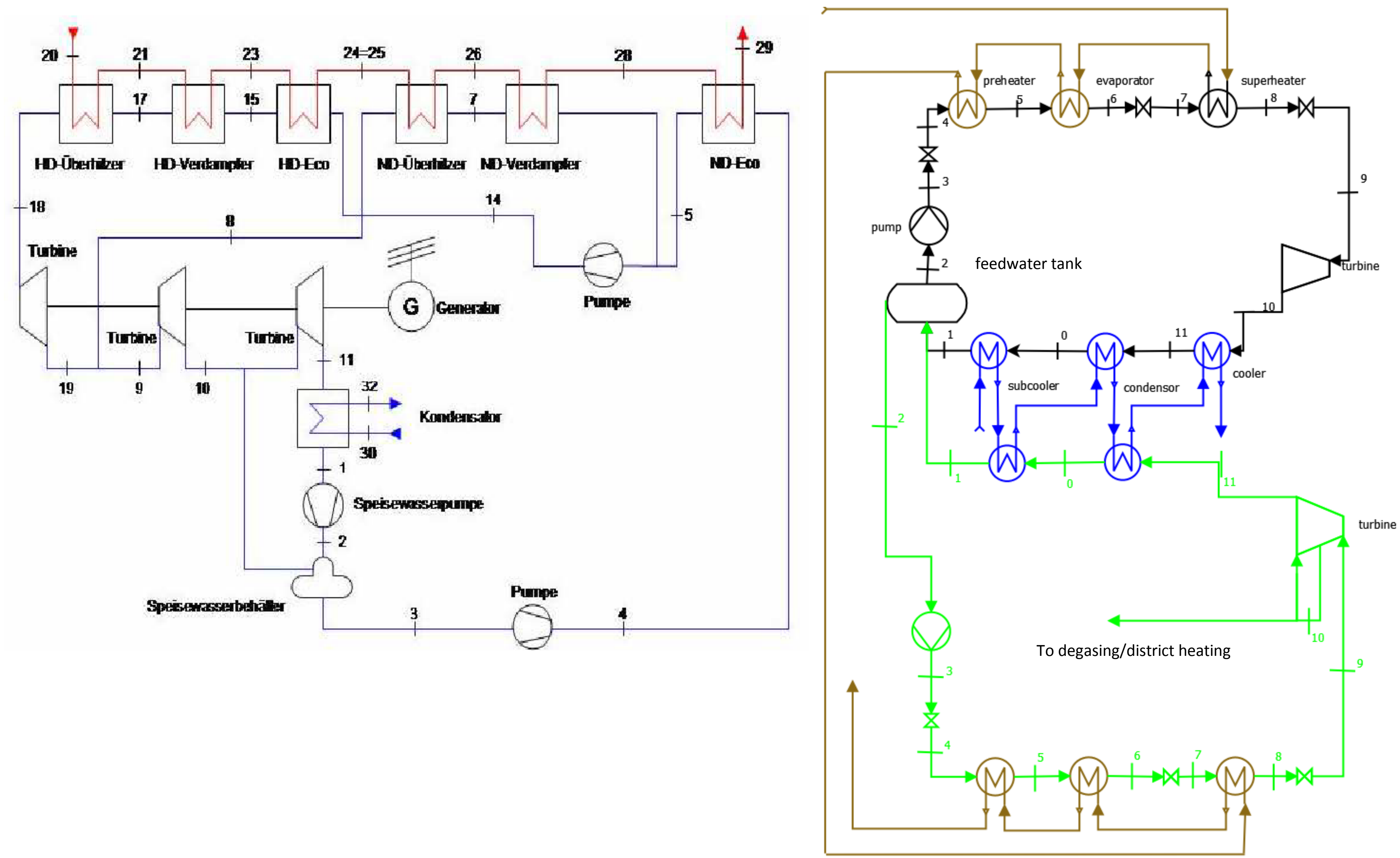


FIGURE 2.5 – COMPARISON OF SRC CYCLES

3. MODELLING OF THE OPTIMIZING ALGORITHMS

For an exact thermodynamic simulation and optimisation of a rankine cycle different tools like EES, Aspen Plus or others are available. Because of the costs for such a program, the need for an easy to use **G**raphical **U**ser **I**nterface – further called GUI – and the requirement to have an easy extendable source code the open source language "Python" is used. The optimisation algorithms take some shortcuts for easy calculating known from technical expertise explained further in this chapter.

Because of its requirements the "ORC and SRC Optimising tool" doesn't gives a full range of variations in the input and design of a cycle.

4.1 SOFTWARE

4.1.1 REFPROP

The NIST Reference Fluid Thermodynamic and Transport Properties Database called "REFPROP" is a program not really containing any experimental information aside from critical and triple points of pure fluids. With equation for the thermodynamic and transport properties it calculates state points of pure fluids and mixtures. It provides very accurate equations and is so used in different programs, like Matlab, Python and many others worldwide. More information can be found at the Homepage of the National Institute of Standards and Technology (NIST).

The REFPROP tool available from the NIST site is only working for Windows but there is an external utility called SVN to make it working on a Linux computer. For many programming languages there are also application programming interfaces to link a code to REFPROP. In this work an interface by Bruce Wernick was used and the permission given by him. So there was no further work need to been done to connect the optimizing tool to the REFPROP equations. In this thesis REFPROP is used to calculate entropy, enthalpy, temperature and pressure for different state points.

4.1.2 PYTHON

PYTHON was used because parts of the ORC optimizing tool have already been coded by Martin Knoglinger in this open Source language. Beside its free open Source license, also for commercial programs, the advantages of PYTHON are the big district and so an easy way to find solutions to upcoming problems, the expendability with packages like scipy (http://www.scilab.org) and numpy for scientific and numerical calculations and the possibility to program object oriented but also to implement procedural programming. PYTHON has a clear and easy to understand syntax and there are a lot of videos and text tutorials available on the Internet. The Scipy package covers some solvers, but they are mostly unconstrained. Nevertheless, in order to calculate the maximum power the program uses one scalar function minimiser, called Brent method as explained in the chapters later. Information about provided solvers in PYTHON generally is given in scipy documentation (Python Software Foundation).

PYTHON is an interpreter language meaning it allows checking immediately their written code line for line. It is also applicable in many different development environments. The Standard download of PYTHON provides an integrated Development environment, called the PYTHON shell. The standard PYTHON package can be downloaded at (Python Software Foundation). Because development of complex PYTHON programs need a powerful debugging tool, what isn't supported by the standard IDLE, NETBEANS (NetBeans) was chosen as a development environment in this thesis. Tutorials and

Information can be found at (Oracle, 2012) and the IDE downloaded from the company's homepage . Net Beans provides an environment for many different programming languages. To link PYTHON to NETBEANS there is a plug-in by the NETBEANS district which can be loaded within the NETBEANS main program. At the start time of this thesis the NETBEANS plug-in for Python was only available in the Version 6.9.1 or lower and not in the newer ones available on the market. For further work on the optimizing tool a newer version of NETBEANS can be used although the differences will be marginal for the given problem.

The download at (Python) provides an advanced development as well as already preinstalled packages (numpy, scipy) within PYTHON. Therefore it is recommended to download the software at this webpage. Once PYTHON is installed, the user is able to start coding.

In this thesis PYTHON version 2.6.5.5 was used, although the more developed version 3 is already available on the internet. Furthermore PYTHON 3 is not compatible with earlier versions and also not with the thesis and work of Martin Knoglinger. However, PYTHON 2.6.5.5 has been considered suitable enough for given ORC problem.

4.1.3 QT AND PYQT

The Qt SDK itself is first developed by Nika (Nokia Qt Development Frameworks) to streamline the creation of applications for symbian and N9 smartphones. The PyQt (Python Qt) is Python bindings developed by Riverbank Computing Limited (Riverbank) for the Qt cross-platform GUI/XML/SQL C++ framework. It provides a wide range of over 600 classes that cover graphical user interfaces, XML handling, network communication, SQL databases, Web browsing and other technologies in Qt. PyQt is distributed under a choice of licences like GPL version 2 or version 3 or a commercial one. The Designer permits the design of Graphical User Interfaces with easy drag and drop movements. The created file by the QT Designer is an XML File which makes it usable in different programming languages. To use it in Python the XML file needs to be transferred in a Python '.py' file. Documentation can be found at http://www.riverbankcomputing.com/static/Docs/PyQt4/html/. As a handbook for Qt programming the Guide by Summerfield (Summerfield, 2008) was used

4.1.4 OTHERS

In addition to the software mentioned above MS Word, MS Excel and Dia were used. Dia, a free software tool (Foundation, 1998-2000) used for creating the flowcharts, MS Excel for table calculations and MS Word (Microsoft)for writing the thesis.

4.2 Decision-making sub- or supercritical ORC process

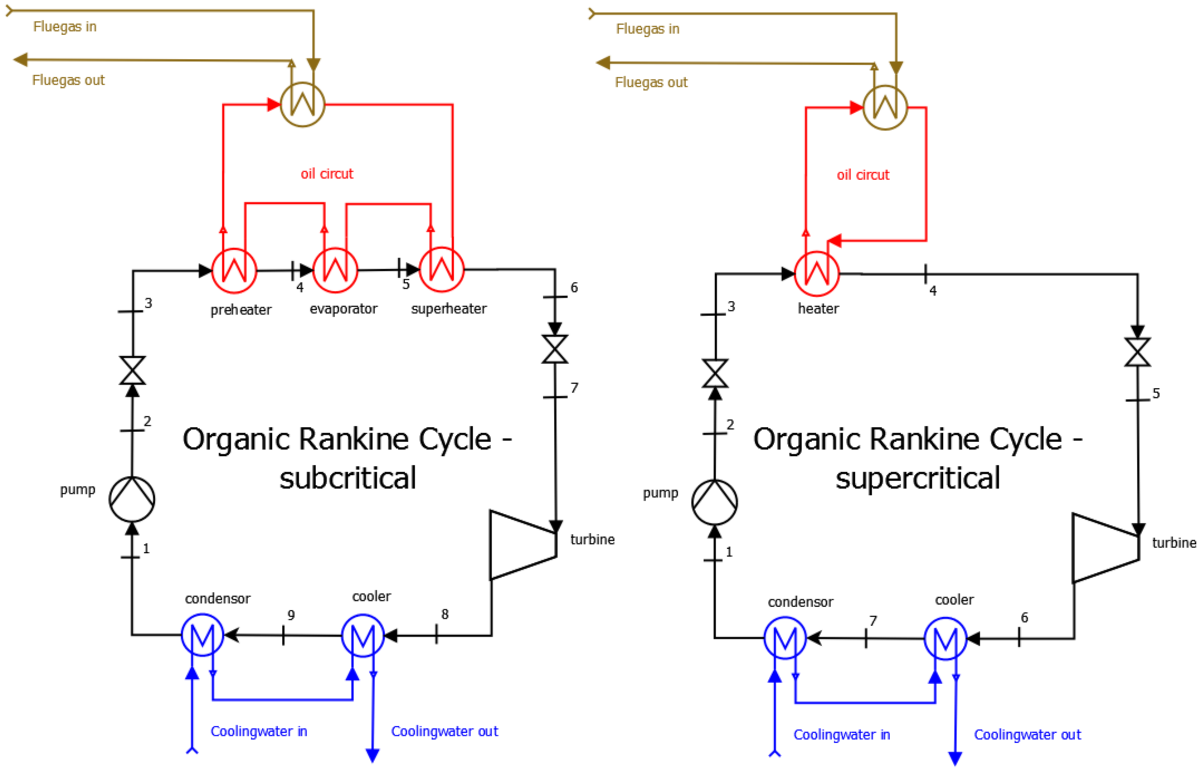


FIGURE 3.1 – COMPARISM SUPER-/SUBCRITICAL ORC

As mentioned in chapters before there are several advantages on using one or the other type of an ORC. To decide whether to use a subcritical or supercritical ORC process from the optimizing point of view the temperature level of the fluegas is the main interest.

One of the boundaries for the supercritical optimisation is that the pressure found needs to be at least 10% higher than the critical pressure of the used fluid. So depending on the Fluid there are certain minimum levels for $T_{ORC,high}$ – the temperature at the top end of the superheater (for subcritical) or heater (for supercritical) - that can be calculated by knowing S_{max} (see later in chapter 4.3.2.2 and Table 3.7)and the minimum pressure or isobar that is 10% in the subcritical area. The results for that are in Table 3.1

Fluid	S_{max} [kJ/kg*K]	P_{crit} [MPa]	P_{min} [MPa]	T_{min} [°K]	T_{min} [°C]
Pentane	1,4005	3,37	3,707	478,20	205,05
Isopentane	1,3865	3,378	3,7158	468,75	195,60
Isobutane	2,3787	3,629	3,9919	417,07	143,92
Octamethyl-trisiloxane	0,71909	1,415	1,5565	570,59	297,44
Toluene	1,1998	4,1263	4,53893	601,54	328,39
Cyclohexane	1,325	4,075	4,4825	561,85	288,70
Cyclopentane	1,3504	4,515	4,9665	521,27	248,12
Propane	Defined by cooling temperature	4,2512	4,67632	Defined by cooling temperature	Defined by cooling temperature

TABLE 3.1-MINIMUM ORC HIGH TEMPERATURES FOR SUPERCRITICAL OPTIMISATIONS

To get the minimum temperature for the fluegas inlet the two pinch point temperature differences (Oil/Fluegas and Oil/ORC) are added –this is the guess value for ΔT_{min} between the T_{high} of all heat exchangers. For example a Pinch point for Oil/Fluegas of 40°K and for Oil/ORC of 10°K leads to a minimum temperature of 428,20°K or 255,05°C for the fluegas when pentane is the used fluid. Fluegas temperatures below that level cannot be optimised with the supercritical algorithms and to use the subcritical optimisation is suggested.

For the optimising itself there are no limits to the top fluegas temperatures but it is always to mention that there is a maximum pressure level for each liquid and so the gap between the fluegas cooling and the ORC fluid heating curve is getting bigger by increasing the fluegas temperatures leading to a greater exergy destruction. In table the maximum applicability of the chosen fluids are shown. The maximum temperature of those fluids are technical not in use because then an SRC cycle is usually implemented.

Fluid	P_{max} [MPa]	T_{max} [°K]
Pentane	100	600
Isopentane	1000	500
Isobutane	35	575
Octamethyl-trisiloxane MDM	30	673
Toluene	500	700
Cyclohexane	80	700
Cyclopentane	200	600
Propane	1000	650

4.2.1 *Summary of Boundaries for the supercritical ORC cycle*

In order to figure out the maximum power output as well as giving a simple and clear input to the user some restrictions have been taken. Some of them were mentioned before.

- Pinch Point in condenser
- Pinch Point at heater to thermal oil
- Maximum oil temperature of 285°C
- Minimum temperature of ORC fluid – as explained before
- no droplets during expansion in turbine

Following assumptions are made in the program:

- steady state in all components
- Isentropic simulation of irreversibility in turbine and pump
- No heat losses in heat exchangers except IHE
- Adiabatic compression/expansion in pump/turbine
- Pressure drop simulation can be done by using the throttles implemented in each configuration
- Fluegas and oil curves run parallel in heat exchangers so a check calculation over the whole Oil/Fluegas heat exchanger is not needed
- No heat losses in pipes and other parts of the plant

For the cooling system the knowledge of the inlet and outlet temperature is assumed. For the outlet temperature National restrictions of some countries can be used as input. Simple water cooling is implemented and other cooling applications such as wet or dry cooling towers have not been considered in this thesis.

In Figures shown the thermal oil, flue gas and cooling water are shown as lines in the T, s-diagram. They visualize the relation properly but do not represent reality. For the given target this wasn't needed.

4.3 THE OPTIMIZING ALGORITHMS

The modelling of the Rankine cycles was different depending on what kind type of process is used. In the ORC processes there is also an option for internal heat transfer. Because the power output is the main target to be maximised, the optimisation steps are starting, if a look to the T,s-diagram is taken, from the right side – the expansion – to the left side of the diagram both ways round. Another limiting factor is the moisture content, which should be less than 10% for the SRC cycles and about 0% at the ORC cycles. This is needed because the existence of droplets in the turbine is not appreciated. Smaller turbines used in ORC cycles don't allow any droplets, bigger turbines used in SRC a maximum of 10% is acceptable. An increase of the live stream pressure cycle would increase the power output but also decrease the thermal efficiency and increase the moisture content as well. Because of that knowledge and test calculations with different moisture contents and live steam pressures the optimum was found at moisture content of 10% or for the ORC cycles at 0%. Higher moisture content up to 15-20% is technically feasible but for safety reasons not used in this work. It can be easily changed in the optimising code of each SRC cycle like shown in Figure 3.2

```
        while True:
            x_phigh = 0.9
            p_high_start, h_high_HD = add_function.p_optimize(p_max,p_min,p_low_start,
```

FIGURE 3.2 – MAXIMUM PRESSURE FOR OPTIMISATION

The option 'x_phigh' gives the moisture content, means 0.9 is 10% moisture content and the subroutine 'p_optimize' from the file 'add_function' gives back the maximum pressure found for the given moisture content.

For the following calculations the minimum temperature differences, the ΔT_{min}, were used to calculate the pressure level of the cooling system and superheater. The given Pinch Points from the input were taken as start values and after a first calculation optimized (This is explained separately for each calculation). The Pinch Point itself is the minimum temperature difference between heating and cooling curve of the heat exchanger networks. When the Pinch Point temperature is used as the starting value for ΔT_{min} check check-calculations are done to ensure that the Pinch Point is maintained.

4.3.1 CALCULATIONS AT FLUEGAS:

Because this tool sets up on the work of Martin Knoglinger (Knoglinger, 2011) his calculations and especially the file 'Flue_gas' , used for the thermal heat capacity and specific entropy, is used and no further explanations than the equation for transferred heat will be explained.

$$\dot{Q}_{fluegas} = \dot{Q}_{transmfered} = \dot{m}_{fluegas} * (cp_{fluegas,in} * T_{fluegas,in} - cp_{fluegas,out} * T_{fluegas,out})$$

EQU. 3.1 – TRANSFERRED HEAT OF FLUEGAS

4.3.2 *OPTIMIZING THE PRESSURE OF THE CONDENSER:*

To save computer resources a short-cut calculation for finding the pressure level of the cooling system was considered. The results were compared to those of the Brent solver.

Following steps were taken to find the minimum useable pressure level fitting to the Pinch Point:

- The maximum cooling water temperature and ΔT_{min} are added,

- At the saturated-vapour line the isobar with the same temperature is found,

Figure 3.3 shows a graphical explanation of the steps taken in the short-cut method:

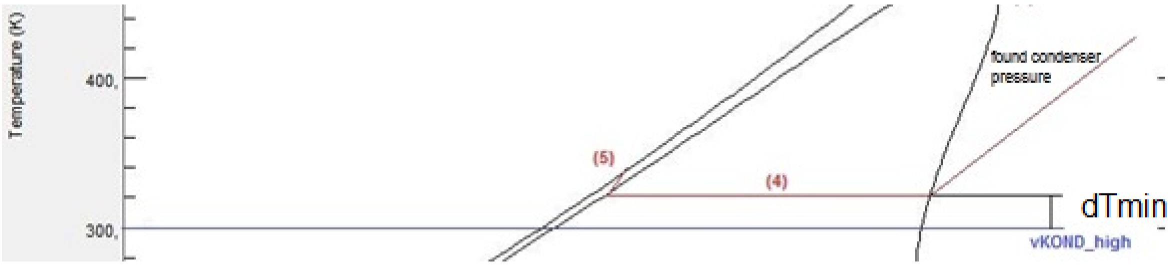


FIGURE 3.3-CONDENSER PRESSURE SHORT-CUT CALCULATIONS

This calculation is a short way and different to those in Martin Knoglinger's subcritical ORC-cycle optimisation (Knoglinger, 2011). The idea was that the minimal differences in the result between Brent and the short-cut are not substantial and so use of less computer resources should save calculation time. In this way the Pinch temperature is some °K higher than the given one. To calculate the temperature at the Pinch point – which will be the temperature at the condenser inlet – Equ. 3.1 and Equ. 3.2 were used. The enthalpy $h_{ORC,cooler}$ is the enthalpy of the state point at the inlet or outlet of the ORC/cooler heat exchanger.

$$Q_{cooler} = \dot{m}_{ORC} * (h_{ORC,cooler,in} - h_{ORC,cooler,out})$$

EQU. 3.2 – HEAT TRANFERED IN COOLER

$$T_{condenser} = \frac{-Q_{cooler}}{m_{cool} * cp_{cool}} + T_{cool,out}$$

EQU. 3.3 – TEMPERATURE AT PINCH POINT/CONDENSER INLET

Equations for rates of mass flow of the cooling cycle will be explained later.

To confirm the applicability of the short-cut the Brent method was implemented into the optimisation and the same calculations were done. The results for the power output of both were compared as shown in table. Because of an average power increase of 2-5% for the Brent method the short-cut calculations were dropped and Brent was implemented in all condenser optimisations. As seen the pressure level for the condenser can be lowered and the need of cooling water minimized. Overall the Brent-method gets closer to the real possible optimisation maximum than the short-cut. For the taken check between the two methods, the following inputs were given and only the cooling inlet and outlet temperature as well as the Pinch Point temperature have been varied.

Inputs difference “cooling short-cut” andBrent-optimisation:

Fluid = 'PENTANE'	T_u = 298.15°K	dT_oil_ORC_end = 40°K
eta_s_t = 0.8	p = 100 kPa atmospheric pressure	flue_gas_pressure = 100kPa
eta_s_p = 0.7	T_flue_high = 700°K	
eta_e_t = 1	T_flue_low = 400 °K	
eta_m_t = 1	m_flue = 100 kg/s	
eta_m_p = 1	dT_flue_oil = 10 °K	

$T_{cooling\ water}$ outlet [°K]	set Pinch Point [°K]	Pinch Point - short cut [°K]	Power Output - short cut [kW/kg fluegas]	Power Output - Brent [kW]	Difference [kW/kg fluegas]	%Diff	mcool - short cut [kg/s pro kg fluegas]	mcool - Brent [kg/s pro kg fluegas]	pcool - short cut [kPa]	pcool - Brent [kPa]
293,15	10	13,2	57,97	59,25	1,28	2,16	4,71	4,68	82	73
298,15	10	14,8	55,97	57,83	1,87	3,23	3,17	3,14	98	83
303,15	10	16,3	54,02	56,56	2,54	4,49	2,40	2,37	116	93
293,15	15	18,2	55,97	57,31	1,34	2,34	4,76	4,73	98	87
298,15	15	19,7	54,02	55,85	1,83	3,28	3,20	3,17	116	99
303,15	15	21,2	52,14	54,53	2,40	4,40	2,43	2,40	136	111
293,15	20	23,1	54,02	55,29	1,27	2,29	4,80	4,77	116	104
298,15	20	24,7	52,14	53,92	1,78	3,31	3,24	3,20	136	117
303,15	20	26,1	50,23	52,58	2,35	4,47	2,45	2,42	159	131

TABLE 3.2 – RESULTS COMPARISM SHORT-CUT/BRENT FOR COOLING SYSTEM

4.3.3 Optimizing a supercritical ORC cycle:

The supercritical ORC is a rather young process so technical knowledge is very rare and a proper short-cut calculation needed to be found. Because REFPROP as a state point database was used, the optimisation steps were visualised in the T,s-diagram.

The main target for the supercritical ORC cycle were the maximisation of the power output, the isobars found by calculations should be at least 10% from the critical point in the supercritical area and there should be no moisture during the expansion. Limiting factors above are the oil temperature, the maximum allowed pressure level given by the fluid properties and the maximum temperature level the certain fluid can be used for. The boundaries at the lower end are the maximum and minimum cooling water temperatures.

4.3.3.1 Calculation oil circuit

The thermal heat transfer oil cycle is located between the flue gas and organic fluid in the system. This configuration operates the flue gas at atmospheric pressure, what brings some advantages in safety and construction. The following paper summarises the differences between using pressurised water and thermal oil (Classen Apparatebau Wiesloch GmbH). The equations from him setting up on those from Drescher (Drescher U. , 2008) where $c_{p,oil}$ (shown in Equ. 3.4) is expressed in kJ/kg and by Baehr (Baehr, 2006) for the specific entropy (shown in Equ. 3.5)used to calculate the exergy destruction in the thermal oil/flue gas heat exchanger.

$$c_{p,oil} = 0.0036 * T + 0.8184$$

Equ. 3.4 - Heat capacity of thermal oil

$$\Delta s = 0.0036 * (T_2 - T_{1)} + 0.8184 * ln(T2/T1)$$

Equ. 3.5 – Specific entropy of thermal oil

To get the mass flow rate in the oil-circuit (Equ. 3.6) the transferred heat from the flue gas and the temperatures $T_{oil,high}$ and $T_{oil,low}$ are needed.

$$\dot{m}_{oil} = \frac{\dot{Q}_{transfered}}{0{,}0018 * \left({T_{oil,high}}^2 - {T_{oil,low}}^2\right) + 0{,}8184 * (T_{oil,high} - T_{oil,low})}$$

Equ. 3.6 – Massflowrate oil

$T_{oil,high}$ and $T_{oil,low}$ are found by subtracting $\Delta T_{min,}$ which is the same like the Pinch Point temperature for the oil/fluegas heat exchanger, from the given $T_{fluw,high}$ or $T_{flue,low}$.

