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Process thermodynamic study of a zero emisión Brayton cycles for de-heating of a gas flow and termal dimensioning of the primary heater

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Preface

This work was carried out in the Fakultät für Maschinenwesen (Proffessur für Thermische Energiemaschinen und Anlagen) of the Technische Universität of Dresden (Germany) from March 2020 to July 2020; under the supervision of Dr. Mario Raddatz and Dr. Thiago Gotelip and the revision of Pr. Dr. Fco. Javier Rey from the E.T.S. of Industrial Engineers of Valladolid (Spain). I want to say thanks them for their help and for their support.

Resumen

El objetivo del presente trabajo es diseñar y analizar termodinámicamente un sistema de enfriamiento que transformará la energía térmica de un flujo de mezcla hidrógeno-agua en energía eléctrica buscando la máxima potencia de salida y el menor coste de capital total. Para lograr esto, las propiedades especiales del dióxido de carbono, como la drástica variación del trabajo de compresibilidad cerca del punto crítico, serán utilizadas en una microturbina de gas de ciclo cerrado Brayton. También se estudiarán algunas de las configuraciones más comunes de sCO_2 (diseños en serie, en paralelo, recuperados y recalentados). La solución de mejor rendimiento será simulada y optimizada con OptDesign, un script de MATLAB® proporcionado por el Dr. Thiago Gothelip. Finalmente, los resultados serán validados y discutidos utilizando el modelo de software EbsilonProfessional®.

Palabras clave: Intercambiador, Brayton, CO2, Supercrítico, Cerrado

Abstract

The aim of the present work is to design and analyze thermodynamically a deheating system which will transform heat energy from a hydrogen-water mixture flow into electrical energy looking for the maximum power output and the lowest overall capital cost. In order to achieve this the special properties of carbon dioxide, like the drastic variation of the compressibility work near the critical point, will be utilized in a closed-loop Brayton cycle based micro gas turbine. Some of the most common sCO_2 configurations shall be studied as well (serial, parallel, recuperated and reheated layouts) and eventually the best performing solution will be simulated and optimized with OptDesign, a MATLAB® script provided by Dr. Thiago Gothelip. Finally results will be validated using EbsilonProfessional® software model and discussed.

Keywords: Exchanger, Brayton, CO2, Supercritical, Closed

Contents

Nomenclature

ΔT	Temperature difference
$\dot{m}$	Mass flow rate
$\dot{Q}$	Heat flow
η	Thermodynamic efficiency
A	Heat transfer surface area
h	Enthalpy
p	Pressure
T	Temperature
U	Overall heat transfer coefficient
W_c	Compressor work input
W_t	Turbine work output

Chapter 1

Introduction

It is widely known in the present world that climate change and global warming is a critical issue in modern society. The use of non-renewable energy sources is slowly damaging the world ecosystem. Although at this moment we are still dependant in these kind of energy sources considerable research has been done in this area. Investment in renewable technologies like wind power, and solar energy, has seen a huge increase in the latest decade. For this reason thermal generation industry who have a dominant share in the energy sector plays an important role. Their environmental impact largely depends on the source of heat used and energy systems transformation includes zero emission technologies. This project explores precisely this energy system transition by researching the potential of zero emission Brayton cycles for a waste heat recovery application.

1.1 Problem description

To recover heat from the external system, two flows are available for this application. Both are composed of hydrogen and water but each one with different mass fraction, mass flow rate, pressure and temperature. In order to absorb the heat from both flows, two heat exchangers will be used. These heat exchangers act as heat sources for the sCO_2 Brayton cycle. First flow, is composed of 2.24% of H_2 at 800ºC, 1.05 bar and a mass flow rate of 0.077 kg/s. The other flow is composed of 0.38% of H_2 at 630 ºC, 1.12 bar and a mass flow rate of 0.089 kg/s. Mass fraction are not necessarily a fixed value, however due to time frame it was not allowed to test further configurations. Both flows are required to decrease their temperature up to 120ºC. From this conditions it can be stated that heat input is fixed and given by equation 1.1, being $Q = 220.505$ kW the total heat input of the system. All of these conditions are gathered on Table 1.1.

$$Q = \dot{m} \cdot C_p \cdot \Delta T \tag{1.1}$$

Parameter	Units	Main HX	Secondary HX
Inlet temperature, T_i	[ºC]	800	630
Outlet temperature, T_o	[ºC]	120	120
Hydrogen mass fraction, $\%H_2$	[%]	2.24	0.38
Water mass fraction, $\%H_2O$	[%]	97.76	99.62
Mass flow rate, $\dot{m}$	$[kg/s]$	0.077	0.089
Inlet pressure, P_i	$[bar]$	1.05	1.12
Heat flow rate, $\dot{Q}$	$[kW]$	125.221	95.284

Table 1.1: Heat sources boundary conditions

On the other hand, working fluid has to fulfil some conditions regarding to its super-critical state. These are a minimum pressure and temperature of 73.8 bar and 31ºC respectively. In regard to temperature, air flow inlet temperature of the cooling system will be 25ºC which will cool down sCO_2 to 35ºC at maximum. Whereas for lower pressure, since it's an independent variable, can always be set to values above the critical point assuring the super-critical state of the fluid.

Compressor and turbine isoentropic efficiency will be set to 80% and 90% respectively as this is a common achievable value by nowadays standards [6]. Another condition, pressure drop, is also a relevant parameter to take into account as it has a huge effect on cycle performance. For this reason a value of a 2% of pressure drop in each heat exchanger (heat sources and cooler) will be set.