In the calculations the assumption was taken, that the oil and flue gas-temperatures rise and fall parallel in the heat exchanger. To validate this, calculations with EXCEL were made. The results displayed in Table 3.4 and Figure 3.4 show that the temperatures approach each other and then depart again. For a pinch point lower than 20°K the difference gets lower smaller than some degrees but a lower pinch point between the thermal transfer oil and flue gas is not practical, so it was decided that there is no need for a check calculation after optimization.

This calculation was done with different rates of mass flow of the flue gas, different temperatures of flue gas inlet and outlet and pinch point. Table 3.3 shows the given input for the displayed results.

Tfluegas in [°K]	Tfluegas out [°K]	Pinch Point [°K]	massflowrate fluegas [kg/s]
573,15	373,15	40	100

TABLE 3.3–INPUT DATA FLUEGAS/OIL HEAT EXCHANGER PINCH POINT 40°K

dq [kJ/kg]	T_{oil} [°K]	$T_{fluegas}$ [°K]	dT [°K]	%Difference
0	533,150	573,150	40,000	100,00
1000	524,647	563,324	38,684	96,7
2000	516,047	553,491	37,444	93,61
3000	507,347	543,640	36,293	90,73
4000	498,542	533,775	35,233	88,08
5000	489,629	523,896	34,268	85,67
6000	480,603	514,005	33,402	83,50
7000	471,462	504,102	32,640	81,60
8000	462,199	494,187	31,988	79,97
9000	452,809	484,260	31,451	78,63
10000	443,289	474,323	31,034	77,59
11000	433,631	464,376	30,745	76,86
12000	423,830	454,420	30,590	76,47
13000	413,879	444,454	30,575	76,44
14000	403,772	434,480	30,708	76,77
15000	393,500	424,498	30,999	77,50
20000	339,353	374,898	35,545	88,86
20861	329,498	365,902	36,404	91,01

TABLE 3.4–RESULTS FLUEGAS/THERMAL OIL HEAT EXCHANGER PINCH POINT 40°K

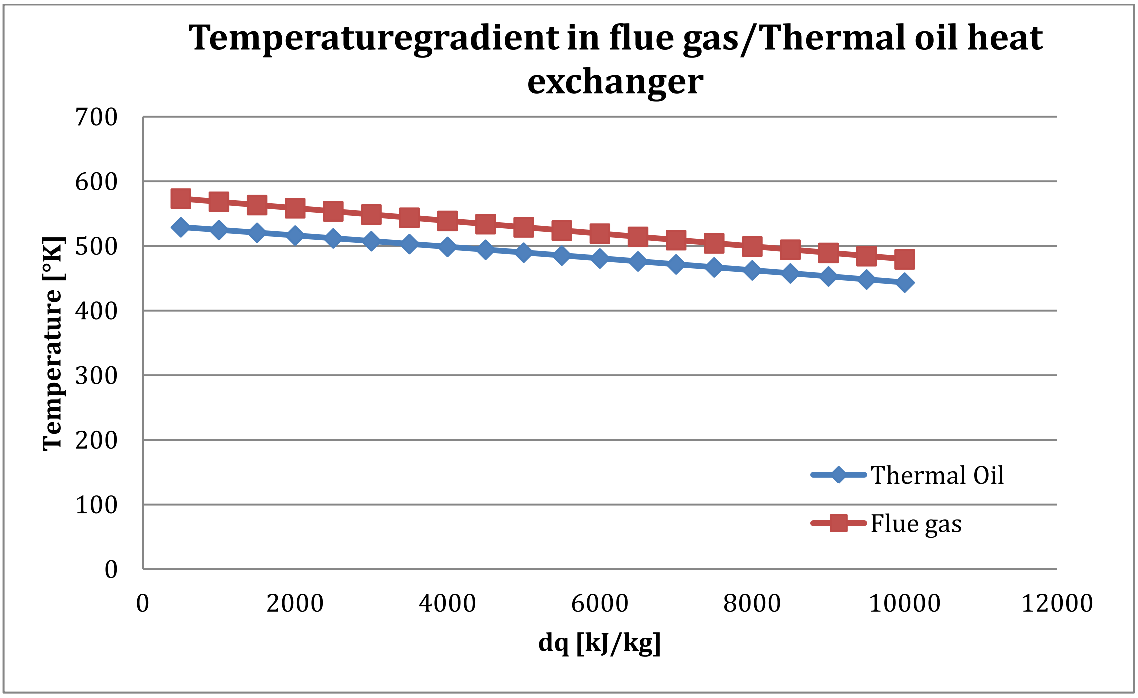


FIGURE 3.4-FLUEGAS/THERMAL OIL HEAT EXCHANGER PINCH POINT 40°K

At the Pinch point of 15°K the difference approaches zero in some parts of the heat exchanger (shown in Table 3.5, Table 3.6 and figure) so there should be no lower Pinch point than this.

Tfluegas in [°K]	Tfluegas out [°K]	Pinch Point [°K]	massflowrate fluegas [kg/s]
573,15	373,15	15	100

TABLE 3.5- INPUT DATA FLUEGAS/OIL HEAT EXCHANGER PINCH POINT 15°K

dq [kJ/kg]	T_{oil} [°K]	$T_{fluegas}$ [°K]	dT [°K]	%Difference
0	558,150	573,150	15,000	100,00
1000	549,921	563,332	11,887	89,41
2000	541,604	553,491	11,887	79,25
3000	533,195	543,640	10,444	69,63
4000	524,693	533,775	9,082	60,54
5000	516,094	523,896	7,803	52,02
6000	507,394	514,005	6,612	44,08
7000	498,589	504,102	5,513	36,75
8000	489,677	494,187	4,510	30,07
9000	480,652	484,260	3,608	24,05
10000	471,511	474,323	2,812	18,75
11000	462,249	464,376	2,128	14,18
12000	452,860	454,420	1,560	10,40
13000	443,340	444,454	1,114	7,43
14000	433,683	434,480	0,797	5,31
15000	423,883	424,498	0,615	4,10
20000	372,485	374,898	2,413	16,09
20861	363,183	365,902	2,718	18,12

TABLE 3.6-RESULTS FLUEGAS/THERMAL OIL HEAT EXCHANGER PINCH POINT 15°K

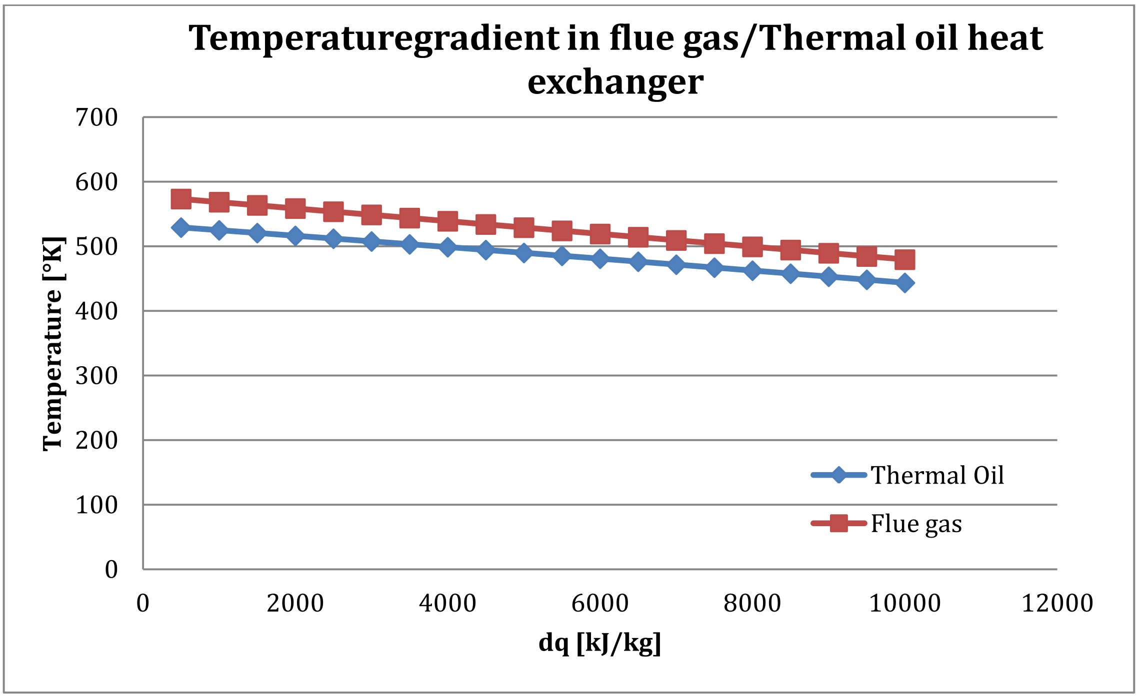


FIGURE 3.5- FLUEGAS/THERMAL OIL HEAT EXCHANGER PINCH POINT 15°K

4.3.2.2 CALCULATIONS BASIC ORC CYCLE:

The following flowsheet (Figure 3.6) gives a short overview to about the simple ORCORC-cycle.

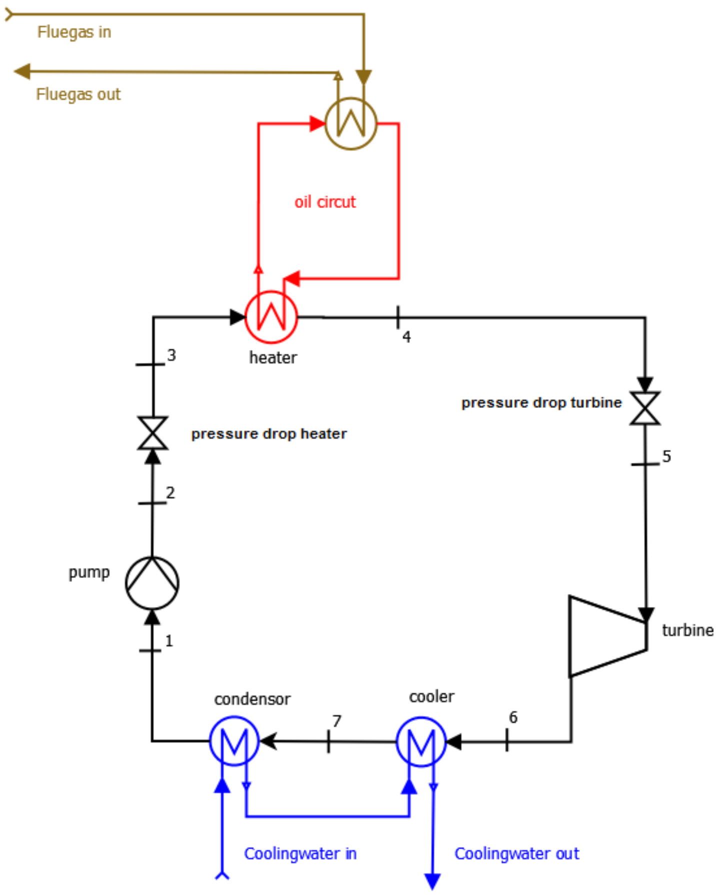


FIGURE 3.6-FLOWSHEET SIMPLE ORC SUPERCRITICAL

Stages and steps in the Basic ORC Cycle:

Needed PRE-caluclations: Optimum pressure P_{OPT} (Stage 4)
Maximum temperature $T_{ORC,high}$ (Stage 4)
Minimum temperature difference in cooling system $dT_{ORC,cool,,min}$ (from Brent)

Stage 1:

Temperature: REFPROP
Pressure: from cooling optimisation
Enthalpy: REFPROP
Entropy: REFPOP

➔ icompression with pump

Stage 2:

Temperature: REFPROP
Pressure: From Stage 4 plus pressure drop in pump
Enthalpy: REFPOP
Entropy: calculated by isentropic compression

➔ pressure drop in heater simulated over valve

Stage 3:

Temperature: REFPROP
Pressure: from Stage 4
Enthalpy: from Stage 2
Entropy: REFPOP

➔ heat transfer in heater to maximum temperature

Stage 4:

Temperature: $T_{ORC,high}$
Pressure: P_{opt}
Enthalpy: REFPROP
Entropy: REFPOP

➔ pressure drop in turbine simulated over valve

Stage 5:

Temperature: REFPROP
Pressure: from Stage 4 minus pressure drop
Enthalpy: from Stage 4
Entropy: REFPOP

➔ expanding to pressure of cooling system

Stage 6:

Temperature: REFPROP
Pressure: from Stage 7
Enthalpy: REFPROP
Entropy: calculated over isentropic expansion

➔ cooling to condenser conditions

Stage 7:

Temperature: REFPROP
Pressure: from $T_{cool,high}$ minus $dT_{cool,ORC,min}$
Enthalpy: REFPROP
Entropy: Refprop

➔ cooling to Stage 1

Inside the program, several calculations were done to get the work of pump and turbine, mass flow rates and different state properties. Those calculations are part of the thermodynamic modelling of a cycle.

The enthalpy after isentropic expansion and compression and the equations for work of the pump and turbine are found in Equ. 3.7 Equ. 3.8, Equ. 3.9 and Equ. 3.10. The enthalpies of the states are known from REFPROP.

$$h_2 = h_1 + \frac{(h_{2s} - h_1)}{\eta_{s,p}}$$

EQU. 3.7 – ENTHAPLY AFTER PUMP

$$w_{pump} = \frac{(h_2 - h_1)}{(\eta_{m,p} * \eta_{e,p})}$$

EQU. 3.8 – WORK OF PUMP

$$h_6 = h_5 - (h_5 - h_{6,s}) * \eta_{s,p}$$

EQU. 3.9 – ENTHALPY AFTER TURBINE

$$w_{turbine} = (h_6 - h_5) * \eta_{m,t} * \eta_{e,t}$$

EQU. 3.10 – SPECIFIC WORK OF TURBINE

For the net work and the thermal efficiency of the cycle, Equ. 3.11 and Equ. 3.12, are defined. In these equations the negative value of the turbine’s specific work is used because the work output is negative.

$$w_{net} = -w_{turbine} - w_{pump}$$

EQU. 3.11 – NET WORK OF ORC CYCLE

$$\eta_{therm}[\%] = \frac{-w_{turbine} - w_{pump}}{h_4 - h_3} * 100$$

EQU. 3.12 – THERMAL EFFICENCY OF ORC CYCLE

For the power output (Equ. 3.14) of the cycle the mass flow (Equ. 3.13) of the ORC working fluid is needed.

$$\dot{m}_{ORC} = \frac{\dot{Q}_{transfered}}{(h_4 - h_3)}$$

EQU. 3.13 – MASSFLOWRATE OF SIMPLE ORC

$$P_{cycle} = w_{net} * \dot{m}_{ORC}$$

EQU. 3.14 – POWER OUTPUT OF ORC CYCLE

To find the maximum power output within the given boundaries the first step was, to find the oil start- and final- temperatures called T_oil_high and T_oil_low are calculated. Because of the assumptions made before, the temperature levels are the corresponding flue gas temperatures minus the pinches between oil and fluegas as given by the user.

The inclination of the dew curve in the T,s diagram for the chosen organic fluids are all positive, so the optimisation was made by finding the specific entropy for each fluid, where the curve has it's inflexion point at the right side from the critical point. For propane the gradient of the curve is a little bit negative. So the trend of the curve is almost isentropic and the so identified entropy is the entropy at the top condenser temperature plus the minimum temperature difference between cooler and ORC. This data are saved in the subroutine 'pressure ORC' as in the Smax_array (see Table 3.7). The found entropy and the T_ORC_high level (2), calculated from the T_oil_high minus the corresponding ΔT_{min}, are used to find the maximum pressure for the ORC cycle (3). The steps 1 to 3 are taken to ensure that the isentropic expansion remains in the superheated area.

As explained in an earlier chapter the temperature and pressure levels for the condenser are found (4) and with the pressure level found for the ORC cycle calculations are possible. Those steps are shown in Figure 3.7.

Fluid	Smax [kJ/kg*K]
Pentane	1,4005
Isopentane	1,3865
Isobutane	2,3787
Octamethyl-trisiloxane MDM	0,71909
Toluene	1,1998
Cyclohexane	1,325
Cyclopentane	1,3504
Propane	Defined by cooling temperature

TABLE 3.7-ENTROPY MAXIMUM FOR SUPERCRITICAL OPTIMISATION

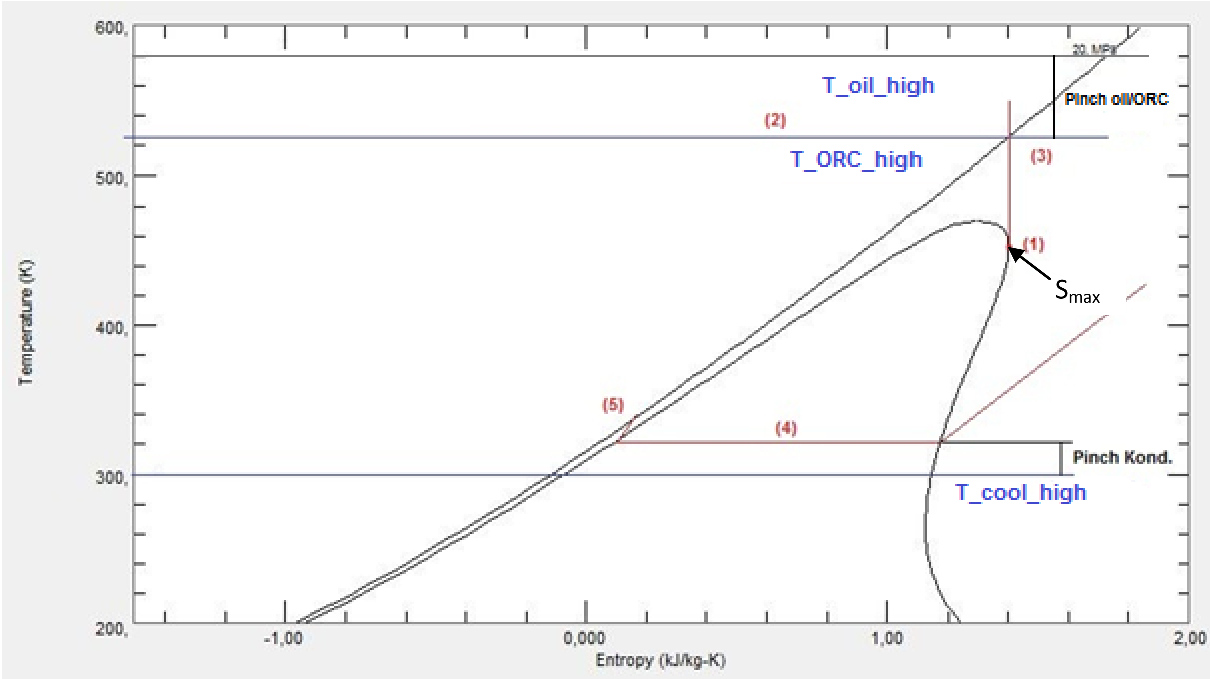


FIGURE 3.7-OPTIMISATION STEPS ORC SUPERCRITICAL

The start value for ΔT_{min} between T_oil_high and the T_ORC_high is given by the Pinch-temperature. To ensure that the minimum temperature difference in the whole ORC/thermal oil heat exchanger doesn't falls below the Pinch, a special auxiliary calculation is done. It is part of the file 'check_tools' and the subroutine called 'ORC_oil_check'. In this subroutine for every 500kJ of transferred heat the temperatures in both cycles – thermal oil and ORC – are calculated. If the difference falls below 90% of the given Pinch point a lower pressure for the ORC cycle is taken and the check calculations are done again. If no suitable pressure in the supercritical area can be found a popup – explained in chapter 5 - appears and the user can decide from between performing subcritical calculations or changing the input. This usually happens if the temperature at the flue gas outlet (T_flue_low) and dependent from, the T_oil_low is too low. The Figure 3.8 displays that this problem appears due of the slope of the isobars and that they get closer in the liquid area so a change in the pressure level does not necessarily effect considerable changes of the temperature in that region. If the T_oil_high and because of that the T_ORC_high is too low, no optimisation in the subcritical area is possible.

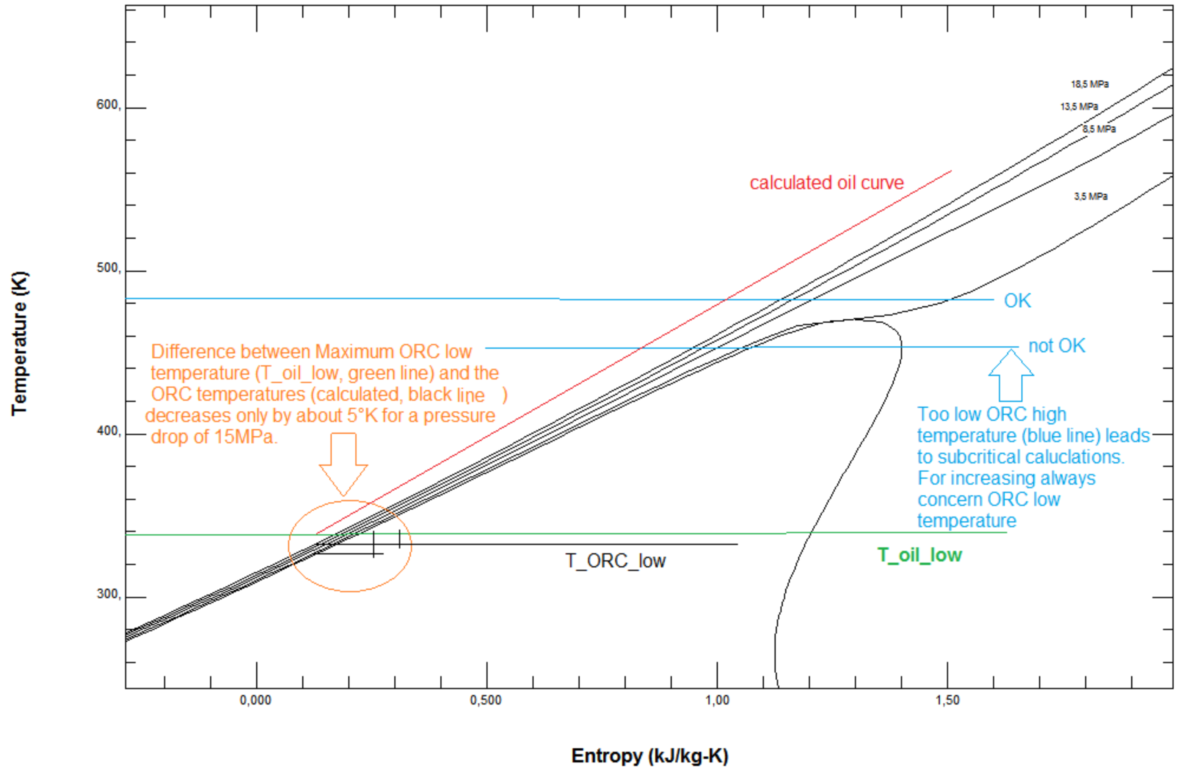
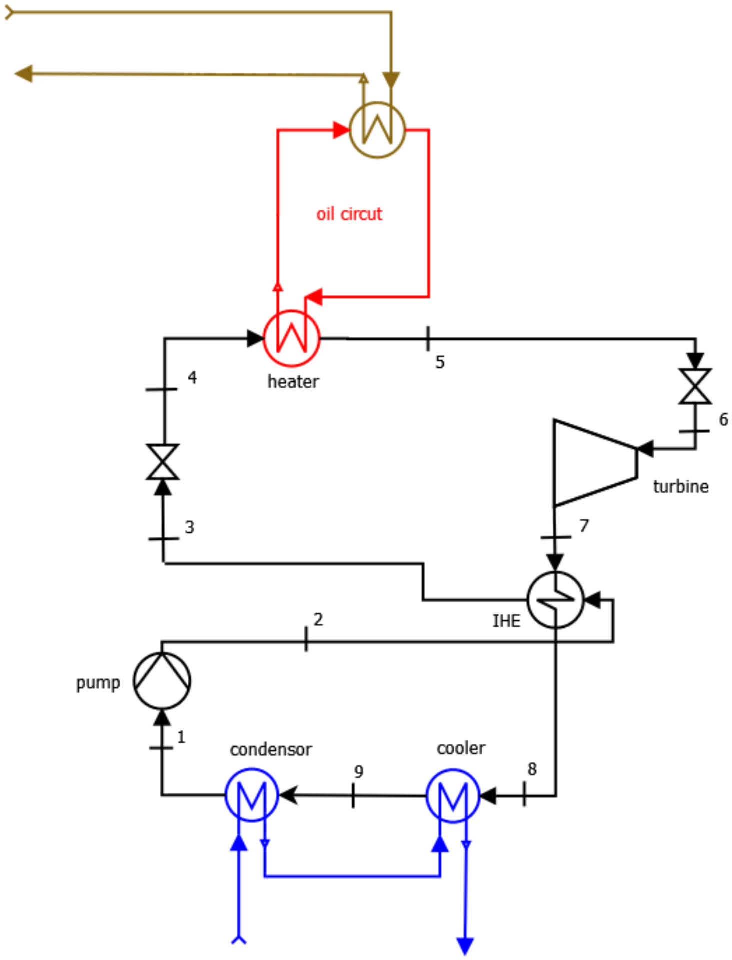


FIGURE 3.8-EXAMPLE: "NO SUPERCRITICAL CALCULATION POSSIBLE (PENTANE)"

4.3.2.3 Calculation ORC cycle with IHE

At fuel based cycles internal heat exchangers are used to reduce fuel consumption. Usually the benefit shows up in using some sensible heat left after the turbine expansion for preheating the working fluid. This leads to a better performance compared to a conventional ORC, although both are operating between the same pressure levels. Figure shows the flow sheet of the ORC with IHE cycle.



Stages and steps in the ORC Cycle with IHE:

Needed PRE-caluclations: Optimal pressure P_{OPT} (Stage 4)
Maximum temperature $T_{ORC,high}$ (Stage 4)
Minimal temperature difference in cooling system $dT_{ORC,cool,,min}$ (from Brent)

Stage 1:

Temperature: REFPROP
Pressure: from cooling optimisation
Enthalpy: REFPROP
Entropy: REFPOP

➔ compression with pump

Stage 2:

Temperature: REFPROP
Pressure: From Stage 4 plus pressure drop in pump
Enthalpy: REFPOP
Entropy: calculated over compression

➔ heat transfer in IHE

Stage 3:

Temperature: REFPROP

Pressure: from Stage 2
Enthalpy: from Stage 2 plus η_{IHE}*(h_8-h_7)
Entropy: REFPOP

➔ pressure drop in heater simulated over valve

Stage 4:

Temperature: REFPROP
Pressure: from Stage 5
Enthalpy: from Stage 3
Entropy: REFPOP

➔ heat transfer in heater to maximum temperature

Stage 5:

Temperature: $T_{ORC,high}$
Pressure: P_{opt}
Enthalpy: REFPROP
Entropy: REFPOP

➔ pressure drop in turbine simulated over valve

Stage 6:

Temperature: REFPROP
Pressure: from Stage 4 minus pressure drop
Enthalpy: from Stage 4
Entropy: REFPOP

➔ expanding to pressure of cooling system

Stage 7:

Temperature: REFPROP
Pressure: from Stage 9
Enthalpy: REFPROP
Entropy: calculated over isentropic expansion

➔ IHE

Stage 8:

Temperature: T_2 + min Approach for IHE
Pressure: from Stage 7
Enthalpy: REFPROP
Entropy: REFPROP

➔ cooling to condensation conditions

Stage 9:

Temperature: REFPROP
Pressure: from $T_{cool,high}$ minus $dT_{cool,ORC,min}$
Enthalpy: REFPROP
Entropy: Refprop

➔ cooling to stage 1

The optimization for this type is almost the same as it is for the simple ORC cycle. The difference is that between state 7 and 8 the internal heat exchanger transfers heat to the working fluid, which is under condition 2. The maximum heat transferred is thereby calculated by Equ. 3.15. An value for recuperation of 0,9 was considered and an approach of 10K between T_2 and T_8 was proper for this optimization (see Equ. 3.16.).

$$Q_{IHE} = \eta_{IHE} * (h_7 - h_8)$$

EQU. 3.15 – TRANSFERRED HEAT IN INTERNAL HEAT EXCHANGER

$$T_8 = T_2 + \text{min\_}approach$$

EQU. 3.16 – APPROACH FOR INTERNAL HEAT EXCHANGER

For the work output the equation is now referenced to states 6 and 7, Equ. 3.17. The equations for the mass flow rates of working fluid (Equ. 3.18) and cooling water (Equ. 3.19) change as well. Further changes in calculations not directly belonging to the cycle, like determination of heat transfer coefficients or exergy destruction won't be explained here.

$$w_{turbine} = (h_7 - h_6) * \eta_{m,t} * \eta_{e,t}$$

EQU. 3.17 – SPECIFIC WORK OF ORC TURBINE ORC WITH INTERNAL HEAT EXCHANGER

$$\dot{m}_{ORC} = \frac{\dot{Q}_{transfered,oil}}{(h_5 - h_4)}$$

EQU. 3.18 - MASS FLOW RATE OF ORC WITH INTERNAL HEAT EXCHANGER

$$\dot{m}_{cool} = \frac{\dot{m}_{ORC} * (h_8 - h_1)}{cp_{cool} * \left(T_{cool,out} - T_{cool,in}\right)}$$

EQU. 3.19 – MASS FLOW RATE OF COOLING LIQUID IN ORC WITH INTERNAL HEAT EXCHANGER

4.3.4 *OPTIMIZING AN STEAM RANKINE CYCLE*

Two types of steam rankine cycles (SRC) have been added to the program, a single pressure and a dual pressure process. Parts of the algorithms for the ORC process, like the cooling system, were used in its optimization also. The big differences in optimizing an SRC compared to an ORC is that water is used as working fluid and therefore has a different shape of the curve in the T,s-diagram. As extra tools subcooling and the possibility to extract heat for district heating or similar were implemented. Main design concepts where found in (Kehlhofer, Hannemann, Stirnimann, & Rukes, 2009).

Because of constructive and process related issues the water in the steam drum can have a certain concentration of solved gases, like oxygen and carbon dioxide, leading to corrosion. To get water with a gas concentration low enough to be used in the process thermal de-aeration can be preformed. The code implementation and theoretical background is explained later in this chapter.

4.3.4.1 DISTRICT HEATING

Because SRC are is often part of energy power plants the implementation of district heating is an eligible feature.

$$\dot{m}_{district\ heating} = \frac{q_{district\ heating}}{h_{10} - (T_{slip} * cp)}$$

EQU. 3.20 – MASS FLOW RATE FOR DISTRICT HEATING

Equ. 3.20 returns the mass flow of the steam needed for district heating. Input needed from the user is the heat quantity, the slip (from district heating system) and offset (to district heating system) temperatures. The steam is extracted at 200 kPa (2 bar) after the first expansion of the turbine. The temperature given as T_{slip} should be the temperature of the water coming from the district heating system plus the minimal approach for the heat exchanger. For this calculations the thermal heat capacity (cp) of water, which is usually used as working liquid, 4,18 kJ/kg*K is used.

The remaining mass flow is expanded to the pressure of the cooling system. The cooling system pressure is found like at the ORC-cycle as explained in chapter 4.3.2

4.3.4.2 DE-AERATION:

Water at higher temperatures and lower pressure has a lower solubility for gases. As in SRC plants heat is already used thermal de-aeration is a proper approach. Figure 3.9 shows a typical feedwater tank with a degasser.

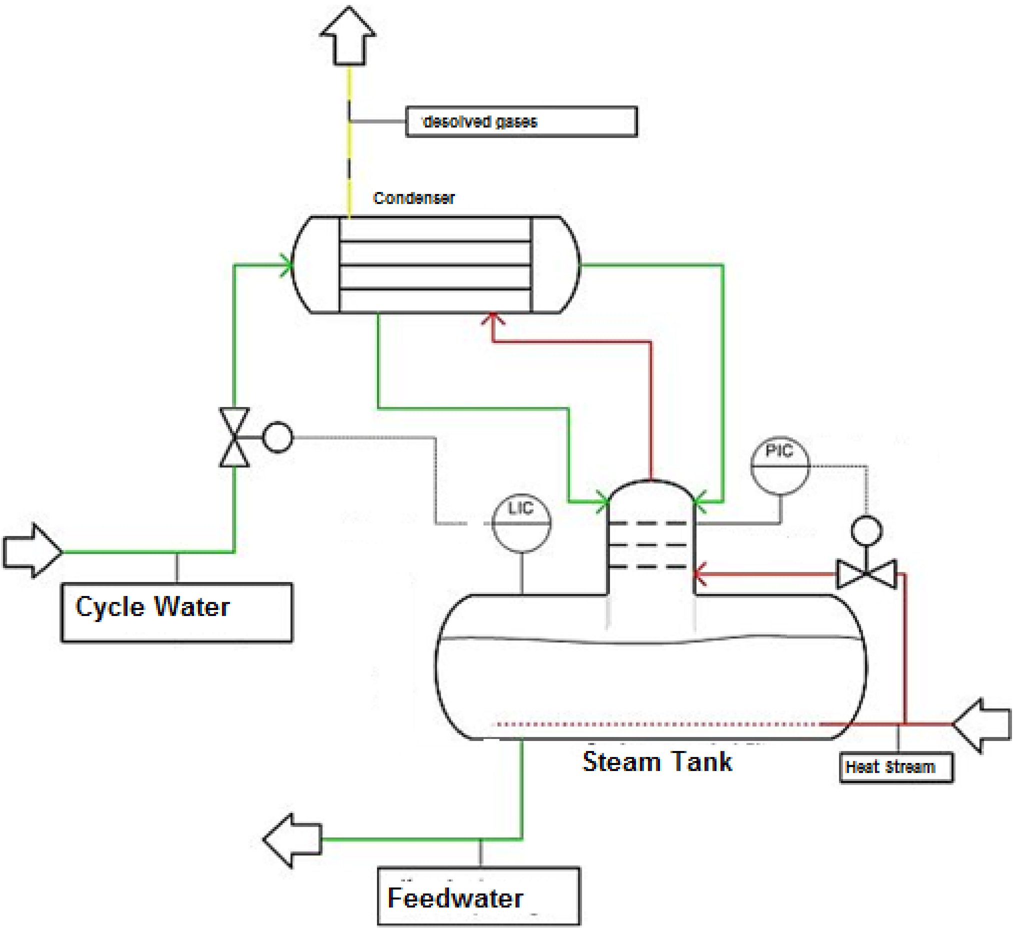


FIGURE 3.9 – DEAERATION

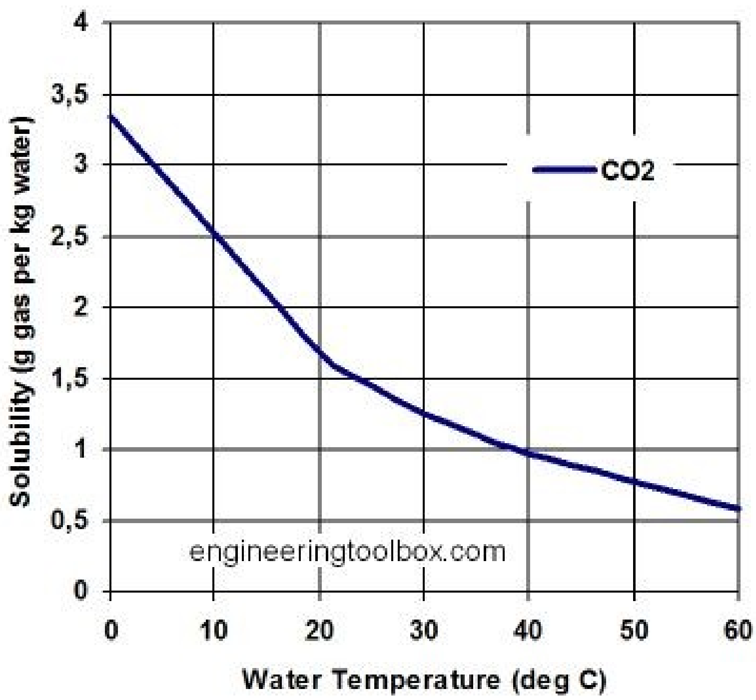


FIGURE 3.10 – SOLUBILITY OF CO2 IN WATER OVER THE TEMPERATURE

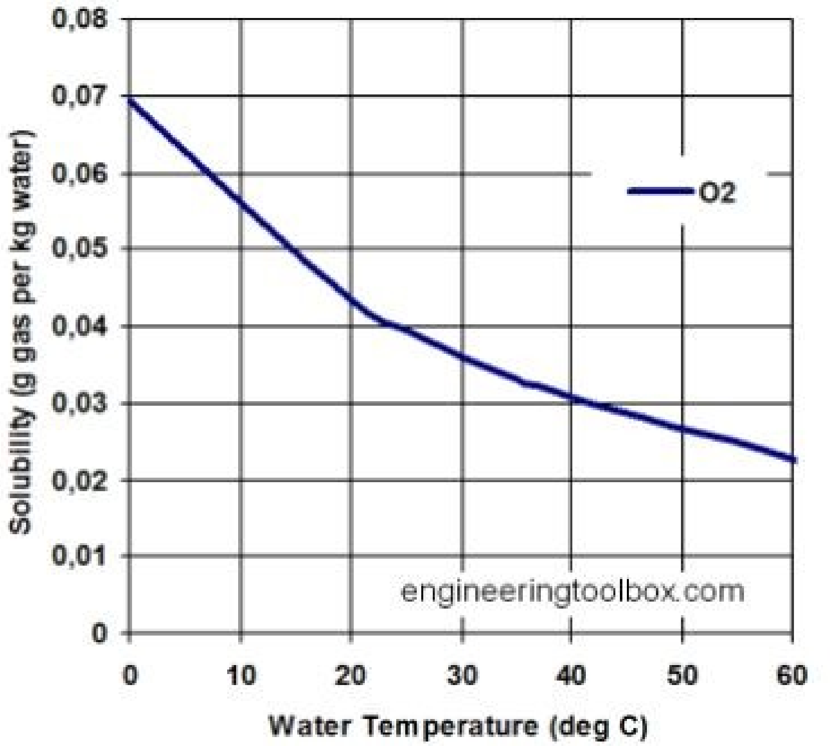


FIGURE 3.11 – SOLUBILITY OF O2 IN WATER OVER THE TEMPERATURE

As shown in Figure 3.10 and Figure 3.11 (The Engineering toolbox) the solubility of the two main gases responsible for corrosion is falling with higher temperatures. For industrial processes temperatures from 102 to 108°C and an enhanced overpressure with of 1,1 to 1,5bar are common. In this work a pressure of 150kPa and a temperature of 378,15°K (103°C) are used. The degassing is performed in the feed-water tank (Stage 2) before the pump with steam from the turbine at 200kPa. The mass flow rate of the steam can be calculated by knowing the enthalpy differences between the state in the feed water tank before degassing and after as well as the enthalpy of the steam extracted from the turbine. All calculated values are returned by REFPROP.

$$\dot{m}_{DEGAS} = \dot{m}_{SRC,top} * \frac{h_2 - h_1}{h_{10} - h_2}$$

Assumptions taken for those these calculations are

- Only the amount of water used for the SRC cycle needs to be degased
- Ideal temperature distribution in the feed tank
- Steam is cooled to same temperature like water after degassing
- No heat loss in the feed water tank
- The different temperatures of the water coming back from the district heating heat exchanger to those in the tank is not taken in account

For easy calculations the amount of steam used for degassing is not reducing the amount of water needed to be degassed.

4.3.4.3 *CALCULATIONS FOR SINGLE PRESSURE CYCLE*

Figure 3.12 shows the flow sheet of the single pressure cycle used in this work. Because of optimising reasons no cooler is needed. Reasons are explained later in this chapter.

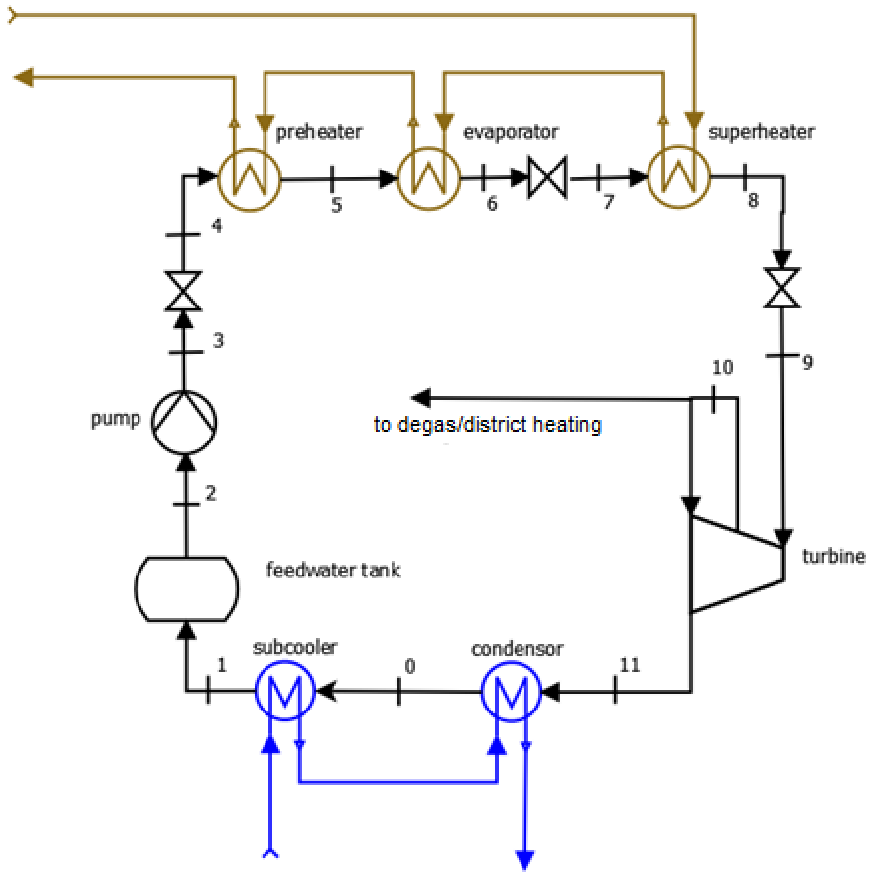


FIGURE 3.12 – FLOWSHEET SINGLE PRESSURE SRC-CYCLE

<u>Stages and steps in the single pressure SRC cycle:</u>

Needed PRE-calculations: Optimum pressure P_{OPT} (Stage 4)
Maximum temperature $T_{ORC,high}$ (Stage 4)
Pressure of cooling system P_{low}

Stage 0:

Temperature: REFPROP
Pressure: from cooling optimisation (P_{low})
Enthalpy: REFPROP
Entropy: REFPOP

➔ subcooling

Stage 1:

Temperature: T_1-$T_{subcool}$
Pressure: from cooling optimisation (P_{low})
Enthalpy: REFPROP
Entropy: REFPOP

➔ Degassing in feedwater tank

Stage 2:

Temperature: 378,15°K
Pressure: 150kPa
Enthalpy: REFPROP
Entropy: REFPOP

➔ compression with pump

Stage 3:

Temperature: REFPROP
Pressure: P_{opt} + pressure drop
Enthalpy: calculated over compression
Entropy: REFPOP

➔ pressure drop in heater simulated over valve

Stage 4:

Temperature: REFPROP
Pressure: P_{opt}
Enthalpy: from Stage 3
Entropy: REFPOP

➔ heat transfer in preheater

Stage 5:

Temperature: REFPROP
Pressure: P_{opt}
Enthalpy: REFPROP
Entropy: REFPOP

➔ heat transfer in evaporator

Stage 6:

Temperature: REFPROP
Pressure: P_{opt}
Enthalpy: REFPROP
Entropy: REFPOP

➔ pressure drop of superheater

Stage 7:

Temperature: REFPROP
Pressure: P_{opt}-$dp_{superheater}$
Enthalpy: from stage 6
Entropy: REFPROP

➔ heat transfer in superheater to maximum temperature

Stage 8:

Temperature: T_{high}

Pressure: from Stage 7
Enthalpy: REFPROP
Entropy: REFPROP

➔ pressure drop of turbine

Stage 9:

Temperature: REFPROP
Pressure: from stage 8 minus $dp_{turbine}$
Enthalpy: from stage 8
Entropy: REFPROP

➔ isentropic expansion to 2000kPa (Degassing, district heating)

Stage 10:

Temperature: REFPROP
Pressure: 2000kPa
Enthalpy: calculated from isentropic expansion
Entropy: REFPROP

➔ isentropic expansion to pressure of cooling system

Stage 11:

Temperature: REFPROP
Pressure: P_{low}
Enthalpy: calculated from isentropic expansion
Entropy: REFPROP

➔ cooling in condenser

The single pressure steam rankine cycle generates steam for the turbine at only one pressure level so the optimization belongs to the pressure and temperature of the so called live-steam.

Boundaries for the chosen cycle:

- Pinch must be achieved at evaporator
- Maximum live steam temperature must be lower than maximum flue gas temperature $T_{flue,top}$ minus pinch and maximum allowed cycle temperature T_{max}
- Maximum moisture content in condenser of 10%
- Maximum pressure given in input (130bar found in literature but more should be possible)

Live steam pressure: A higher live steam pressure causes a higher enthalpy drop and therefore a higher turbine output. On the other side a higher live steam pressure leads to a lower steam mass flow rate of steam due to a higher evaporation temperature. In the used configuration, as shown in table, the effect of a higher live steam pressure is greater than the effect of the lower steam mass flow rate of steam massflowrate. However the efficiency of the cycle decreases with the higher live steam pressure due to a higher stack temperature and therefore worse utilization of energy (shown in Figure 3.13 - (Kehlhofer, Hannemann, Stirnimann, & Rukes, 2009)). On the other hand the lower exergy loss leads to a higher turbine output. Another boundary is the higher moisture content exhausted into the condenser at a high live steam pressure. Because high moisture content leads to problems in the turbine, an maximum moisture content of 10% was used for the given problem. The impact of the live-steam pressure on moisture content, efficiency and turbine output is shown in Figure 3.14 - (Kehlhofer, Hannemann, Stirnimann, & Rukes, 2009).

As found in (Kehlhofer, Hannemann, Stirnimann, & Rukes, 2009) a higher live steam pressure leads to the following advantages:

- smaller exhaust section in the turbine
- smaller volume flow leading to smaller piping and valve dimensions
- reduction of cooling water requirements and so cooling water equipment

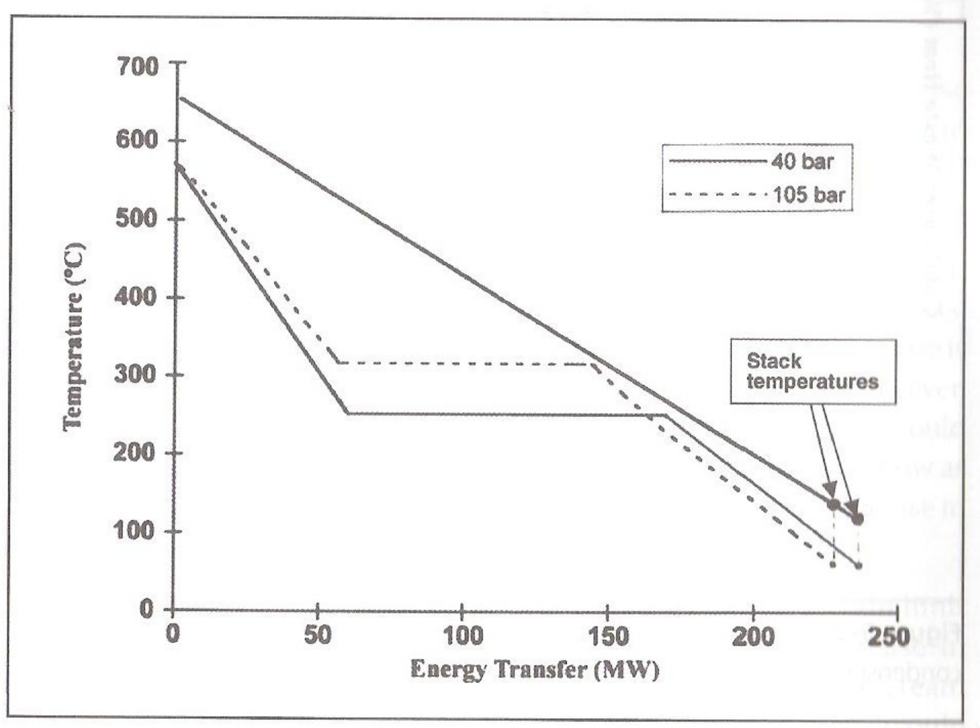


FIGURE 3.13 – EFFICIENCY DEPENDING ON LIVE STEAM PRESSURE

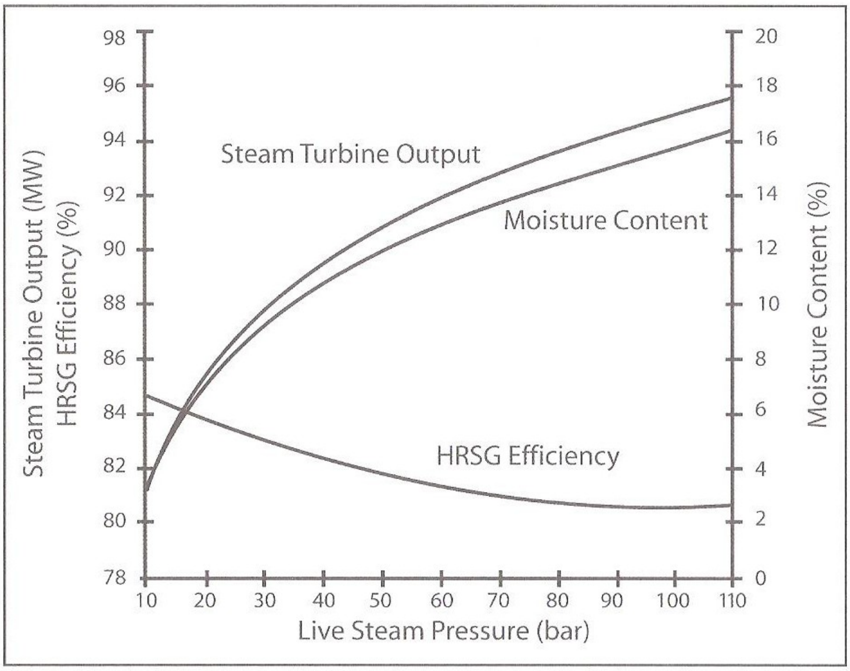


FIGURE 3.14 – IMPACT ON OF LIVE STEAM PRESSURE ON MOISTURE CONTENT, TURBINE PRESSURE, EFFICIENCY

Live steam temperature: For a given live steam pressure raising the maximum (superheating) temperature leads to a slight decrease of turbine output. This effect occurs because higher temperature results in a higher enthalpy drop but less energy can be used for steam production. As the lower temperature limit the moisture content is again the critical effect. A lower temperature leads to an higher moisture content and again here is the same limit, as mentioned before, 10% moisture content in the condenser. Lowering the pressure level makes it possible to lower the live steam temperature and holding the same moisture content but this would result in a lower output. The dependency of the live steam temperature on efficiency, turbine output and moisture content is shown in figure Figure 3.15 (Kehlhofer, Hannemann, Stirnimann, & Rukes, 2009).-

Some other advantages from a lower live steam temperature are:

- lower temperature needs reduces requirements for materials

- better usage of exer

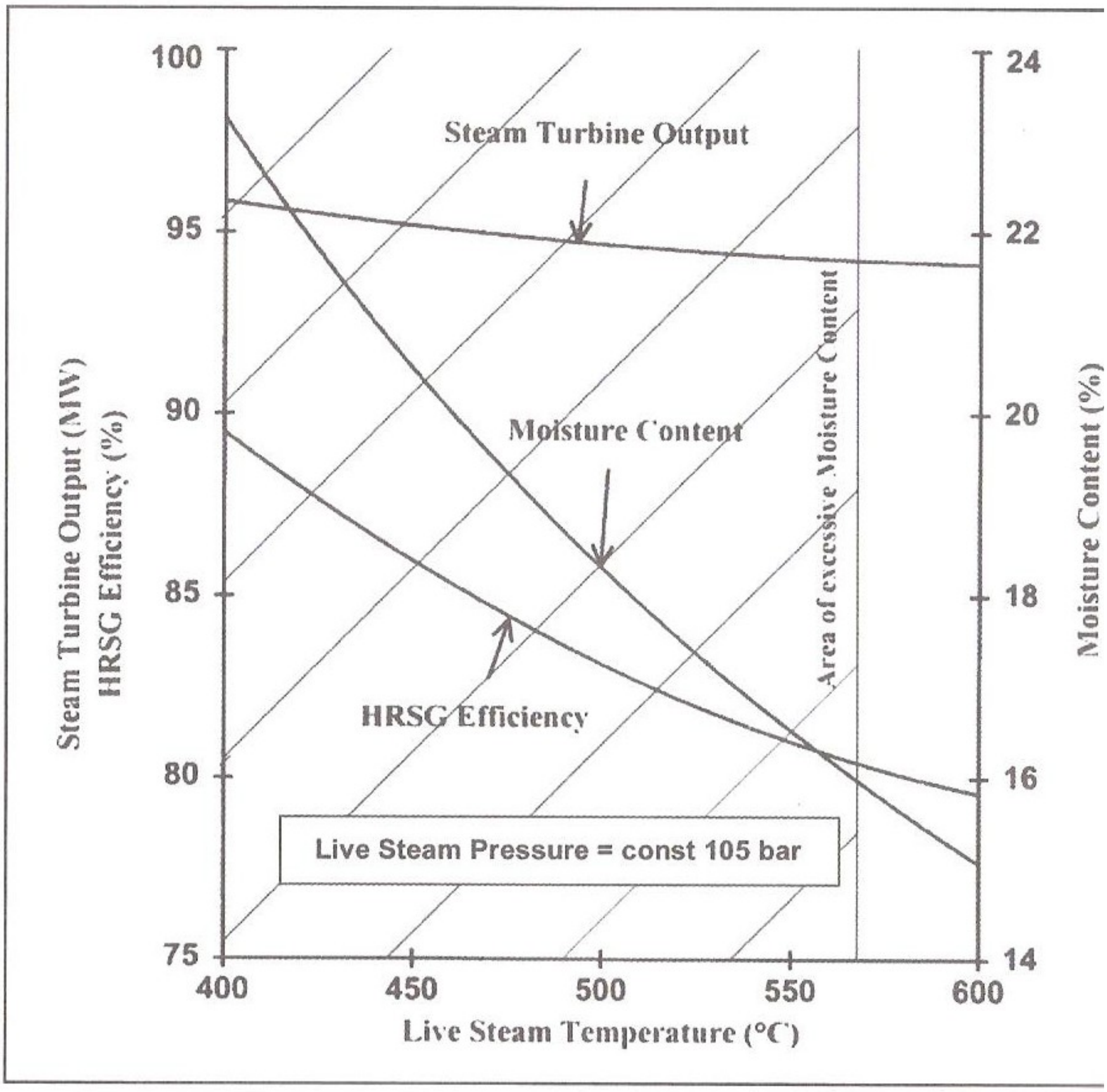


FIGURE 3.15 - DEPENDECNY OF MOISTURE CONTENT/EFFICIENCY AND POWER OUTPUT ON TEMPERATURE

Optimizing procedure:

The assumptions made before show that there is an optimum for the output between maximizing the pressure, minimizing the temperature and keeping a maximum of 10% moisture content. This optimum is found (like shown in Figure 3.16) by finding the maximum pressure level with a BRENT optimization (1) then getting the cooling system pressure (2), as done in the ORC cycles, finding the state point with a moisture content of 10% (3) and then calculating the maximum temperature over “reverse” isentropic expansion from the found determined state point to the maximum pressure level (4).

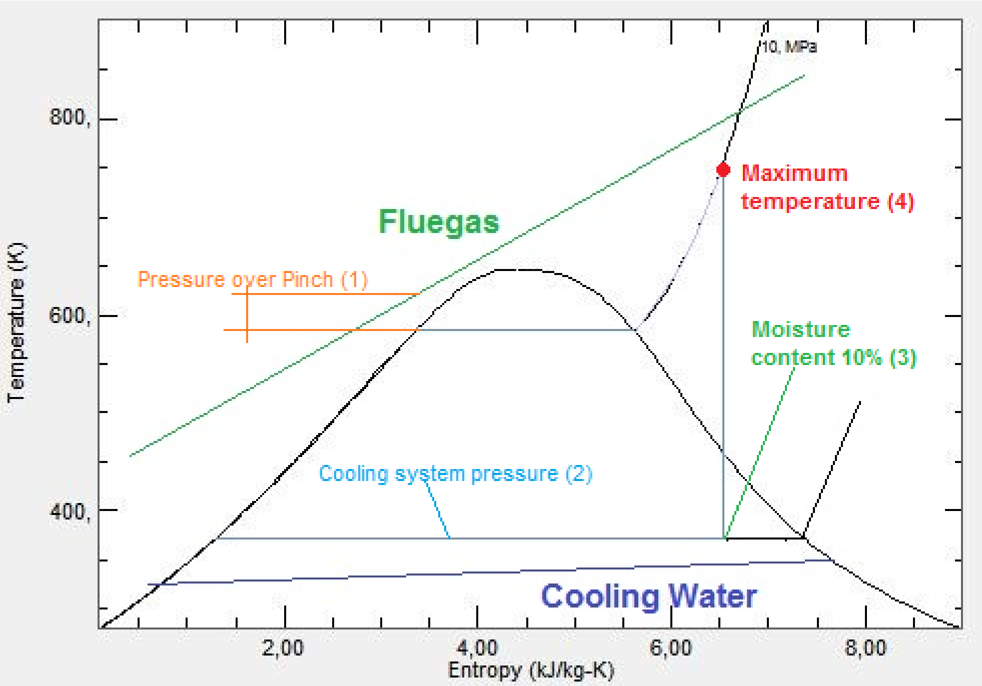


FIGURE 3.16 – OPTIMIZING STEPS SINGLE PRESSURE SRC

In the case if the found temperature is higher than the flue gas temperature minus the pinch point or higher than the maximum allowed temperature – from input – the pressure is minimized until it fits to the boundaries.

4.3.4.4 CALCULATIONS DUAL PRESSURE CYCLE

Figure 3.17 shows the flows sheet of the dual pressure cycle. The used one was seen as a proper approach for the dual pressure cycle as there are many different used in the industry.

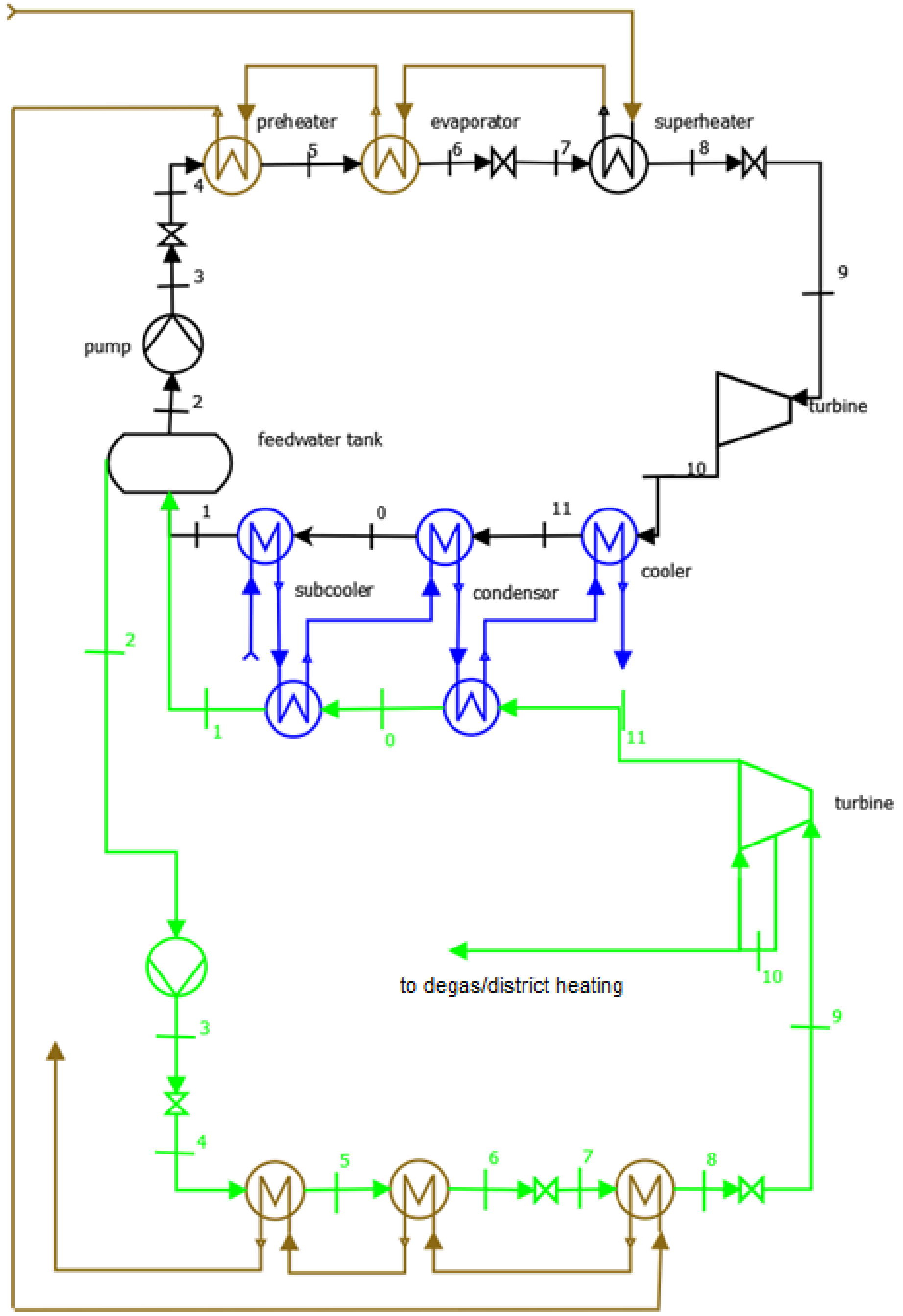


FIGURE 3.17 – FLOWSHEET DUAL PRESSURE CYCLE

<u>Stages and steps in the dual pressure SRC cycle (LP part, green):</u>

Needed PRE-calculations: Optimum pressure P_{OPT} (Stage 4)
Maximum temperature $T_{ORC,high}$ (Stage 4)
Pressure of cooling system P_{low}

Stage 0:

Temperature: REFPROP
Pressure: from cooling optimisation (P_{low})
Enthalpy: REFPROP
Entropy: REFPOP

➔ subcooling

Stage 1:

Temperature: T_1-$T_{subcool}$
Pressure: from cooling optimisation (P_{low})
Enthalpy: REFPROP
Entropy: REFPOP

➔ Degassing in feedwater tank

Stage 2:

Temperature: 378,15°K
Pressure: 150kPa
Enthalpy: REFPROP
Entropy: REFPOP

➔ compression with pump

Stage 3:

Temperature: REFPROP
Pressure: P_{opt} + pressure drop
Enthalpy: calculated over compression
Entropy: REFPOP

➔ pressure drop in heater simulated over valve

Stage 4:

Temperature: REFPROP
Pressure: P_{opt}
Enthalpy: from Stage 3
Entropy: REFPOP

➔ heat transfer in preheater

Stage 5:

Temperature: REFPROP
Pressure: P_{opt}
Enthalpy: REFPROP
Entropy: REFPOP

➔ heat transfer in evaporator

Stage 6:

Temperature: REFPROP
Pressure: P_{opt}
Enthalpy: REFPROP
Entropy: REFPOP

➔ pressure drop of superheater

Stage 7:

Temperature: REFPROP
Pressure: P_{opt}-$dp_{superheater}$
Enthalpy: from stage 6
Entropy: REFPROP

➔ heat transfer in superheater to maximum temperature

Stage 8:

Temperature: T_{high}
Pressure: from Stage 7
Enthalpy: REFPROP
Entropy: REFPROP

➔ pressure drop of turbine

Stage 9:

Temperature: REFPROP
Pressure: from stage 8 minus $dp_{turbine}$
Enthalpy: from stage 8
Entropy: REFPROP

➔ isentropic expansion to 2000kPa (Degassing, district heating)

Stage 10:

Temperature: REFPROP
Pressure: 2000kPa
Enthalpy: calculated from isentropic expansion
Entropy: REFPROP

➔ isentropic expansion to pressure of cooling system

Stage 11:

Temperature: REFPROP
Pressure: P_{low}
Enthalpy: calculated from isentropic expansion
Entropy: REFPROP

➔ cooling in condenser

<u>Stages and steps in the dual pressure SRC cycle (HP part, black):</u>

Needed PRE-calculations: Optimal Optimum pressure P_{OPT} (Stage 4)
Maximum temperature $T_{ORC,high}$ (Stage 4)
Pressure of cooling system P_{low}

Stage 0:

Temperature: REFPROP
Pressure: from cooling optimisation (P_{low})
Enthalpy: REFPROP
Entropy: REFPOP

➔ subcooling

Stage 1:

Temperature: T_1-$T_{subcool}$
Pressure: from cooling optimisation (P_{low})
Enthalpy: REFPROP
Entropy: REFPOP

➔ Degassing in feedwater tank

Stage 2:

Temperature: 378,15°K
Pressure: 150kPa
Enthalpy: REFPROP
Entropy: REFPOP

➔ compression with pump

Stage 3:

Temperature: REFPROP
Pressure: P_{opt} + pressure drop
Enthalpy: calculated over compression
Entropy: REFPOP

➔ pressure drop in heater simulated over valve

Stage 4:

Temperature: REFPROP
Pressure: P_{opt}
Enthalpy: from Stage 3
Entropy: REFPROP

➔ heat transfer in preheater

Stage 5:

Temperature: REFPROP
Pressure: P_{opt}
Enthalpy: REFPROP
Entropy: REFPROP

➔ heat transfer in evaporator

Stage 6:

Temperature: REFPROP
Pressure: P_{opt}
Enthalpy: REFPROP
Entropy: REFPROP

➔ pressure drop of superheater

Stage 7:

Temperature: REFPROP
Pressure: P_{opt}-$dp_{superheater}$
Enthalpy: from stage 6
Entropy: REFPROP

➔ heat transfer in superheater to maximum temperature

Stage 8:

Temperature: T_{high}
Pressure: from Stage 7
Enthalpy: REFPROP
Entropy: REFPROP

➔ pressure drop of turbine

Stage 9:

Temperature: REFPROP
Pressure: from stage 8 minus $dp_{turbine}$
Enthalpy: from stage 8
Entropy: REFPROP

➔ isentropic expansion to pressure of cooling system

Stage 10:

Temperature: REFPROP
Pressure: P_{low}
Enthalpy: calculated from isentropic expansion
Entropy: REFPROP

➔ cooling in desuperheater

Stage 11:

Temperature: REFPROP
Pressure: P_{low}
Enthalpy: REFPROP
Entropy: REFPROP

➔ cooling in condenser

The dual pressure cycle provides steam at two different pressure levels resulting in a better usage of the flue gas heat to attain a higher turbine output.

Boundaries for the chosen cycle:

- Pinch must be achieved at both evaporators
- Maximum cycle temperature (HP-Part) must be lower than $T_{flue,top}$ and T_{max}
- Maximum pressure of HP-cycle can be chosen at input
- Pressure of LP cycle >300kPa and <800kPa
- Maximum moisture content in condenser 10%
- Flue gas Input temperature of LP-cycle is flue gas output temperature of HP-cycle

Live steam pressure: From (Kehlhofer, Hannemann, Stirnimann, & Rukes, 2009) says that for selecting the pressure levels for the HP- and LP-cycles two considerations need to be made. For the HP-cycle the pressure level must be high to attain proper exergetic utilization of exhaust gas heat. The LP-cycle pressure must be relatively low to get a good energetic usage of flue gas heat leading to a higher turbine output. The pressure in the LP-cycle should not be lower than 300 kPa because if the enthalpy drop is getting very small, the mass flow of steam gets very high and therefore the engine becomes expensive. The effect of a lower pressure and therefore higher steam mass flow rate is the same as in the single pressure cycle but has a higher influence in the LP-cycle. Figure 3.18 shows the effect of a rising HP pressure and therefore rising turbine output. Lower LP-pressure is leading to a higher turbine output caused by a higher steam mass flowrate. Assumptions for the moisture content made before are the same.

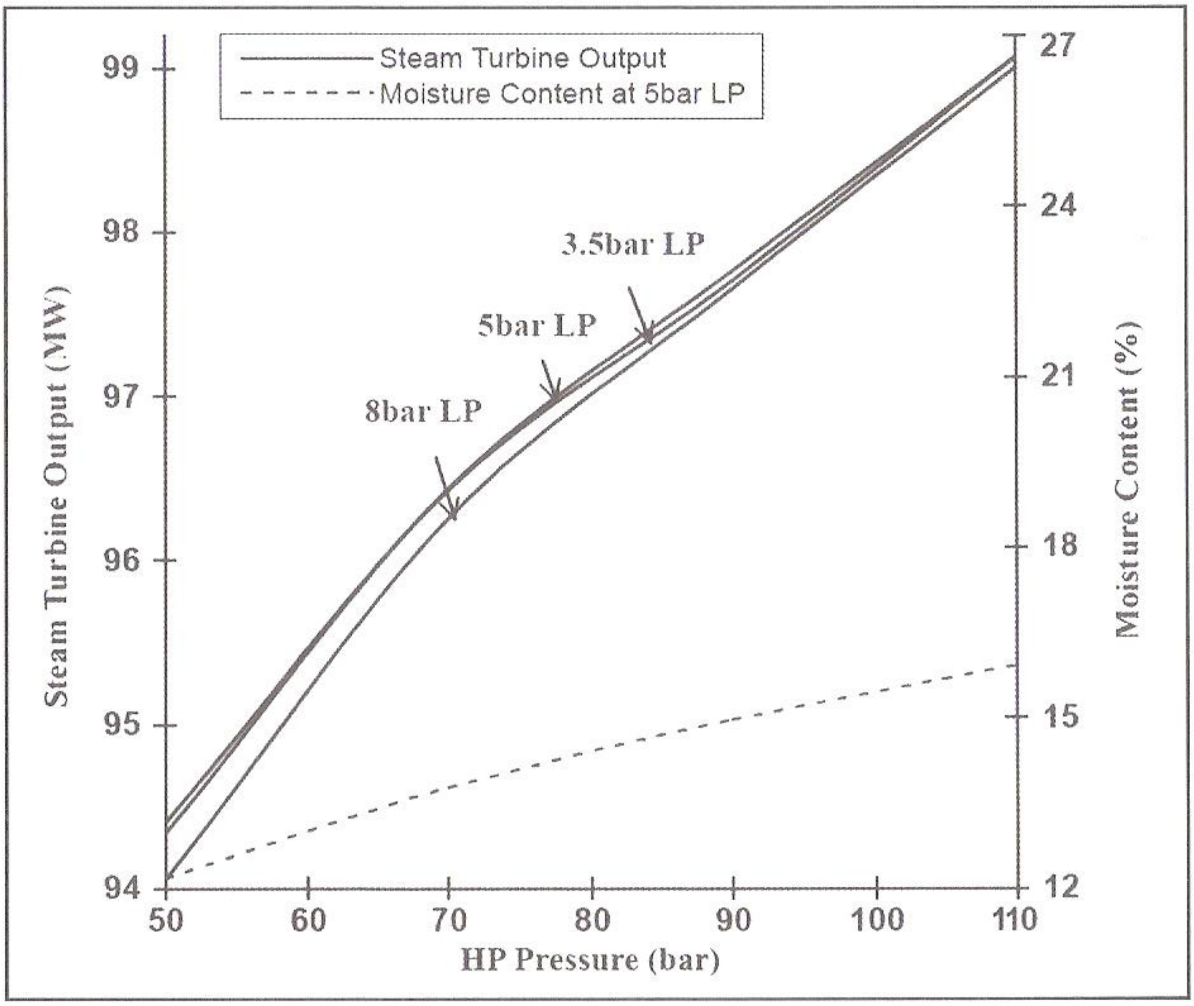


FIGURE 3.18 – MOISTURE CONTENT AND TUBRINE OUTPUT DEPENDING ON HP PRESSURE

Live steam temperature: In the opposite to the single pressure cycle an increase in live steam temperature leads to a substantial improvement in the turbine output. The assumption for the single pressure cycle is here the same for the HP-part, a slight increase of the maximum temperature leads to a lower turbine output but in the case of the dual pressure cycle it makes more energy available for the LP-cycle, an effect which more than compensates the loss. Figure 3.19 - (Kehlhofer, Hannemann, Stirnimann, & Rukes, 2009) shows the effect of a higher HP-and and LP-temperature on steam turbine output for a dual pressure cycle. For the temperature in the LP-cycle the same effects are seen and so the same assumptions for LP-cycle and the single pressure cycle are made. For the simplicity of the cycle a LP-part without a superheater is the optimum but sometimes a superheater is needed because a higher temperature at the same pressure level leads, as already said, to a decrease in moisture content. Because of the higher temperature in the HP-cycle and therefore an expansion to the superheated steam conditions it is possible that a desuperheater is needed.

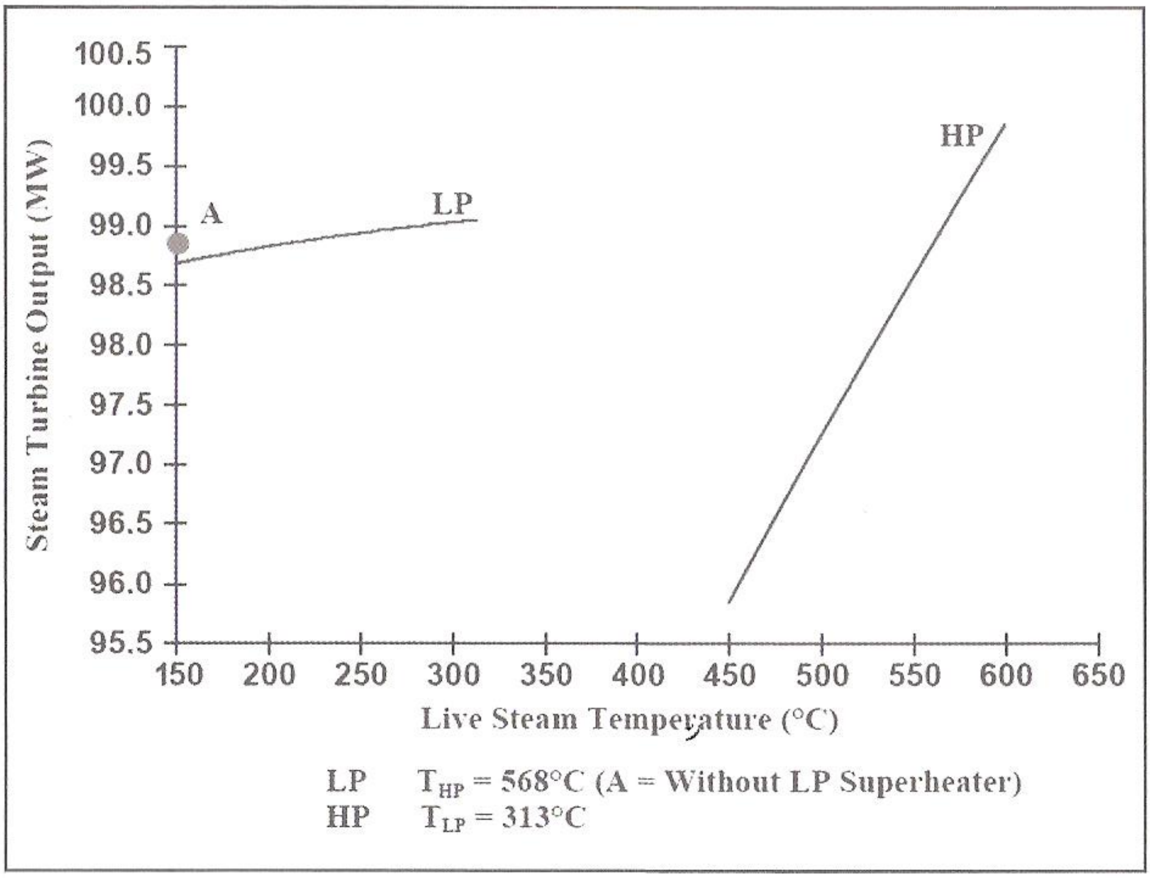


FIGURE 3.19 – TURBINE OUTPUT DEPENDENCY ON PRESSURE LEVEL IN HP- AND LP- SYSTEM

Out of the assumption made before the maximum turbine output for the dual pressure cycle with a maximum moisture content of 10%, the optimization goals are the following.

- Maximum superheating in the HP-cycle
- Maximum pressure in the HP-cycle
- Superheating in the LP-cycle only if needed due to the moisture content of maximum 10%
- LP pressure between 300kPa and 800kPa

Optimizing procedure:

The program first optimizes the HP-cycle and afterwards the LP-cycle. For the HP-part the maximum pressure level is found with the pinch, given in the input, at the evaporator inlet (1). Because maximum superheating is needed the optimum temperature for the cycle is calculated with the maximum flue gas temperature minus the minimum temperature difference, which in this case is the

Pinch (2). For the cooling system pressure the same optimization like the ORC-cycle is done. After this, all state points of the HP-cycle can be calculated. For the LP-cycle the outlet temperature of the HP-cycle needs to be raised, what would lead to a lower mass flow rate of working fluid massflowrate in the HP-part. This effect is used to raise the temperature step by step by lowering the mass flow rate in the HP-part every calculation loop by 0.05kg/s. Now for every step the LP-cycle is optimized and the overall power output calculated. The optimization steps for the LP-part are the same as in the single pressure cycle with first finding maximum pressure by pinch point optimization (5), getting the state point with moisture content of 10% at the cooling system temperature (6) and then finding optimal temperature at top pressure level (7). Because the LP pressure level needs to be between 300 and 800kPa the step is skipped if a lower maximum pressure is found. After calculating all steps, this solution combined with the highest power output is taken to calculate the optimum cycle. Figure 3.20 shows a graphical overview of the necessary steps in the temperature-entropy diagram.

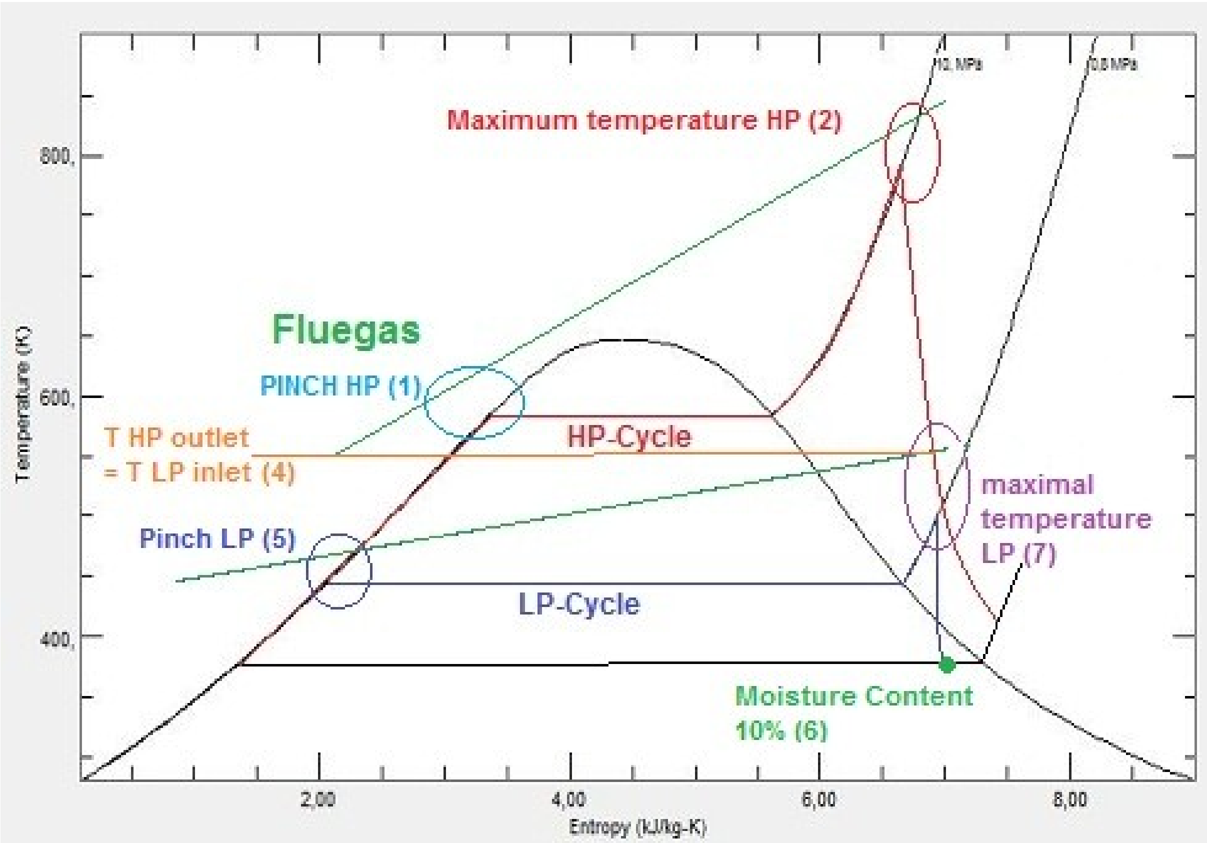


FIGURE 3.20 – OPTIMIZING STEPS DUAL PRESSURE SRC

4.3.4.5 *SPECIAL OPTIMIZATION STEPS IN THE SRC CYCLE*

Because of the nature of the algorithms some inputs can lead to no or wrong results and therefore separate calculation are done in case of an error.

The first can happen because the flue gas output temperature, $T_{flue,low}$ is too low, so no possible pressure level for the given inputs is found. This happens because of the slope of the isobars; the same way as it happens in the ORC cycle. Because in SRC facilities the effect of a higher flue gas outlet temperature is not as crucial as in ORC cycle, due to the higher available temperature gap, an automatic adaption of the outlet temperature is made. It is increased by 1°C as long until there is a possible result is found.

The second case origins from the moisture content. If moisture content of 10% leads to a non satisfying result as shown in Figure 3.21 the moisture content gets decreased step by step till it fits to the calculations. This results in a lower turbine output but makes calculations possible. This problem can happen, if the gap between maximum temperature/pressure and cooling pressure is big and the isentropic coefficient of the turbine leads to a result in the two phase region.

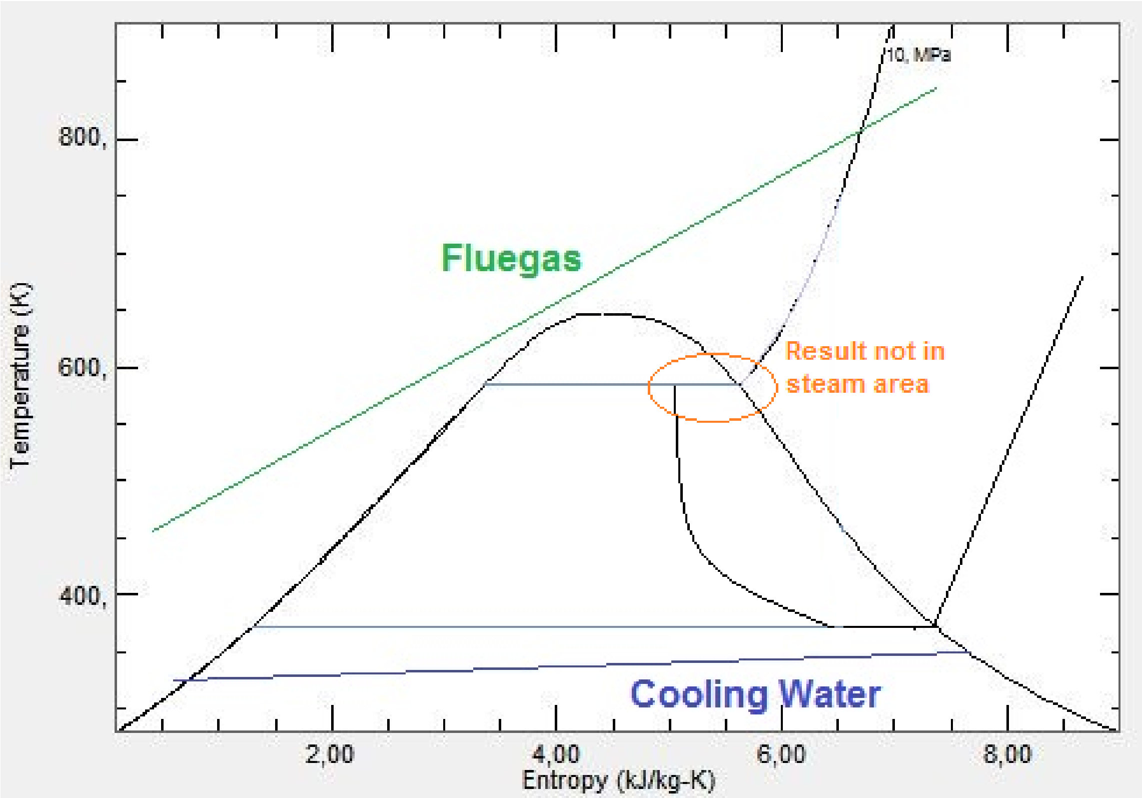


FIGURE 3.21 – BAD RESULT WITH HIGH MOISTURE CONTENT

4. USERS GUIDE – HOW TO USE THE OPTIMIZING TOOL

This part of the essay will give an introduction in how to use the optimising tool and how you can influence the output can be influenced. Furthermore pop-up messages that can appear are explained and solutions for output failures are given.

The design of the GUI – Graphical user interface - is a mixture of giving the user a maximum number of options on defining the cycle but also not taking too much space so it gets confusing. Some further input possibilities are hidden behind buttons that open on a click. If the display resolution or the screen size is too small for displaying the whole window, this can be done by resizing it by hand. For that take the window bottom or side and bring it to a proper size. If the width is too big for the screen make it a smaller so the windows activates auto-scrolling.

5.1 PROGRAMMING A GRAPHICAL USER INTERFACE WITH PYTHON AND QT

For designing the GUI the program, as explained in chapter 4 of this essay, "Qt Designer" was used. After saving the design file a python file needs to be created by starting the tool "makepyqt" and pressing the "Build" button. The tool converts the Design file (for example named MainDlg.ui) to a file readable for python and renames it to ui_MainDlg.py. This file contains all the graphical information for displaying the designed windows but without all data calculated or displayed, as well as no connection to any other widows. If this file would be started directly there would be no button working and no input transferred and results displayed. To connect the design to the calculations an extra file is needed.

This file starts importing the needed files and loading a new class calling the right design files (shown in Figure 4.1). The variable 'self' used in this file is a list of all values written into it and can be transferred inside a program. So values don't need to be transferred each on its own to a different routine in the same file but can be changed and deleted by every of its subroutines.

```
from Designs import ui_resultsSrc1Dlg
from charts import Diagramqt
from charts import TshT
from SRC import reportSRC1Dlg

import sys
import numpy as np

from PyQt4.QtCore import *
from PyQt4.QtGui import *


class resultsSrc1(QDialog, ui_resultsSrc1Dlg.Ui_resultsSrc1Dlg):
```

FIGURE 4.1 – IMPORTS GUI FILE

To connect a push button with a routine, to perform the next steps or to close the window, the code in Figure 4.2 is used. There are two short commands used for Closing a window, ‘self.close’ or ‘self.reject’ whtich terminates the window and all of its data, or ‘self.accept’ to close the window without terminating the data written in the variables.

```
        self.connect(self.ok_pushButton, SIGNAL("clicked()"),self.accept)
        self.connect(self.TshT_Diagram_pushButton, SIGNAL("clicked()"),self.show_TshT)
        self.connect(self.qtdiagram_pushButton, SIGNAL("clicked()"),self.show_diagramqt)
        self.connect(self.showreport_pushButton, SIGNAL("clicked()"),self.show_report)
        self.connect(self.print_pushButton, SIGNAL("clicked()"),self.do_print)
```

FIGURE 4.2 – CONNECTING A PUSH-BUTTON

To perform further calculations without closing the window or just to do side calculations a subroutine can be called by ‘self.subroutine’. In the optimizing tool a subroutine loading the Q,T-diagram would be ‘show_diagramqt’ (shown in Figure 4.3). This routine calls the subprogram ‘Diagramqt’ with its class ‘draw_qt’ and sends all needed variables to it.

```
    def show_diagramqt(self):
        Diagramqt.draw_qt(self.OPT_SRC_STATES[2],self.T_SRC_STATES[4],self.Heatsource[4]
```

FIGURE 4.3 - SUBROUTINE

The subroutine used to print the file is called ‘do_print’. With the code ‘grabWidget’ at first a pixmap (shown in Figure 4.4) – a picture – of the whole displayed page is taken, and is then transferred to the printer. Options for the printer are size and orientation of a page (shown in Figure 4.5). To use ‘grabWindow’ would also work but only if the full window is displayed. When the screen size is too small for displaying the whole window only those parts would be printed that can be seen.

```
        self.document = QPixmap
        self.document = QPixmap.grabWidget(self)
```

FIGURE 4.4- GRAB PIXMAP

```
    def do_print(self):
        self.printer = QPrinter()
        self.printer.setPageSize(QPrinter.Letter)
        self.printer.setOrientation(QPrinter.Landscape)
        dialog = QPrintDialog(self.printer, self)
        if dialog.exec_():
            painter = QPainter()
            painter.begin(self.printer)
            painter.drawPixmap (0, 0, self.document)
            painter.end()
```

FIGURE 4.5 – PRINT A WINDOW

To connect to an input on the design file the code shown in Figure 4.6 is used. Writing the data of an input to a certain variable is done directly or as used in this work in a different subroutine. Those subroutines get called by the ‘connect’ command and are used to change outputs in the design or to increase the value, for example if the input is in °C but calculations are in °K (shown in Figure 4.7). The command 'value()' gets back the value of the input and 'set' changes a option of an output.

```
#flue gas temperatures
self.connect(self.Tfluein_spinBox,SIGNAL("valueChanged(int)"), self.updateData_heatsourcesink)
self.connect(self.Tflueout_spinBox,SIGNAL("valueChanged(int)"), self.updateData_heatsourcesink)
self.connect(self.Tcoolin_spinBox,SIGNAL("valueChanged(int)"), self.updateData_heatsourcesink)
self.connect(self.Tcoolout_spinBox,SIGNAL("valueChanged(int)"), self.updateData_heatsourcesink)
```

FIGURE 4.6 – CONNECT AN INPUT

```
self.massflowrate = self.massflowrate_spinBox.value()
self.T_fluegas_in = self.Tfluein_spinBox.value() + 273.15
self.Tflueout_spinBox.setMaximum(self.Tfluein_spinBox.value()-5)
```

FIGURE 4.7 – SUBROUTINE TO WRITE DATA FROM INPUT TO A VARIABLE

Figure 4.9 shows the code lines for writing data to an output label. Also the design of the text can be changed.

```
self.m_cool_label.setText("m<sub>cool</sub> = "+"%s" " kg/s" % ('{0:.0f}'.format(Heatsink_opt[0])))
self.m_cool_label.setFont(QFont('OldEnglish',12))
```

FIGURE 4.8 – SET TEXT OF AN LABEL

5.1 THE MAIN WINDOW

To start the program you need to double-click on the File START.pyc (shown in Figure 4.9)

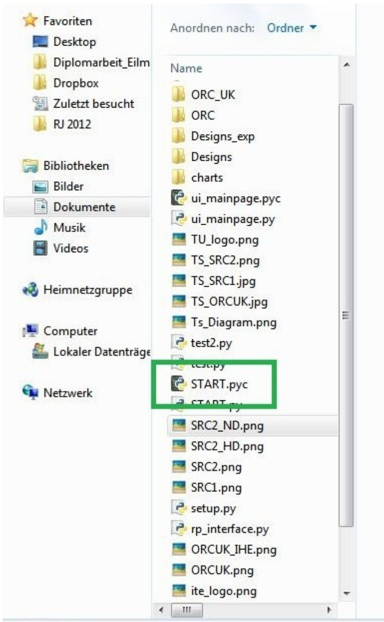


FIGURE 4.9 – START.PYC

The window (shown in Figure 4.10) is the first Window that will appear after double-clicking on the START.pyc file. The tab on top shows the possibilities of the different cycle types that can be optimised. It can be chosen from ORC-subcritical – programmed by Martin Knogglinger- ORC-supercritical, SRC1-stage and SRC 2-stages. By clicking on one of the tabs a short summary of the optimising parameters for the chosen cycle appear as well as buttons to get to certain flowsheets or the optimizing input window. The Button 'More?" brings up some information about the program and where further help can be found.

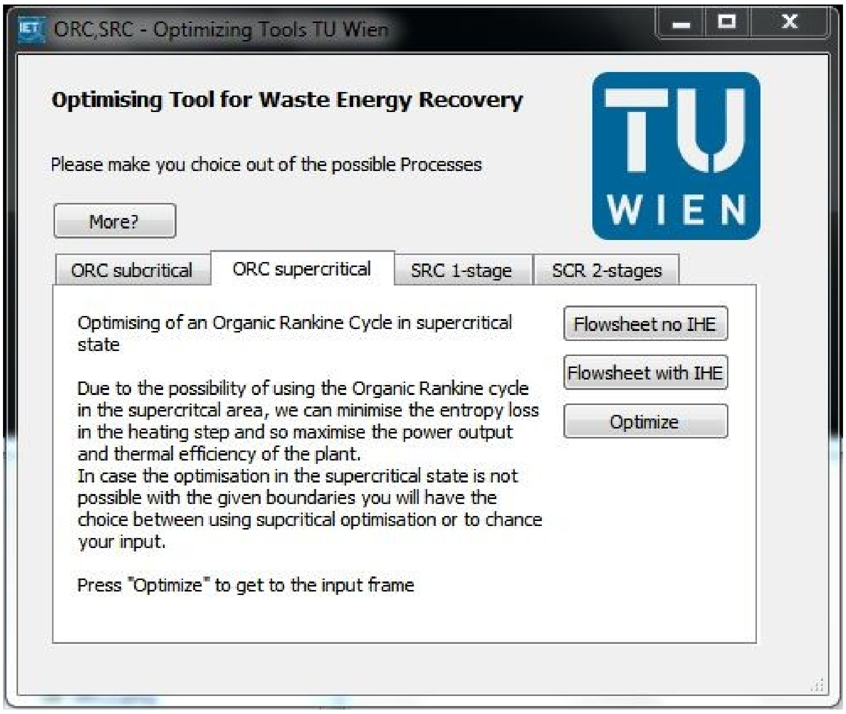


FIGURE 4.10 – MAIN WINDOW

5.2 CYCLE DATA INPUT WINDOWS

The Input Windows provide the optimising calculations with all needed values and give the user enough free space to implement certain special features – like district heating – in his cycle. The input boxes automatically limit all input values. The Input Windows of different cycles look very similar to each other so only special features of each cycle will be explained later on.

On the top left corner of all Input Windows the flue gas composition can be changed. The push buttons invoke the flue gas properties window (shown in Figure 4.11) and when clicking on "OK" the flue gas window closes so the updated integral specific heat capacities as well as the heat input returns in the upper right corner within the main window. This section doesn't differ between the different cycles.

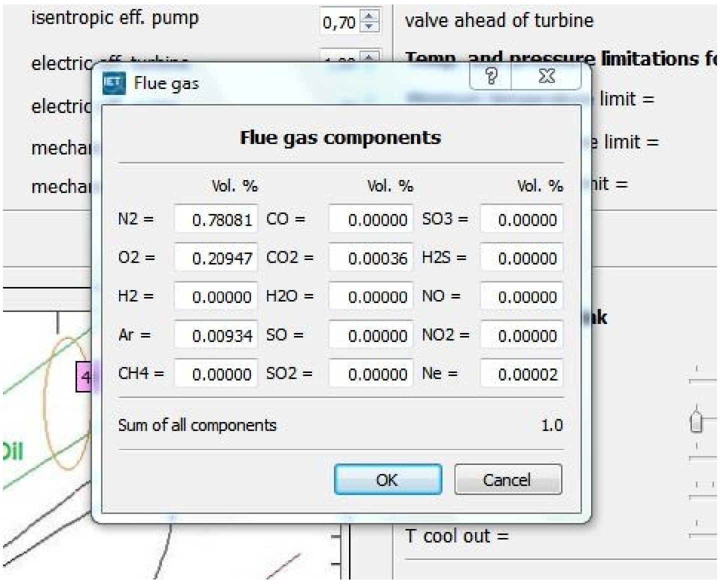


FIGURE 4.11 – FLUE GAS COMPONENTS

In the bottom left corner a temperature-entropy-diagram is shown with a short explanation where the pinch points for the calculations are taken.

By clicking on the "calculate" button in the bottom right corner the optimising steps starts and a result window will appear when the calculations are finished. There can also appear a pop-up window with further options or errors that came back from the calculations. By clicking the "X" on the right top or the "close" button the main Window appears.

5.2.1 *ORC SUPERCRITICAL INPUT WINDOW*

This window (shown in Figure 4.12) requests all the Information needed to optimise a supercritical ORC Cycle. On the left side of the middle section the used Fluid, used type of cycle – with or without internal heat exchanger – and the efficiencies of the turbine and pump are definable. By changing the working fluid the new limits for this fluid appear on the right side above the inputs for pressure drops. On the right bottom corner input for Temperatures and mass flow rate of flue gas and cooling fluid can be found as well as defining of the Pinch points and environment temperature is needed.

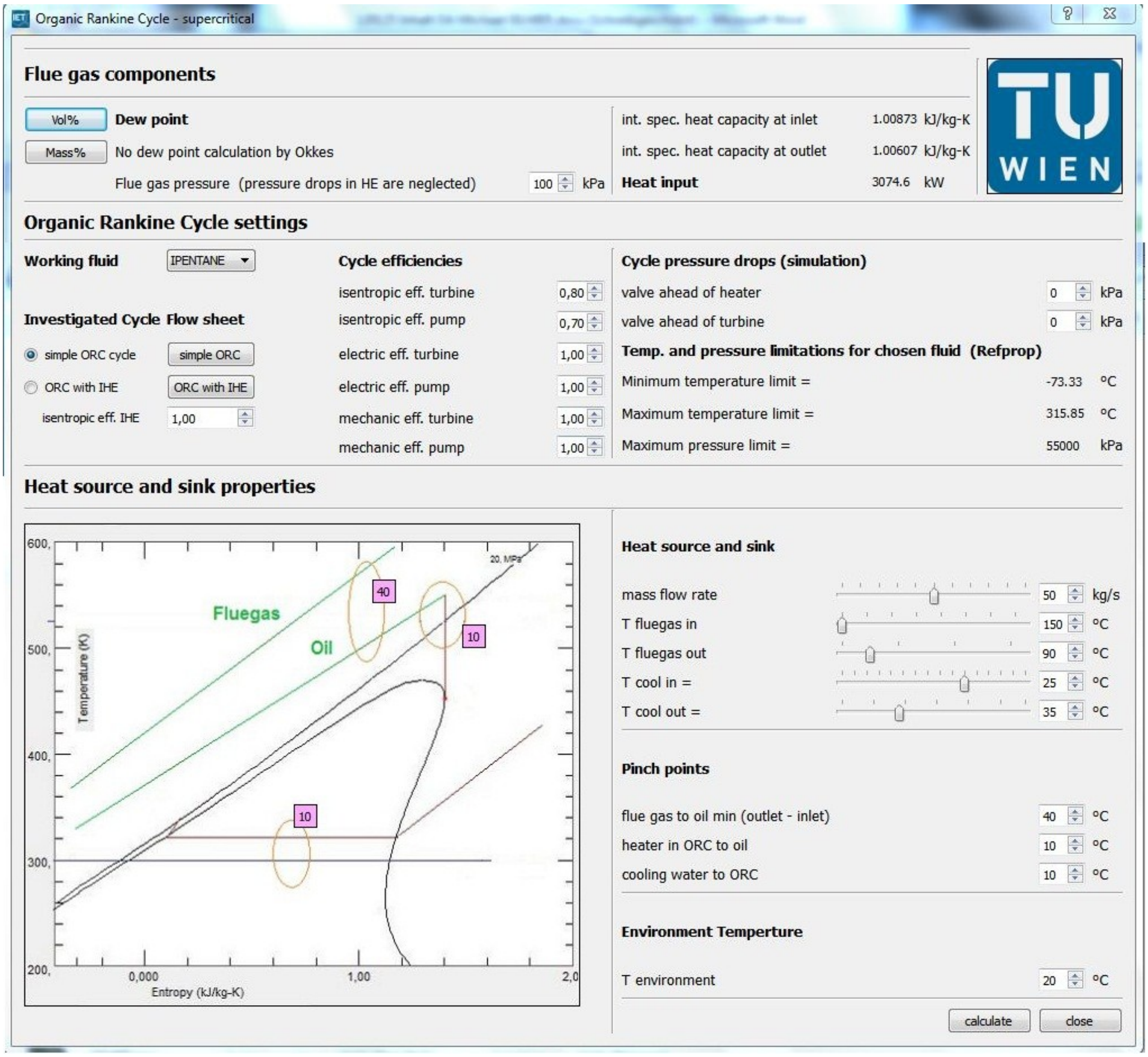


FIGURE 4.12 – ORC SUPERCRITICAL INPUT WINDOW

5.2.2 SRC 1-STAGE INPUT WINDOW

This input Window (shown in Figure 4.13) differs from those for ORC cycle in the middle section where there is no option to choose a certain fluid. In the code of the input window 'WATER' has already been chosen and the limits are shown in the right part of the section. Below the efficiencies all information for district heating is asked. Because of optimizing reasons there is no internal heat exchanger implemented. More information about that can be found in section 3 of this essay. In the right bottom there are inputs for the temperatures of flue gas and cooling liquid, pinch points as well as subcooling and maximum temperature. The maximum temperature is needed if the maximum material working temperature is limiting the possible maximum cycle temperature. If there are no material limits needed put in a higher or at least the same temperature as the "flue gas in" temperature. fluegas in" temperature. The maximum pressure for the SRC cycle is also possible to set in this input window. In the literature a limit of 130bar is often seen but as a result of better materials also higher pressures will be possible and this feature was added to take that in account.

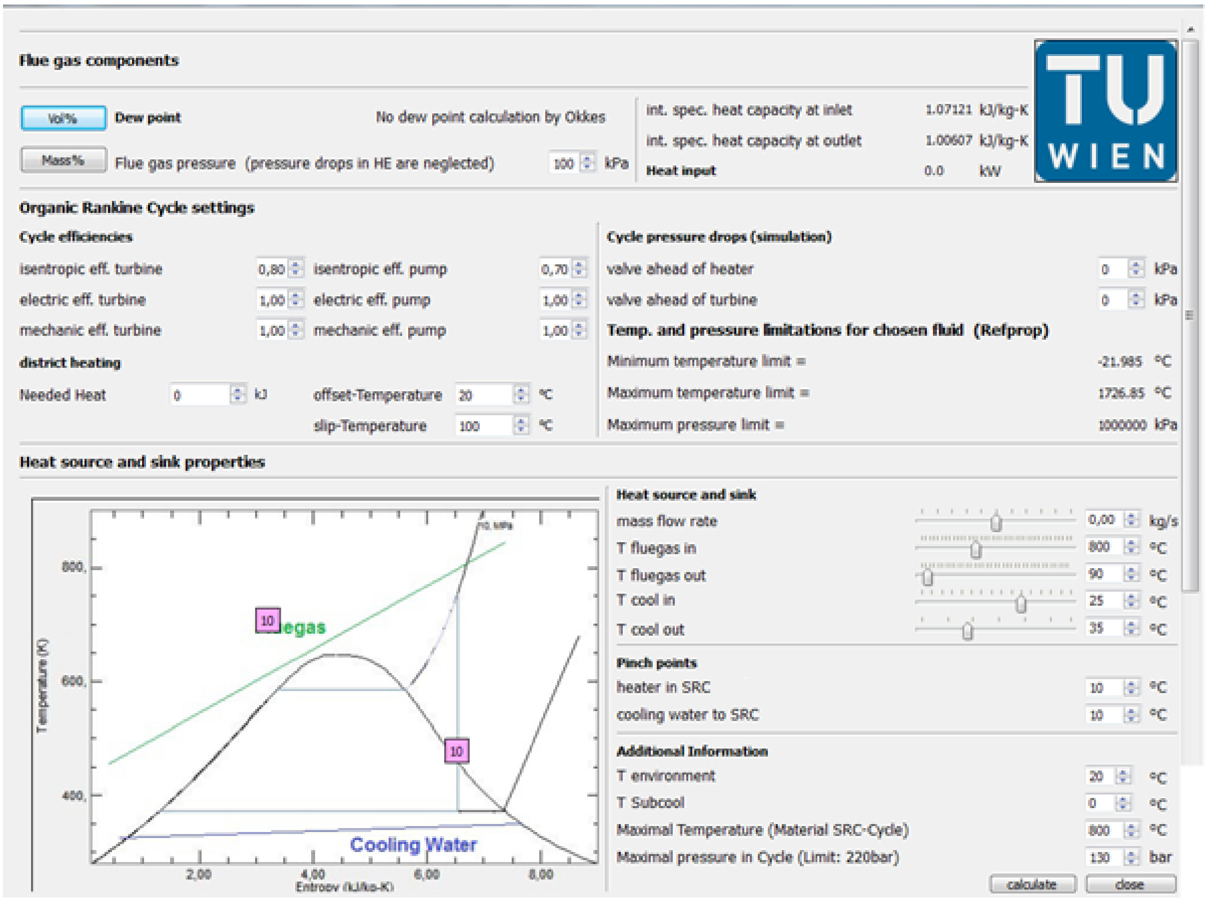


FIGURE 4.13 – SRC 1-STAGE INPUT WINDOW

5.2.2 SRC 2-STAGES INPUT WINDOW

The input window (shown in Figure 4.14) for the 2-stages SRC Cycle gives the same inputs like the SRC 1-stage input window, the only different section is in the middle on the left side. Here the efficiencies for the turbine and pump in both cycles – High pressure (HP) and low pressure (LP) – can be defined.

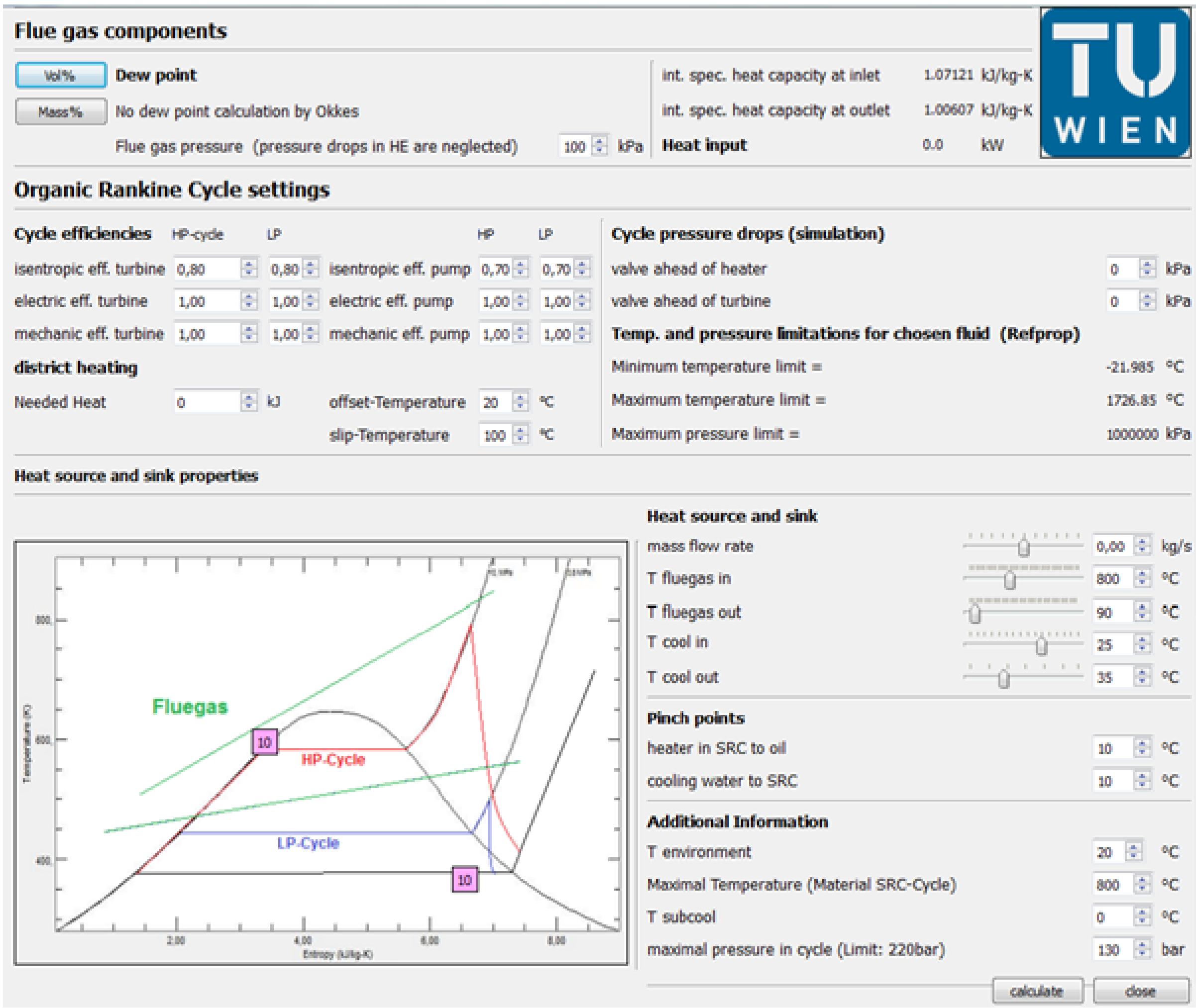


FIGURE 4.14 – SRC 2-STAGES INPUT WINDOW

5.3 Cycle Data Output Window

5.3.1 The Result Window

After the calculation has successfully been finished the "Result"-window (shown in Figure 4.15 – ORC supercritical cycle) will pop up with a short overview of the optimised cycle data. Those Windows in all Cycles look very similar and give the same options for performing the next steps. By clicking on the "print page" button the Print-GUI of your system appears and the whole window is printed to a file or with a printer.

On the left side a flow sheet with the temperatures, pressure drops as well as cycle data like mass flow rate and pressures levels are shown. Furthermore the transferred heat, power output and thermal efficiency are declared. On the right side of the "Result"-window the heat transfer coefficient, transferred heat and exergy destruction is shown for each heat exchanger component individually. On the bottom right side there are push-buttons to invoke temperature-entropy or heat-temperature diagrams as well as a report of all data for printing. By pressing the "ok" button the input window appears again and all calculated cycle data are deleted.

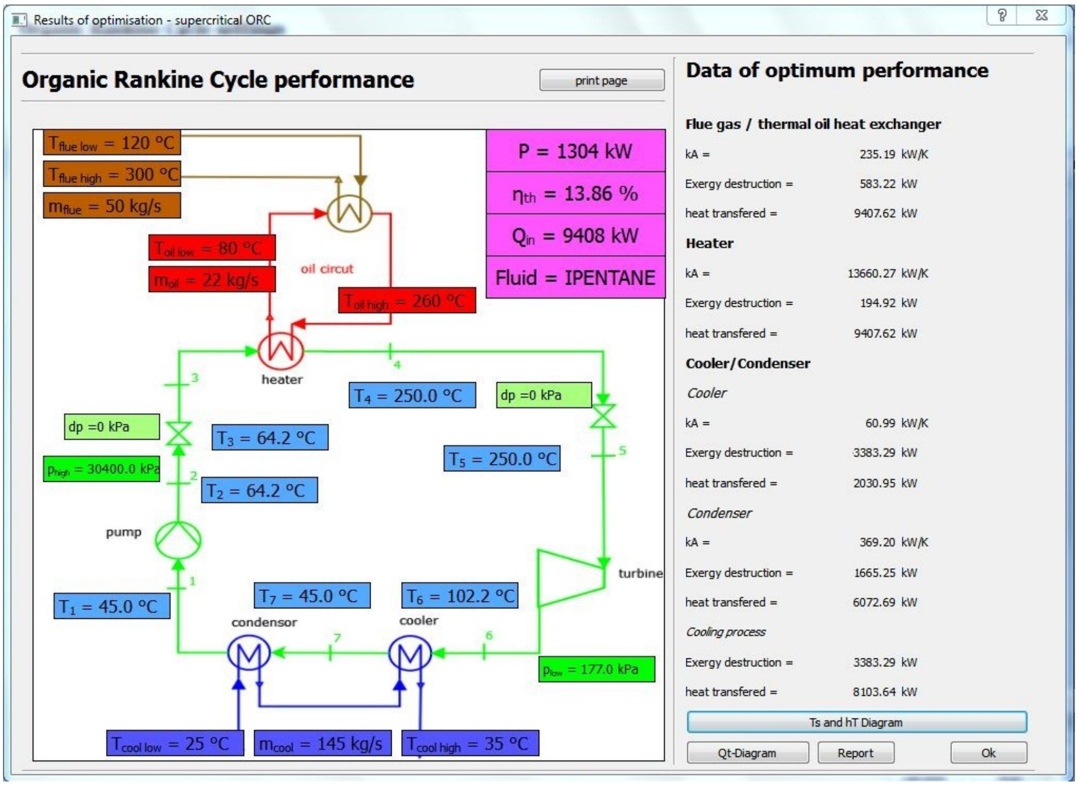


FIGURE 4.15 – RESULT WINDOW

Differences between the cycles belong to the different amount of data each cycle calculation returns. So some have more heat exchangers and others like the 2-stages SRC cycle the two HP- and LP-cycles.

The SRC 2-stages cycle (shown in Figure 4.16) shows at the first page the HP-cycle flow-sheet with temperature and further information as well as the heat transfer coefficient, transferred heat and exergy destruction for the LP- and HP-cycle. By pressing the “show LP-cycle” button on the right bottom of the flow-sheet the LP-cycle flow-sheet (shown in Figure 4.17) appears. The diagrams – T,s and Q,T – are invoked by pressing the appropriate push-buttons. Because of graphical issues the Q,T-Diagrams sare divided in HP and LP cycle.

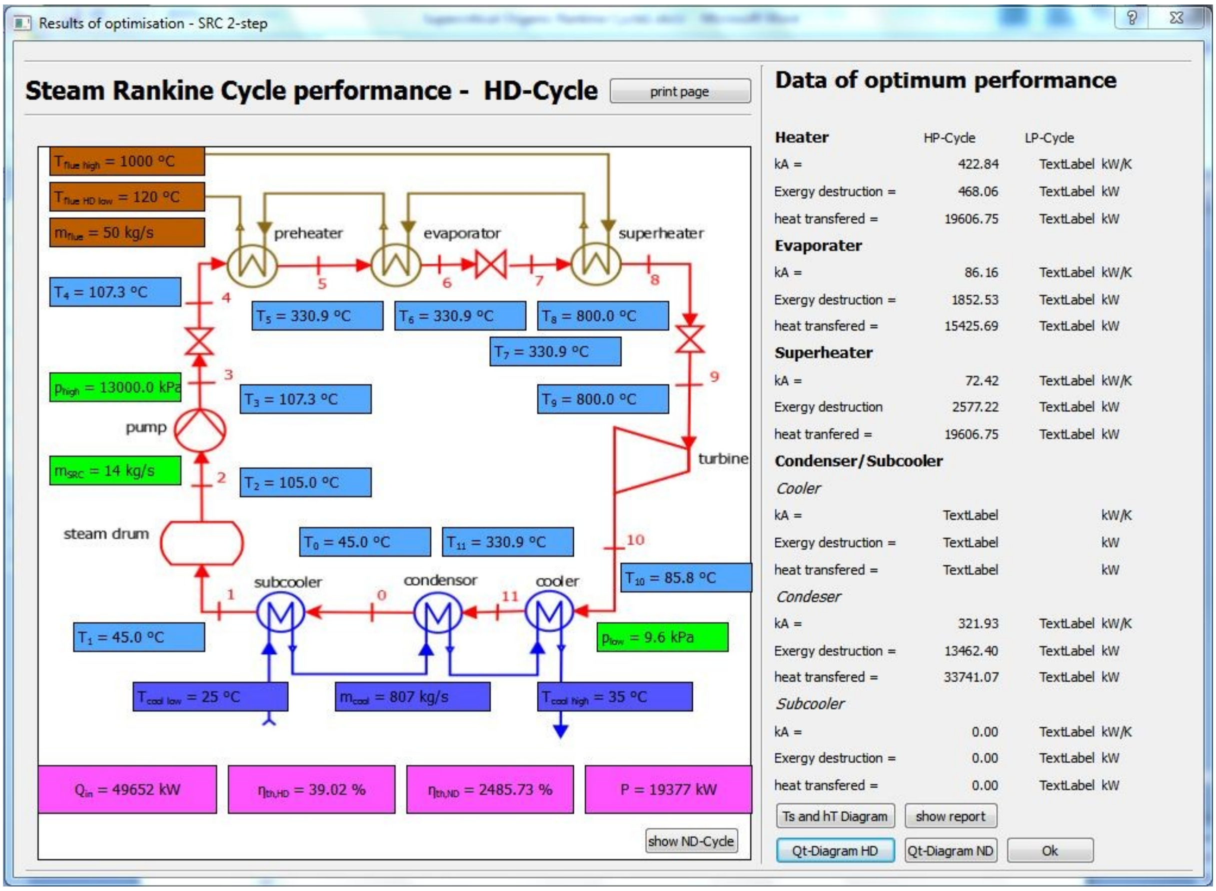


FIGURE 4.16 – SRC 2-STAGES HP-RESULT WINDOW

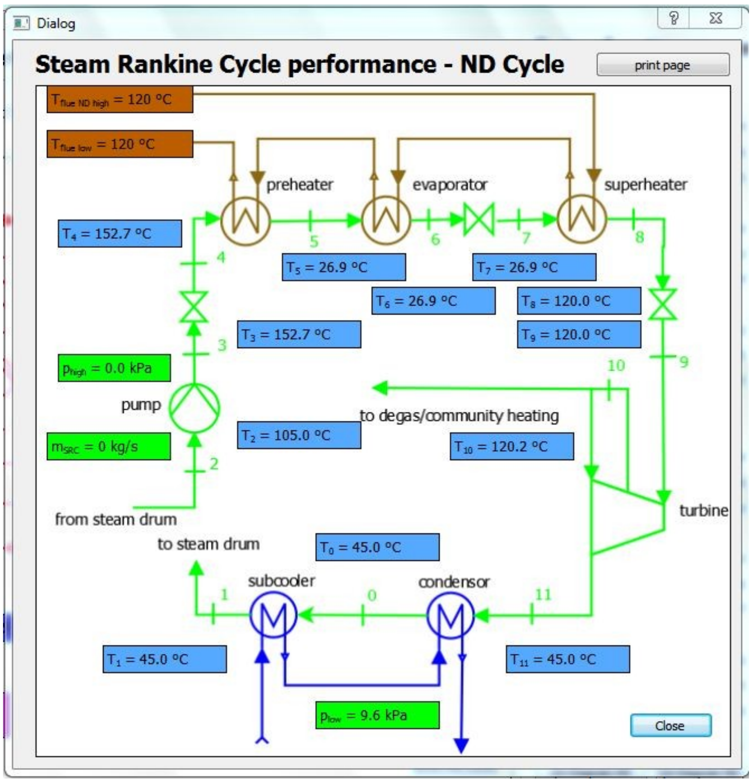


FIGURE 4.17 – SCR 2-STAGES NDLP-RESULT WINDOW

In the SRC 1-stage cycle (shown in Figure 4.18) at the bottom of the flow-sheet the text "Temperature of flue output was adapted for optimizing reasons" can appear. This will happen if the flue-gas outlet temperature is too low for optimizing calculations. Further information you can find in chapter 4.

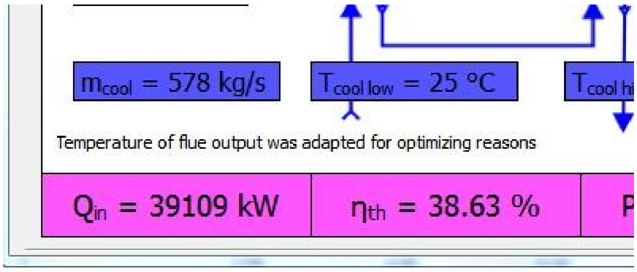


FIGURE 4.18 – TEMPERATURE ADAPTION

5.3.2 *The Report Window*

When clicking on "Report" in the "Result"-Window a new Windows (shown in Figure 4.19) appears with all data, from input or calculated, for the defined cycle. It gives a good overview and with the push-button "print report" all information can be printed to a file or on a printer.

On top of the Report there is all Input data given by the user, in the middle part all calculated data like massflowrates, pressure levels and power output and at the bottom all data of the cycle states can be found. Above the state data the heat transfer coefficient, transferred heat and exergy destruction of each heat exchanger is displayed. Again the HP- and NP-cycle Report for the SRC 2-stages are separately reachable. By opening the Report file from the SRC 2-stages HD Result Window the Report of HP-cycle appears. With the push-button "show NP" the report window of the NP-cycle appears and with the button "show HP" the program switches back to the HP-cycle report window.

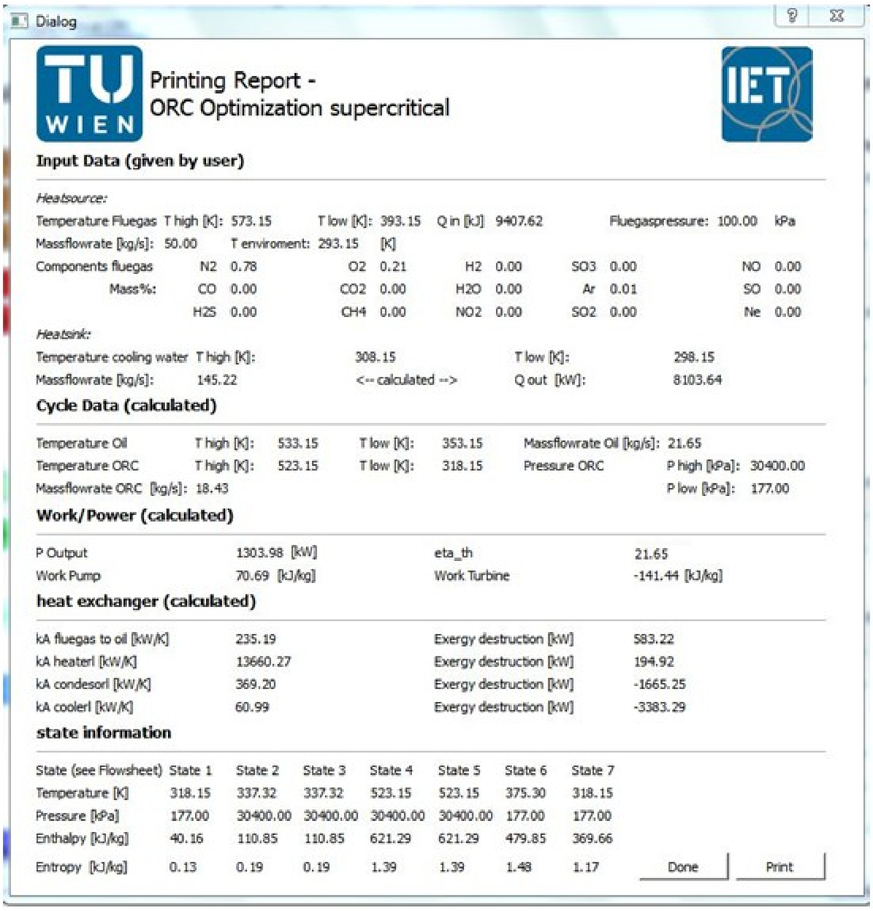


FIGURE 4.19 – REPORT WINDOW

5.4 DIAGRAMS

5.4.1 *PROGRAMMING A DIAGRAM IN PYTHON AND QT*

To print data as a diagram to the output window the module pyplot (PythonPlot) needs to be imported from matplotlib. This module gives easy commands to print a diagram and implement all data. Only a short overview of the code will be shown here.

The plot tools are always started with the 'plt.' code, calling the pyplot module, followed by what it should do. To start a plot the following code lines (shown in Figure 4.20) is needed.

```
plt.subplot(121)
plt.title('Ts-Diagramm')
plt.grid(True)
plt.xlabel("S [kJ]",fontsize = 20)
plt.ylabel("T [K]",fontsize = 20)
```

FIGURE 4.20 – START PLOT

With the code .subplot a new plot on the page is started. If there are two plots on the same page a new subplot is started with the code .subplot(122) for an new plot on the right side (shown in Figure 4.21) as used for the T,h-diagram. The commands '.title',' .xlabel' and '.ylabel' write Labels to the top of the plot as well as to the axes and '.grid(True)' shows a grid in the plot.

```
plt.subplot(122)
plt.title('hT-Diagramm')
plt.grid(True)
plt.xlabel("h  [kJ]",fontsize = 20)
plt.ylabel("T  [K]",fontsize = 20)
```

FIGURE 4.21 – NEW SUBPLOT

To print data in the diagram the code .plot with several inputs is given. To print a line the input shown in Figure 4.22 and for dots the one in Figure 4.23 is written. To make a simple line with no specialities the code 'ro' is changed to 'rl'.

```
    plt.plot(h_SI_iso_cond,T_iso_cond, color = 'green', linewidth=0.5)
```

FIGURE 4.22 – PLOT LINE

```
plt.plot(S_ORC_STATES, T_ORC_STATES, 'ro')
```

FIGURE 4.23 – PLOT RED DOTS

To print text next to the state points of the T,s and T,h-diagram a special code (shown in Figure 4.24) was implemented. The code checks for every state if temperature and specific entropy/enthalpy of two states are the same and if not the state number is printed next to the state point. If temperature and entropy/enthalpy are the same the state numbers are added together divided by a slash and then printed to the state point. The Numeration code to write a text is '.text' and in the

```
    for i in range(len(H_ORC_STATES)):
        if(i == 11): # because then i+1 is over the maximum in array
                plt.text(H_ORC_STATES[i]+0.5, T_ORC_STATES[i], i , fontsize=14)
        else:
            if (round(H_ORC_STATES[i+1],1) == round(H_ORC_STATES[i],1) and round(T_ORC_STATES[i+1],0) == round(T_ORC_STATES[i],0)):
                if (name == ''):
                    name = str(i)
                else:
                    name = name + '/' + str(i)
            elif(round(H_ORC_STATES[i-1],1) == round(H_ORC_STATES[i],1) and round(T_ORC_STATES[i-1],0) == round(T_ORC_STATES[i],0)):
                name = name + '/' + str(i)
                plt.text(H_ORC_STATES[i-1]+0.5, T_ORC_STATES[i-1], name , fontsize=14)
                name = ''
            else:
                plt.text(H_ORC_STATES[i]+0.5, T_ORC_STATES[i], i , fontsize=14)
```

FIGURE 4.24 – STATE POINT NUMERATION

To show the plot the code 'plt.show()' at the end of the subplot is written and a new plot can be started.

5.4.2 T,S AND T,H – DIAGRAM

The temperature entropy and temperature entropy diagrams are commonly used in thermodynamics to visualise changes of states in a process. For a reversible (ideal) process the area under the curve would be the transferred heat in the system. Both are also used for visualising optimising steps in this work, as explained earlier in this essay.

The phase diagrams used in this program show on the left side the temperature/ specific entropy and on the right side of the page a temperature/specific enthalpy phase diagram. The blue curve is the steam-water curve under which steam and water coexist of the specified fluid and the green line are the pressure levels of the cooling system and the preheater/evaporator/superheater sequence. State points are displayed as red dots with their number correlating to the cycle flow sheet. The diagrams can be printed to a file or on a printer by clicking the "Printer" sign in the tools menu at top of the diagrams. An T,s and T,h-diagram is shown in Figure 4.25

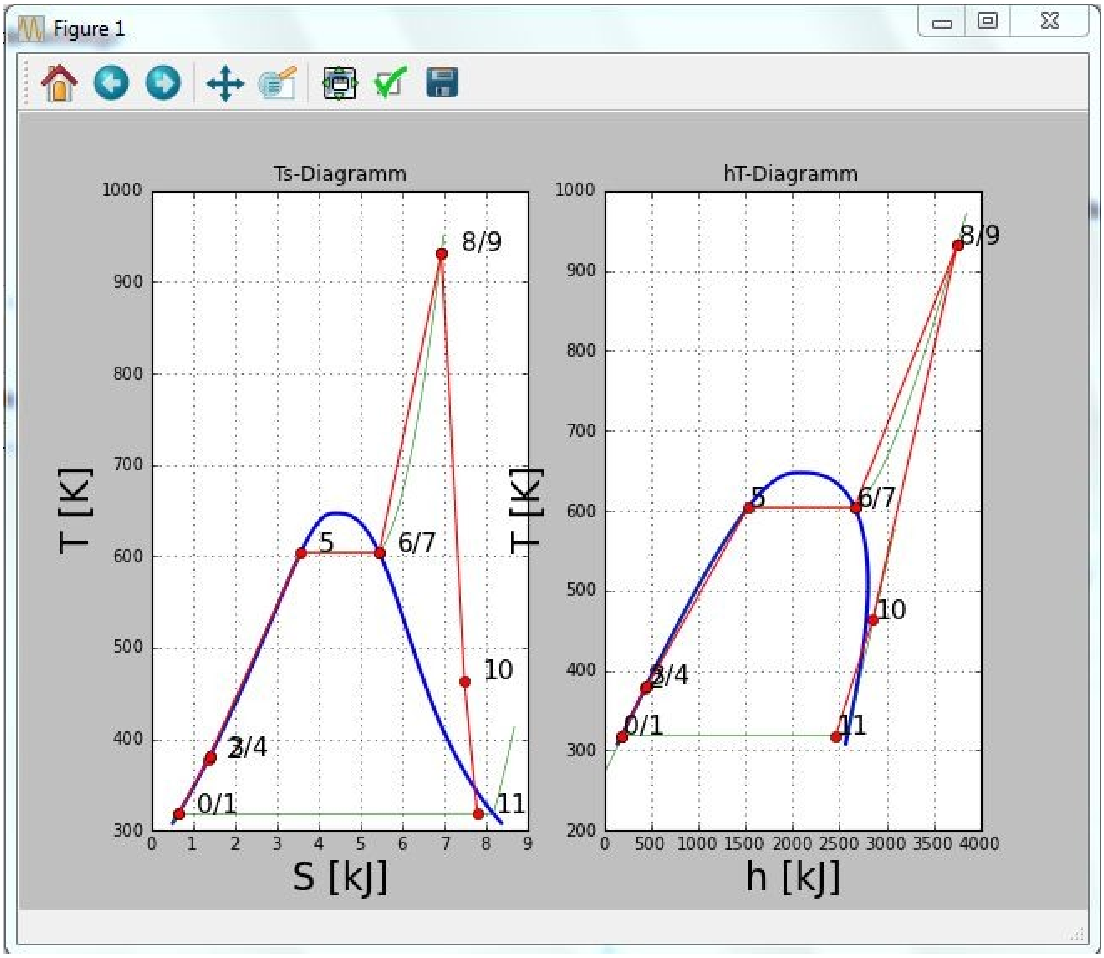


FIGURE 4.25 – T,S AND T,H-DIAGRAM

5.4.3 Q,T – DIAGRAM

The Q,T-diagram shows the temperature of the flue gas, oil and Cycle fluid over the transferred heat in the preheater/evaporator/superheater sequence. It gives a good overview over the reached pinch point and temperature differences in certain parts of the plant. A Q,T-diagram is shown by clicking on the “show QT” button on one of the result windows. An example from an SRC 1-stage cycle is shown in Figure 4.26

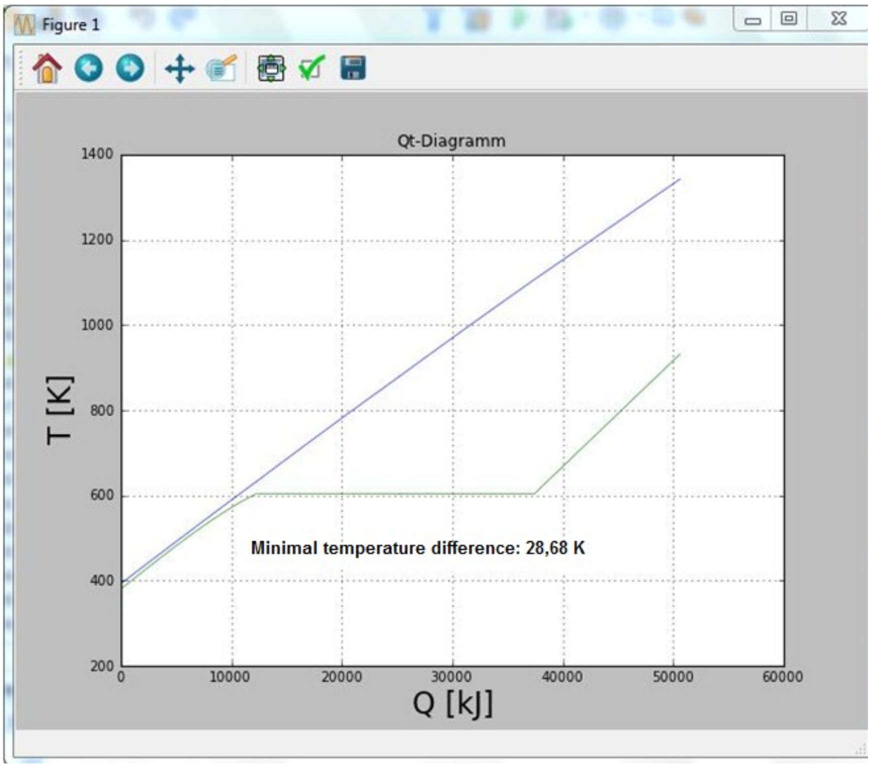


FIGURE 4.26 – Q,T DIAGRAM

5.5 ERROR MESSAGES

Several errors can occur during optimisation. Most of them happen in the background and give no feedback in the GUI to the user. All error messages are shown in the black command window running in the back. There are certain errors coming with a Pop-up message at the frontend GUI.

The message “An unknown error was created” means that an error occurred and the reason is a mistake in the code. This can be solved by excluding the ‘try’ and ‘except’ code in the “MAIN” file of the specific cycle and run the same routines again. In the output of the code program (for example NetBeans) the exact description of the error can be found there.

If in an SRC optimisation the Message “Optimisation doesn’t lead to satisfying result” appears there is no pressure level found with a satisfying pinch point. This happens when the combination of fluegas out and fluegas in temperature is not optimal for the given pressure levels. Because the fluegas low temperature in case of unsatisfying results is automatically adapted the temperature of the flue gas inlet should be raised or the pinch point lowered.

The message box shown in Figure 4.27appears if the fluegas inlet temperature is too low for optimizing in the supercritical area. The options possible are going back to the input and raise the flue gas inlet temperature or lower the pinch points between oil and ORC or flue and oil, do subcritical calculations with the given input or cancel the optimisation.

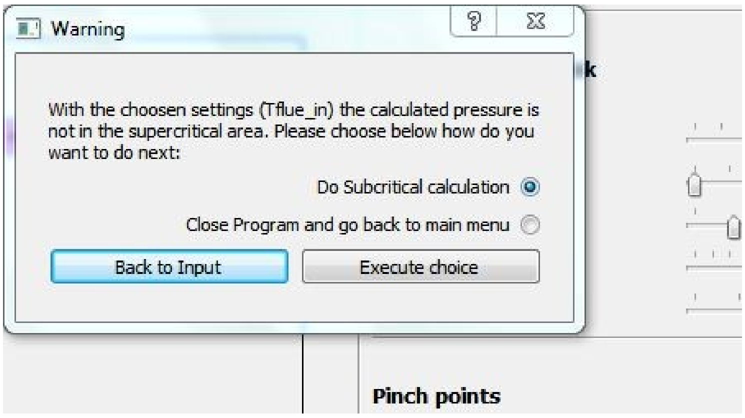


FIGURE 4.27 – PRESSURE ERROR ORC MESSAGE BOX

5. SIMULATIONS

In order to show the work of the tool, some simulations were done. The input was given from a cement producing facility. A List of the chosen input variables is shown in Table 5.1. For the simulations several other inputs, for example electrical efficiency of the turbine and fluegas composition, are needed. As they have not been available the standard values, given in the input windows, were taken. As working fluids isopentane, for the subcritical ORC, and propane, as proposed in (Wang, Zhang, Ouyang, & Thao, 2011) for the supercritical ORC, were used.

Input variables:

fluegas specification:		
input temperature	250	°C
output temperature	100	°C
massflowrate	22,3	kg/s

Pinches:			
Fluegas to oil	10	K	
oil to ORC	10	K	in all heat exchangers
cooling system	6	K	

cooling system:		
cooler temperature inlet	15	°C
cooler temperature outlet	25	°C

efficiencies:		
isentropic efficiency turbine	87	%
mechanical efficiency turbine	98	%
isentropic efficiency pump	72	%
mechanical efficiency pump	98	%

TABLE 5.1 – INPUT VARIABLES FOR THE SIMULATIONS IN THE OPTIMIZING TOOL

The results are listed below and show the most important output values calculated by the optimizing tool. Further there are the T,s- and h,T- diagrams.

6.1 Results for the subcritical ORC

variable	value	unit
Power Output	62	kW
η_{th}	17,39	%
Q_{in}	358	kW
P_{high}	2000	kPa
P_{low}	104,6	kPa
Temperatures		
T_{max}	155	°C
T_{low}	28,8	°C
massflowrates		
oil	0,995	kg/s
ORC	0,69	kg/s
cooling system	7	kg/s

TABLE 5.2 – SIMULATION RESULTS SUBCRITICAL ORC

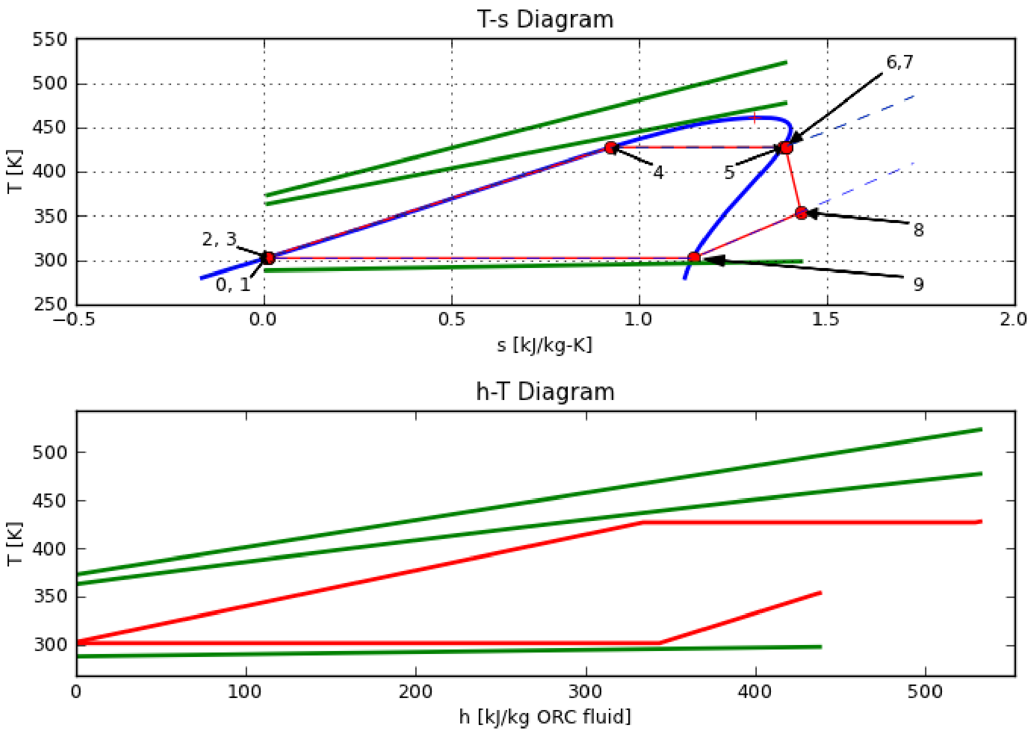


FIGURE 5.1 – T,S- AND H,T-DIAGRAM FROM SIMULATION OF A SUBCRITICAL ORC

6.2 Results for the supercritical ORC

variable	value	unit
Power Output	71	kW
η_{th}	19,98	%
Q_{in}	358	kW
P_{high}	6375	kPa
P_{low}	77	kPa
Temperatures		
T_{max}	230	°C
T_{low}	28,2	°C
massflowrates		
oil	1	kg/s
ORC	0,61	kg/s
cooling system	5	kg/s

TABLE 5.3 – SIMULATION RESULTS SUPERCRITICAL ORC

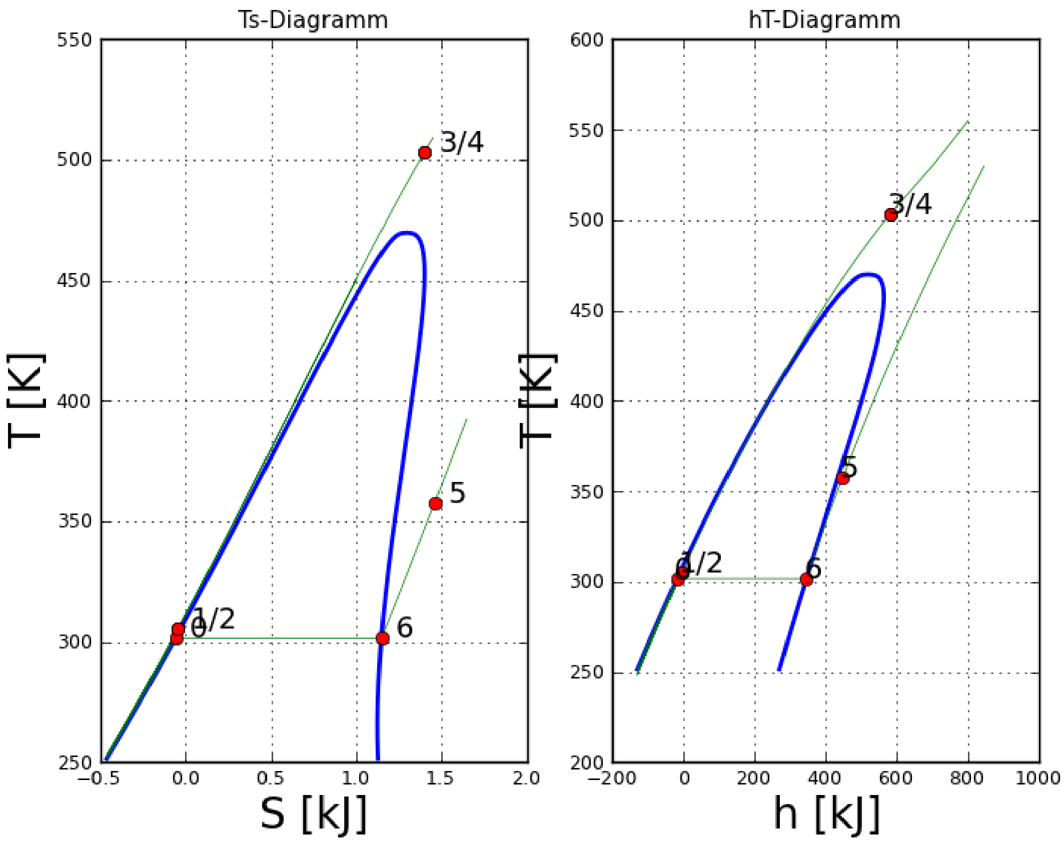


FIGURE 5.2 – T,S- AND H,T-DIAGRAM FROM SIMULATION OF A SUPERCRITICAL ORC

6.3 Results for the single pressure SRC

variable	Value	unit
Power Output	56	kW
η_{th}	16,9	%
Q_{in}	332	kW
P_{high}	99	kPa
P_{low}	4,5	kPa
Temperatures		
T_{max}	127,8	°C
T_{low}	31	°C
massflowrates		
SRC	0,14	kg/s
cooling system	1,4	kg/s

TABLE 5.4 – SIMULATION RESULTS OF SINGLE PRESSURE SRC

The results for the single pressure SRC show that a proper optimisation is not possible. The given temperatures are too low and the pressure gap for expansion is very small. Extra heat is needed for thermal deaeration because the top pressure is on 99kPa and 200kPa were suggested for the output of the turbine. The feedwater for that reason is heated to at temperature of 105°C and a pressure of 105kPa what is higher as the found maximum pressure after the optimisation.

6.4 Results for the dual pressure SRC

variable	Value	unit
Power Output	45	kW
η_{th}	14	%
Q_{in}	358	kW
HD cycle		
P_{high}	57,9	kPa
P_{low}	4,5	kPa
ND Cycle		
P_{high}	n/a	
P_{low}	n/a	
Temperatures		
T_{max}	240°C	°C
T_{low}	100	°C
massflowrates		
SRC	0,14	kg/s
cooling system	1,4	kg/s

TABLE 5.5 - SIMULATION RESULTS OF DUAL PRESSURE SRC

The optimisation for the dual pressure cycle with the same input shows that only the high pressure cycle is optimised and the low pressure cycle couldn't give a higher output. Because the minimum temperature for the dual pressure cycle is not adapted as for the single pressure we get a lower power output than with the single pressure.

6.5 Summary of the results

The simulation results show that for the given input a small ORC facility is a suitable approach. Such an ORC facility can provide a higher power output as well as a higher thermal efficiency. Between the sub- and supercritical ORC there is an increase of almost 15% in power output and 2,5% for the thermal efficiency. (see Figure 5.3 and Figure 5.4).

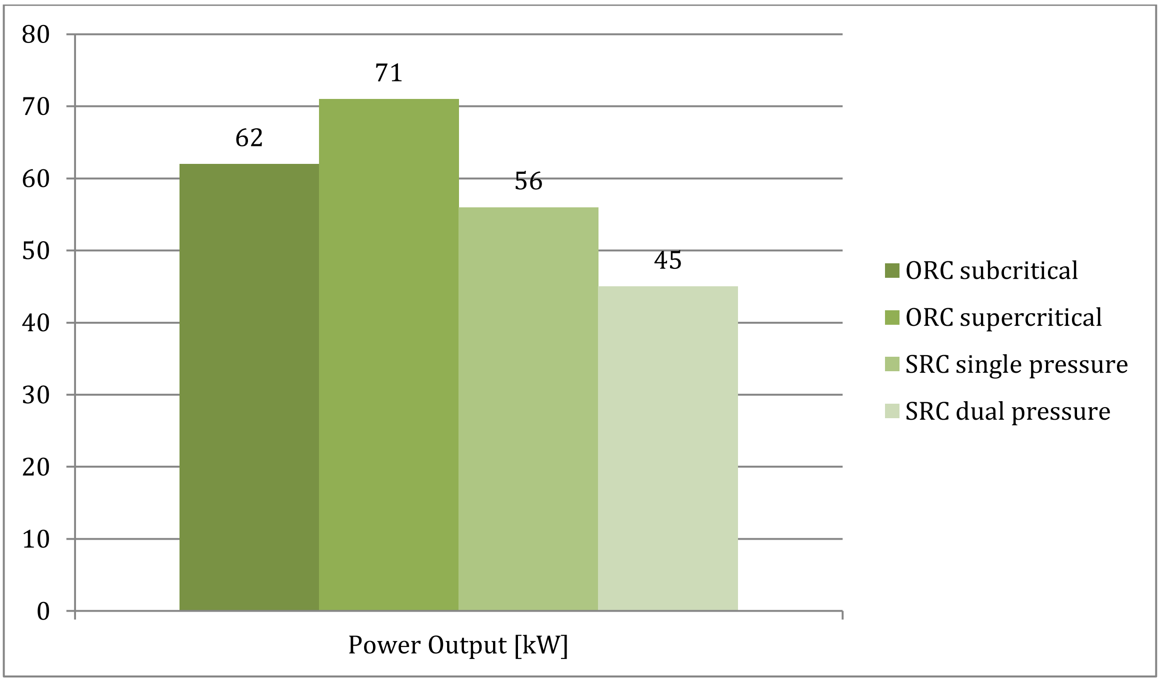


FIGURE 5.3 – SIMULATION RESULTS: INCREASE IN POWER OUTPUT

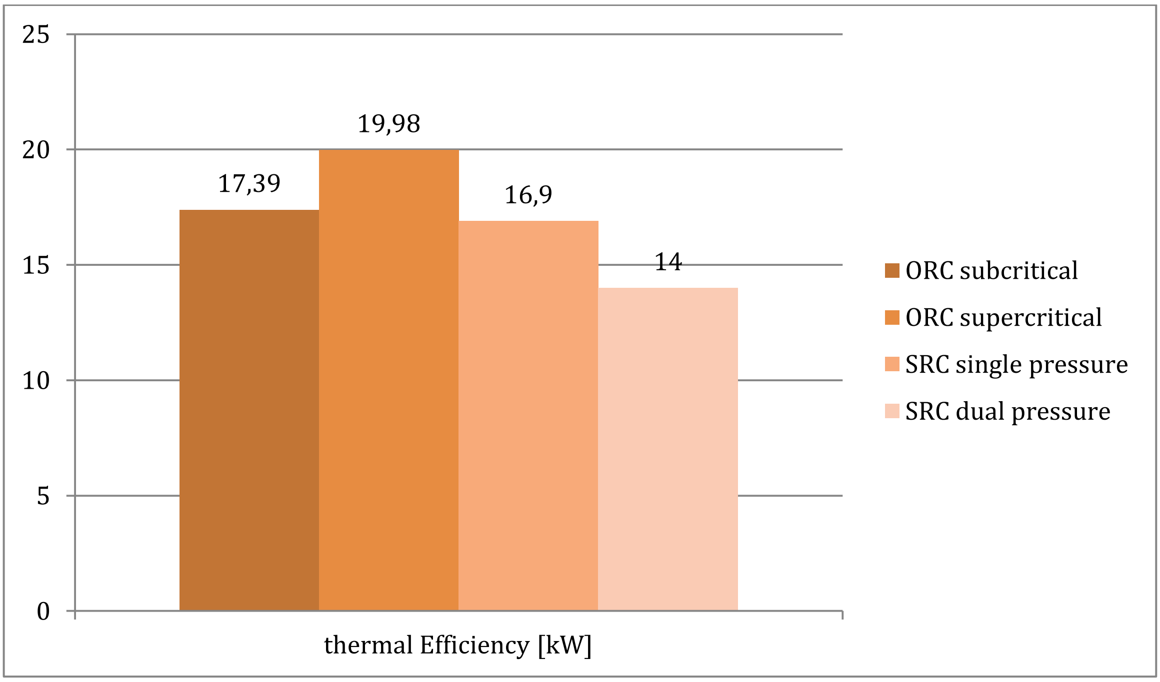


FIGURE 5.4 – SIMULATION RESULTS: INCREASE IN THERMAL OUTPUT

6. FUTURE WORK

This thesis can act as a base for further work on the optimizing tool. Cost calculation can be a part of it. As there is an output of pump and turbine data, the size of those parts can be determined. As turbine costs are a major financial part and up to 60% of a total ORC plant the determination is a crucial part when designing a plant. . DiPippo (Di Pippo R. , 2007)suggests a model on how to estimate the turbine size depending on sonic velocity. Rowshanzadeh (Rowshanzadeh) also suggests an equation to compute the turbine size based on volumetric flow and isentropic enthalpy difference. There are different types of small ORC turbines available from small scroll expander to axial turbines. For the design of heat exchangers the program provides kA values which are crucial for calculating the size dependent on the thermal heat coefficient k. For future work on heat exchanger design, an estimation of the heat transfer coefficient should be the main target. Some guesses as well as experiences for these values can be found in following literature (Di Pippo R. , 2007), (Lukawski, 2009), (Caixia), (McMahan, 2006) and (Rowshanzadeh). Dimensioning the cooling system can also be part of the cost calculations. The implementation of a material database for heat exchanger cost calculations, because of a big difference in temperatures between ORC and SRC different expensive materials are needed, can also be part of it.

Also a wider range of implemented cycles or additional options is suggested. As already mentioned there is the possibility of adding new fluids to the ORC calculations. Adding subcooling is a proper way to improve the efficiency by decreasing the average minimum temperature to get closer to the ideal Carnot process.

For the SRC feedwater preheating can be added, as this is leading to a lower use of fuel. Also different types of deaeration, like vacuum deaeration, can be implemented. This can lead to lower feedwater temperatures and therefore higher turbine output. Other optional improvements are to implement reheating after the first expansion, preheating the low pressure circuit with heat from the cooling system of the high pressure circuit. For the dual pressure cycle there are also different types for high and low sulfur fuels. Beside single and dual pressure cycles there is also the possibility of a triple pressure cycle. Implementation of this cycle is complex because of lot of optimizing variables. More information can be found in Kehlhofer (Kehlhofer, Hannemann, Stirnimann, & Rukes, 2009).

As a third cycle, especially for low temperature waste heat, the Kalina process is an option and (GAJEWSKI, LEZUO, NÜRNBERG, RUKES, & VESPER, 1989) gives additional information on the Kalina process.

7. REFERENCES

Baehr, H. (2006). *Thermodynamik* (13th Edition ed.). Springer.

Borsukiewicz-Gozdur, A. (2010). Influence of heat recuperation in ORC power plant on efficiency of waste heat utilization. *archives of thermodynamics, 31*(4), pp. 111-123.

Buecker, D. a. (2006). Reference Equations of State for the Thermodynamic Properties of Fluid Phase n-Butane and Isobutane.

Bull, J. (2010, May). *oCoCarbon*. Retrieved from http://oco-carbon.com/2010/05/19/eroei-of-electricity-generation/

Caixia, S. (n.d.). *Feasibility Study of Geothermal Utilization of Yangbajain Field in Tibet Autonomous Region, P. R. China.* Thesis, University of Iceland.

Classen Apparatebau Wiesloch GmbH. (n.d.). *http://www.apparatebau-wiesloch.com*. Retrieved from Wärmeübertragung mit organischen Wärmeträgern: Thermalölanlagen.

Colonna, P. N. (2011). Multiparameter Equations of State for selected Siloxanes. *Fluid Phase Equilibria*(244), pp. 193-211.

Concoucil, I. E. (2012). NON-PAPER OF THE SERVICES OF THE EUROPEAN COMMISSION ON ENERGY EFFICIENCY DIRECTIVE. Brüssel.

Di Pippo, R. (2007). *Geothermal Power Plants - Principles, Applications, Case studies and Environmental Impact* (2nd Edition ed.). Butterworth-Heinemann.

Di Pippo, R. (2007). *Geothermal Power Plants - Principles, Applications, Case studies and Environmental Impact (2nd Edition ed.).* Butterworth-Heinemann.

Drescher, U. (2008). *Optimierungspotenzial des Organic Rankine Cycle für biomassegefeuerte und geothermische Wärmequellen* (1st Edition ed.). Logos Verlag Berlin GmbH.

Drescher, U. B. (2007). Fluid selection for Organic Rankine Cycle in biomass power and heat plants. *Applied Thermal Engineering*(27), pp. 223-228.

Eurostat. (n.d.). *Eurstat - Statistik der Europäischen Kommission*. Retrieved from http://epp.eurostat.ec.europa.eu

F-Chart Software . (n.d.). *F-Chart Software*. Retrieved from http://www.fchart.com/

Fischer, J. (2010). *Arbeitsmedien für ORC-Prozesse zur Abwärmenutzung von Biogasanlagen.*

Foundation, T. F. (1998-2000). *The Free Software Foundation and the authors*. Retrieved from http://live.gnome.org/Dia

GAJEWSKI, W., LEZUO, A., NÜRNBERG, R., RUKES, B., & VESPER, H. (1989). *Der Kalina-Prozess. VGB Kraftwerkstechnik.*

Gedanitz, H. D. (2008). unpublished equation.

Geothermal-Recources. (n.d.). *Geothermal Recources*. Retrieved from http://www.geothermal-resources.com.au/

Green-Rock-Energy. (2009). *Green Rock Engergy*. Retrieved from http://www.greenrock.com.au/media/20090213%20SourcesofGE.jpg

Hofbauer, H. (2010). Skriptum zu Brennstoff- und Energietechnologie.

http://www.scilab.org. (n.d.).

Kehlhofer, R., Hannemann, F., Stirnimann, F., & Rukes, B. (2009). *Combined-Cycle Gas&Steam Turbine Power Plants.* PennWell Books.

Knoglinger, M. (2011). *Organic Rankine Cycle for waste heat recovery, Diplomarbeit.*

Köhler, S. (2005). *Geothermisch angetriebene Dampfkraftprozesse.* Potsdam.

Lemmon, E. H. (2010). NIST Standard Reference Database 23: Reference Fluid Thermodynamic and Transport Properties-REFPROP, Version 9.0, National Institute of Standards and Technology, Standard Reference Data Program.

Lemmon, E. W. (2006). Short Fundamental Equations of State for 20 Industrial Fluids. *Journal of Chemical Engineering Data*(51), pp. 785-850.

Lemmon, E., McLinden, M., & Wagner, W. (2009). Thermodynamic Properties of Propane. III. A Reference Equation of State.

Ling, L., & Gang, P. (2012, November). Optimization of low temperature solarthermal electric generation with Organic Rankine Cycle in different areas. *Elsevier/Applied Energy*, pp. 3355-3365.

Lukawski, M. (2009). *Design and Optimization of Standardized Organic Rankine Cycle Power Plant for European Conditions.* Thesis, RES - The School for Renewable Energy Sources.

McMahan, A. (2006). *Design and Optimization of Organic Rankine Cycle Solar-Thermal Power Plants.* Thesis, University of Wisconsin-Madison.

Microsoft. (n.d.). Retrieved from http://www.microsoft.com/en-us/default.aspx

Mobil. (n.d.). *http://www.mobil.com/Mexico-English/Lubes/PDS/GLXXENINDMOMobiltherm_603.aspx*.

NetBeans. (n.d.). *NetBeans*. Retrieved from http://netbeans.org/

NIST. (n.d.). *National Instutute of Standards and Technology*. Retrieved from http://www.nist.gov

Nokia Qt Development Frameworks. (n.d.). Retrieved from http://qt.nokia.com/

Opitz, E. (2011). *Auslegung von ORC- und Dampfkraftprozessen zur Abwärmenutzung, Diplomarbeit.*

Oracle. (2012). *Net Beans Wiki*. Retrieved from http://wiki.netbeans.org/Main_Page

Orr, L. (n.d.). *Changing the World's Energy Systems.*

Penoncello, S. G. (1995). A Thermodynamic Property Formulation for Cyclohexane. *International Journal of Thermophysics, 16*(2), pp. 519-531.

Polt, A. P. (1973). Fluid Thermodynamic Properties for Light Petroleum Systems.

Python Software Foundation. (n.d.). *http://docs.scipy.org*. Retrieved from /doc/scipy/reference/optimize.html.

Python Software Foundation. (n.d.). *http://www.python.org*.

Python, X. (n.d.). *http://www.pythonxy.com*.

Riverbank. (n.d.). *Riverbank Computing - PyQT Reference Guide*. Retrieved from http://www.riverbankcomputing.com/static/Docs/PyQt4/html/

Rowshanzadeh, R. (n.d.). *Performance and cost evaluation of Organic Rankine Cycle at different technologies.* Thesis, KTH Sweden.

Roy, J. P. (2010). Parametric optimization and performance analysis of a waste heat recovery system using Organic Rankine Cycle. *Energy*(35), pp. 5049-5062.

Saleh, B. K. (2007). Working fluids for low-temperature organic Rankine cycles. *Energy*(32), pp. 1210-1221.

Span, R. a. (2003). Equations of State for Technical Applications. II. Results for Nonpolar Fluids. *International Journal of Thermophysics, 24*(1), pp. 41-109.

Summerfield, M. (2008). *Rapid GUI Programming with Python and Qt.* Pearson Education, Inc.

The Engineering toolbox. (n.d.). *Engineering Toolbox*. Retrieved from http://www.engineeringtoolbox.com/gases-solubility-water-d_1148.html

Union, E. (n.d.). *European Union/Energy*. Retrieved from http://ec.europa.eu/energy/efficiency/eed/eed_en.htm

Union, E. (n.d.). *Summaries of EU legislation - Energy efficiency*. Retrieved from http://europa.eu/legislation_summaries/energy/energy_efficiency/index_en.htm

Vetter, C., Wiemer, H.-J., & Kuhn, D. (2011). Thermodynamische Auslegungrechnung eines Überkritischen ORC-Prozesses zur optmimierten geothermischen Stromerzeugung.

Wagner, W. a. (2002). The IAPWS Formulation 1995 for the Thermodynamic Properties of Ordinary Water Substance for General and Scientific Use. (A. I. Physics, Ed.) pp. 1-149.

Wang, E., Zhang, H., Ouyang, M., & Thao, B. (2011). Study of working fluid selection of organic Rankine cycle (ORC) for engine waste. *Elsevier/Energy (36)*, pp. 3406-3418.

Wei, D. L. (2007). Performance analysis and optimization of organic Rankine cycle for waste heat recovery. (Elsevier, Ed.) *Energy Conversion & Management*(48), pp. 1113-1119.

8. APPENDIX

8.1 NOMENCLATURE:

Basically all top values in ORC, SRC, flue gas, oil or cooling cycle are named with _top and all minimum with _low. Temperatures are given by the user in °C but then calculated to °K for the equations. Output is in °C again.

symbol in essay	symbol in programming code	Definition	Unit
$T_{ORC,high}$	T_ORC_high	maximum temperature in the ORC-cycle	[°K], [°C]
S_{max}	s_max	entropy at inclination point	[kJ/kg*K]
P_{crit}	P_krit	pressure at critical point	[MPa], [kPa]
ΔT_{min}	dT_	minimum temperature between heat exchanger and cycle	[°K], [°C]
	x_phigh	maximum moisture limit after expansion	[-]
$\dot{Q}_{fluegas}$	q_fluegas	heat flow from flue gas to process	[kJ/s]
$\dot{Q}_{transfered}$	q_trasferred	transferred heat from heat exchangers (fluegas/cycle), identical with $\dot{Q}_{fluegas}$	[kJ/s]
$\dot{m}_{fluegas}$	m_fluegas	mass flow flue gas	[kg/s]
$cp_{fluegas,in}$	cp_flue_in	heat capacity at flue gas inlet	[kJ/kg*K]
$cp_{fluegas,out}$	cp_flue_in	heat capacity at flue gas outlet	[kJ/kg*K]
$T_{fluegas,in}$	T_flue_high	temperature at flue gas inlet	[°K],[°C]
$T_{fluegas,out}$	T_flue_low	temperature at flue gas outlet	[°K],[°C]
Q_{cooler}	q_cooler	heat flow from cycle to cooler	[kJ/s]
$\dot{m}_{ORC}, \dot{m}_{SRC}$	m_ORC, m_SRC	mass flow of cycle medium	[kg/s]
$h_{ORC,cooler,in}$	h_ORC_STATES_XX	enthalpy of ORC fluid at cooler inlet	[kJ/kg*K]
$h_{ORC,cooler,out}$	h_ORC_STATES_XX	enthalpy of ORC fluid at cooler outlet	[kJ/kg*K]
m_{cool}	m_cool	mass flow in cooler	[kg/s]
cp_{cool}	cp_cool	heat capacity of water in cooler	[kJ/kg*K]

$T_{cool,out}$	T_cool_high	temperature at cooler outlet	[°K], [°C]
	eta_s_t	isentropic efficiency turbine	[-]
$\eta_{s,p}$	eta_s_p	isentropic efficiency pump	[-]
$\eta_{e,t}$	eta_e_t	electric efficiency turbine	[-]
$\eta_{e,p}$	eta_e_p	electric efficiency pump	[-]
$\eta_{m,t}$	eta_m_t	mechanic efficiency turbine	[-]
$\eta_{m,p}$	eta_m_p	mechanic efficiency pump	[-]
	T_flue_high	temperature level at flue gas inlet	[°K], [°C]
	T_flue_low	temperature level at flue gas outlet	[°K], [°C]
	T_u	surrounding temperature	[°K], [°C]
	dT_oil_ORC_end	minimum temperature difference at fluegas inlet and ORC after heater	[°K], [°C]
p_{cool}	p_cool	pressure level cooling system	[kPa]
$c_{p,oil}$	cp_oil	heat capacity of oil	[kJ/lg*K]
$T_{oil,high}$	T_oil_high	maximum temperature in oil circuit	[°K], [°C]
$T_{oil,low}$	T_oil_low	minimum temperature in oil circuit	[°K], [°C]
w_{pump}	w_p	work of pump	[kJ/kg]
$w_{turbine}$	w_t	work of turbine	[kJ/kg]
w_{net}	w_net	net work	[kJ/kg]
η_{therm}	nth	thermal efficiency	[%]
$\dot{m}_{ORC}$	m_ORC	mass flow ORC fluid	[kg/s]
P_{cycle}	Pcycle	work output	[kW]
Q_{IHE}	q_IHE	heat flow in internal heat exchanger	[kJ/s]
η_{IHE}	n_IHE	efficiency of heat exchanger	[-]
$\dot{m}_{district\ heating}$	m_FW	mass flow of water in SRC to district heating	[kg/s]

$q_{district\ heating}$	q_cheating	heat flow for district heating	
T_{slip}	T_slip	slip temperature for district heating	[°K],[°C]
$\dot{m}_{DEGAS}$	m_degas	mass flow for dearation	[kg/s]
$\dot{m}_{SRC,top}$	m_SRC_1	SRC mass flow before steam extraction	[kg/s]
h_{xx}	h_xRC_STATES_OPT_XX	enthalpy: stage point given by number in nomenclature	[kJ/kg]
s_{xx}	s__xRC_STATES_OPT_XX	entropy: stage point given by number in nomenclature	[kJ/kg*K]
T_{xx}	T__xRC_STATES_OPT_XX	temperature: stage point given by number in nomenclature	[K]
p_{xx}	p__xRC_STATES_OPT_XX	pressure: stage point given by number in nomenclature	[kPa]

TABLE 8.1 – LIST OF VARIABLES USED

In the program code there are several variables used but they declare themselves by their name. Following is a list of some shortcuts used:

ORC	variable for ORC cycle
SRC	variable for SRC cycle
_HD_	high pressure cycle
_ND_	low pressure cycle
_STATES_OPT	optimised state points
_SI_X	state values calculated with Refprop
_vec	vector of more than one value
_sup	variable in superheater
_ev	variable in evaporator
_cond	variable in condenser
_p_	state point of cooling system or heat circuit at pinch point
_pp_	state point of cooling system at start of condenser (can be the same like _p_)

8.2 FLOW CHARTS OF OPTIMISING PROCEDURES:

8.2.1 ORC SUPERCRITICAL:

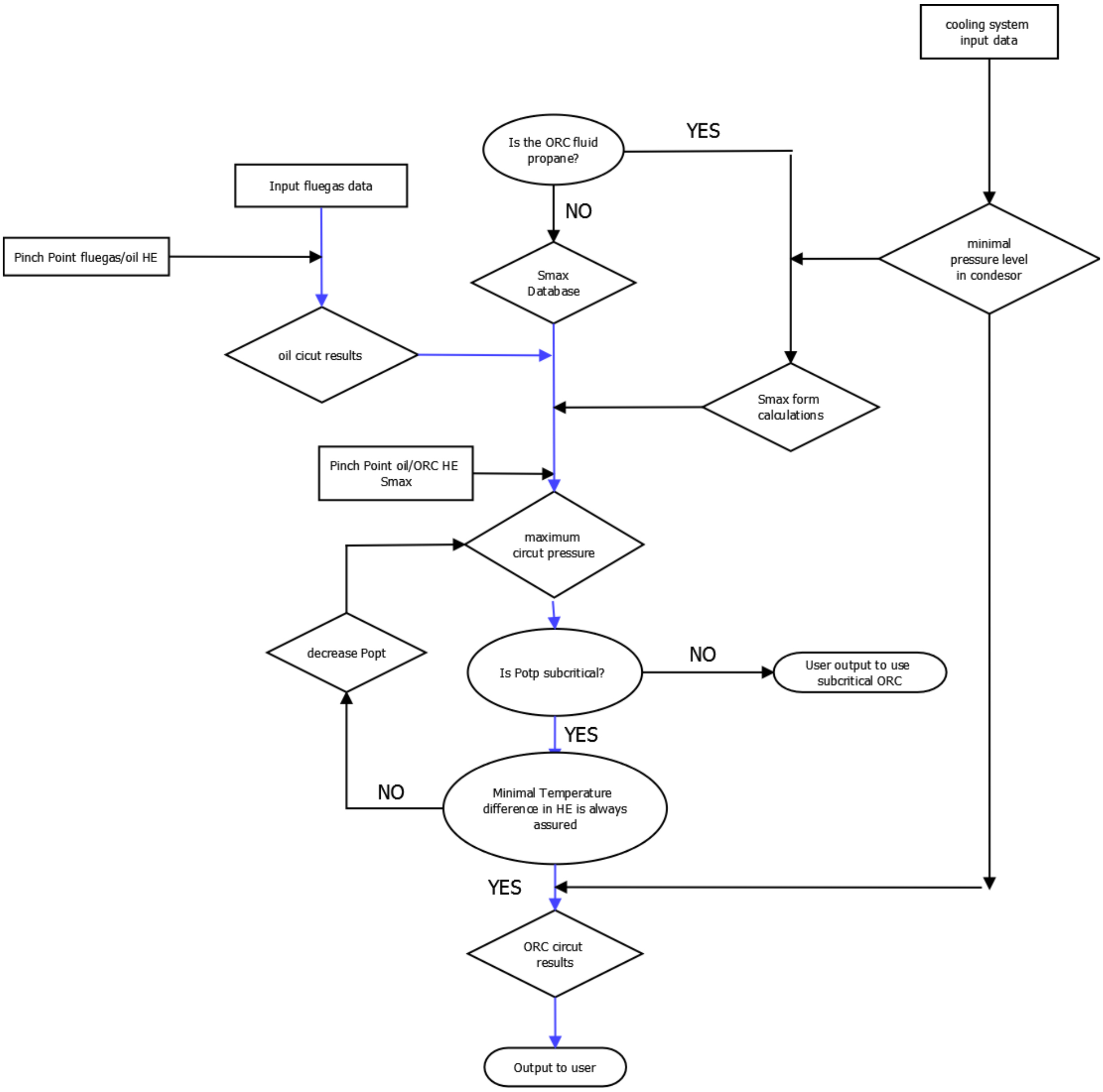


FIGURE 8.1 – CALCULATION STEPS ORC SUPERCRITICAL

8.2.2 SRC SINGLE-PRESSURE:

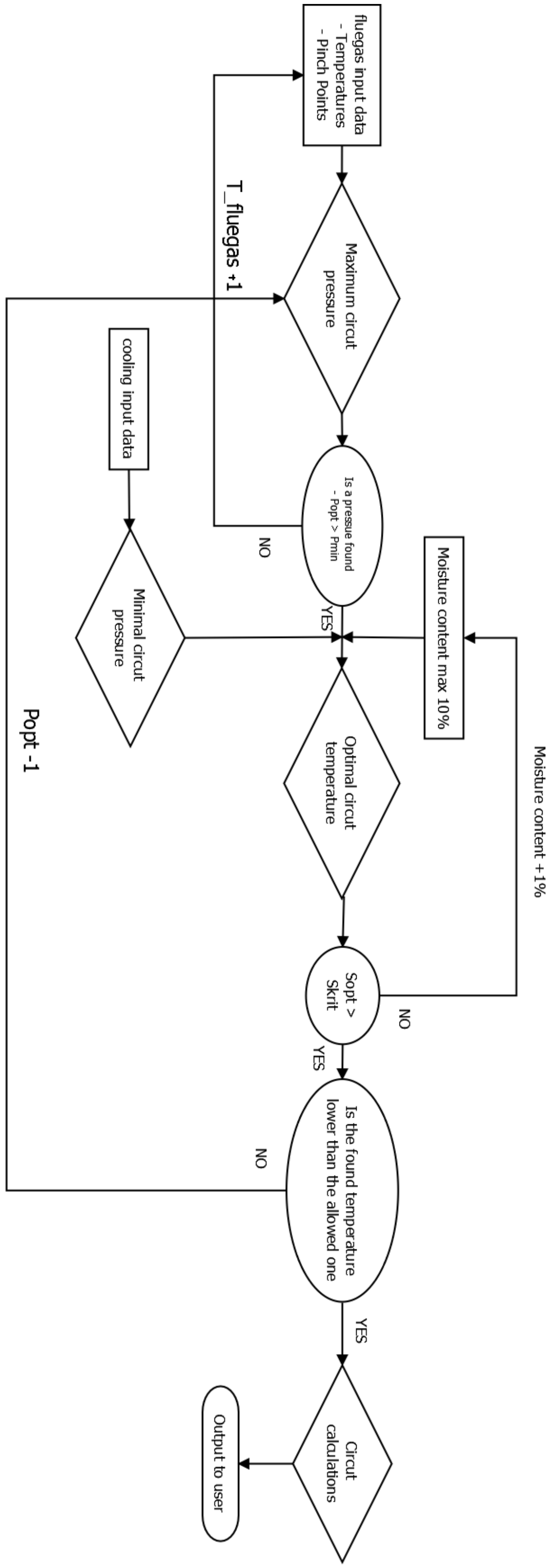


FIGURE 8.2 - CALCULATION STEPS SRC SINGLE PRESSURE CYCLE

8.2.3 SRC DUAL-PRESSURE:

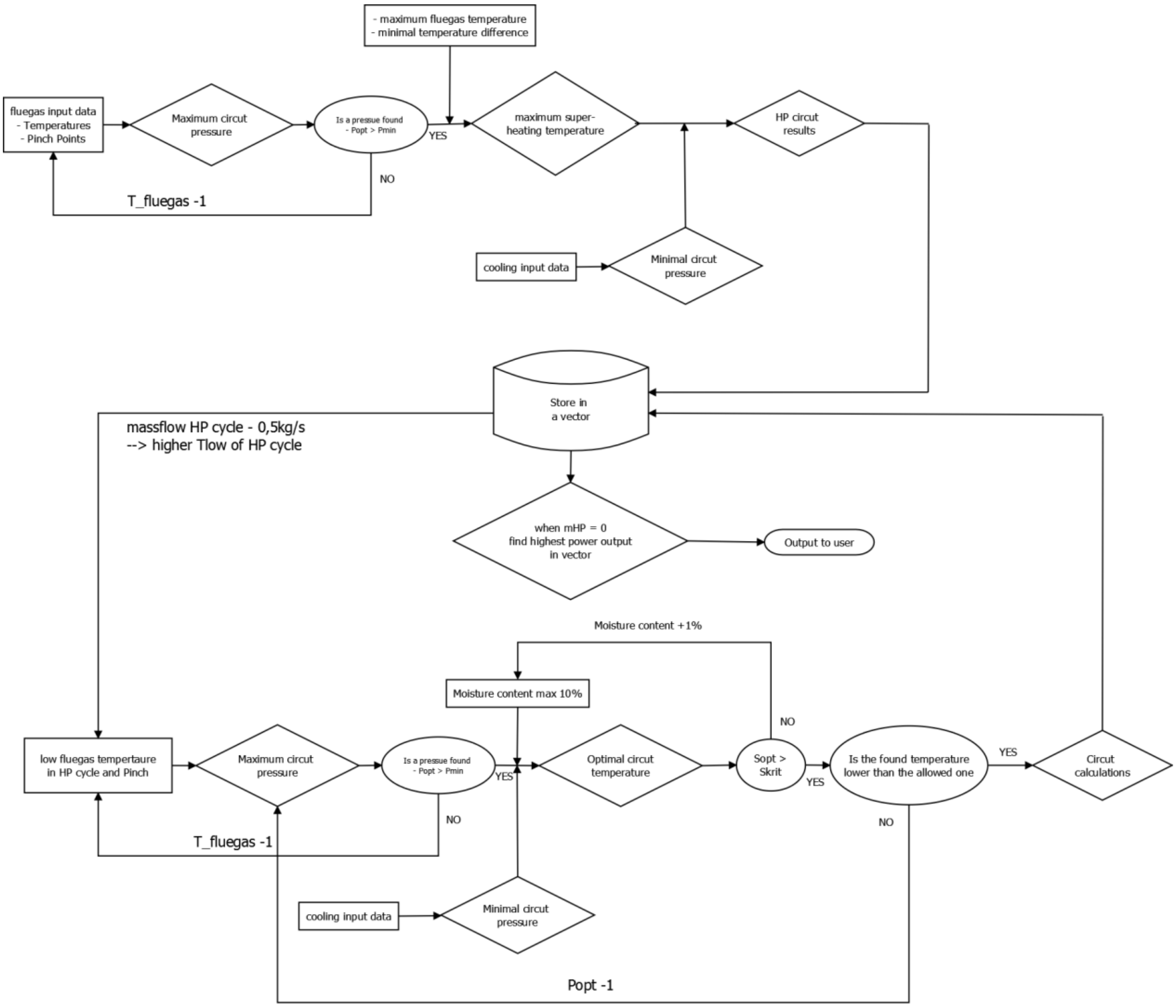


FIGURE 8.3 - CALCULATION STEPS SRC DUAL PRESSURE CYCLE