1.2 Objectives

Although there has been quite a lot prior research one in the field of super-critical carbon dioxide power cycles, many of them only focus on medium to high load level cases. This waste heat project deviates from the common view, studying and analysing several ways to use a heat flow to produce electricity that otherwise would be lost energy. To achieve this, an optimization process will be performed on the most suitable cycle layout always looking for maximize and minimize net power and total heat transfer surface area, respectively. First one, being the difference between the power output of the turbine and the power input needed by the compressor, has a direct impact on the electrical energy production rate. And the other, the sum of the surface areas of the three heat exchangers present in the system, is correlated to the total cost of the system. Therefore the aim of the current research is to find the best performance conditions while keeping the capital costs as low as possible. The objectives of the present research are outlined as follows.

1. Propose an alternative to waste heat recovery of high temperature using S-CO2 cycles.

2. Analyzing the potential of power generation with the S-CO2.

3. Optimization of the system to achieve the best operation performance.

4. Economical estimation analysis of the equipment.

1.3 Report organization

Chapter 2 presents a description of the main advantaged and disadvantages of the utilization of the supercritical CO_2 power cycles along with some of the unique properties of the fluid compared to its competitor in the power generation field like helium and steam. Then a by a quick review of the main events in the history of the cycle starting from middle of 20th century until the present day.

Chapter 3 discusses several configuration layouts: parallel, serial, recuperated and re-heated. Performance improvement over the simplest layout is studied and the feasibility and limitation of each one. Finally, the most suitable is selected for the optimization procedure in the latter chapter.

Chapter 4 analyses and optimizes the selected cycle layout which is performed through the variation of the different variables that have influence on net power output and total heat exchanger surface area. Then validation of the model will take place along with a cost estimation of the overall waste heat recovery system.

Chapter 5 summarizes the work done in the thesis and brings the conclusions.

Chapter 2

Background and history

In this chapter the advantages of the supercritical CO_2 over other fluid will be discussed along with its applications in the modern day industry. Also, a brief description of the most important events in the history will be presented.

2.1 Fluid characteristics

In most cases standard air, composed of 21% oxygen and 79% nitrogen is the working fluid choice. Being an "unlimited" resource this fluid is a must when working with open loop cycles like in aviation. But closed loop cycles doesn't have such limitation as the loss of working fluid is reduced to leaks inside the system. Inside this frame, several possibilities lie ahead. A noble gas like helium is a feasible option, particularly for nuclear power applications. Its main advantage over normal air is its high heat capacity value. Being an inert gas, like argon, its also less prone to corrosion making potential maintenance costs lower.

However, early studies of the project showed that using air and inert gases as working fluid the compressor temperature outlet was too high. This caused a temperature difference problem in the heat sources since a hot outlet temperature of 120ºC is needed for an optimum functioning of the external hydrogen engine. For this reason, a fluid with low compressibility work like super critical carbon dioxide was the choice. Thanks to its properties near the critical point the compressor work needed was lower and thus the temperature outlet at the compressor was also lower than the boundary limit of 120ºC.

Apart from this, super critical offers some other advantages over the other working fluids [3]. For example, due to its high power density, it can reach the same efficiency (about 45%) as a helium based cycle but at lower operation temperatures (550ºC compared to 850ºC). Thermal stress is reduced and, therefore, long lifetime can be assured. Also, this property makes super critical carbon dioxide based turbines installations incredibly small and compact compared to its counterparts (around 1/10 the size of a steam turbine and 1/3 of helium turbine). As a result of this the number of stages in both compressor and turbine is also lower. In terms of cost, $S - CO_2$ clearly wins as world inventory of helium is quite limited although its storage capacity is better.

2.2 History of supercritical carbon dioxide

There is no doubt that gas turbine technology has been a breaking point in the history of the human being. Dealing with heat energy has been a rough issue since it is one the most degraded forms of energy. The opportunity of transforming the heat into a profitable form of energy capable of infinite applications like electricity has brought society like we see it now. The oldest reference dates from 1948, when Sulzer Bros patented a partial condensation cycle [7]. After that the potential of this new power cycle was realized and research increased along with popularity.

In 1967, Ernest G. Feher developed a regenerative CO_2 cycle which worked entirely above critical pressure and compression was achieved in liquid phase [4]. Due to its compactness it was proposed for electric power generation and shaft propulsion. Also, he found out a pinch-point problem at the recuperator which was a limitation for the cycle efficiency. However, he failed in optimizing the compressor inlet pressure keeping it at a constant value. In regard to pressure drop, he stated that a fractional value of 0.075 translated into a 5% decrease in cycle efficiency. Finally he concluded that compressor efficiency did not have a significant influence in cycle performance and that the volume to power ratio had low values.

In the same time, Angelino did a thorough study mostly on condensation cycles [1, 2]. He stated that CO_2 power cycles have better efficiency rates over 650ºC of inlet turbine temperature, however at lower temperatures, while being less efficient this cycle can also be competitive due to its simplicity and compactness which reduces the overall costs. He also mentions the geographical limitation of the low temperature cooling water requirement. In regard to the pinch-point problem, he proposed four compound condensation cycles, being the re-compression layout the most promising in terms of performance. The study, fails in some aspects such as the low compressor efficiency estimation (85%), the assumed pressure drop fractional value and the excessive minimum temperature difference in the recuperator which can be improved with the utilization of modern compact heat exchanger technology. Besides, the compressor inlet temperature was too close to the critical point which may lead to cavitation issues. Another relevant aspect mention in Angelino's research was the low expansion work compared to steam. This means very low volumetric flow rate and therefore, compact turbomachinery as shown in Figure 2.1. Overall despite the deficiencies the study remains as an excellent comparison of the different CO_2 layouts.

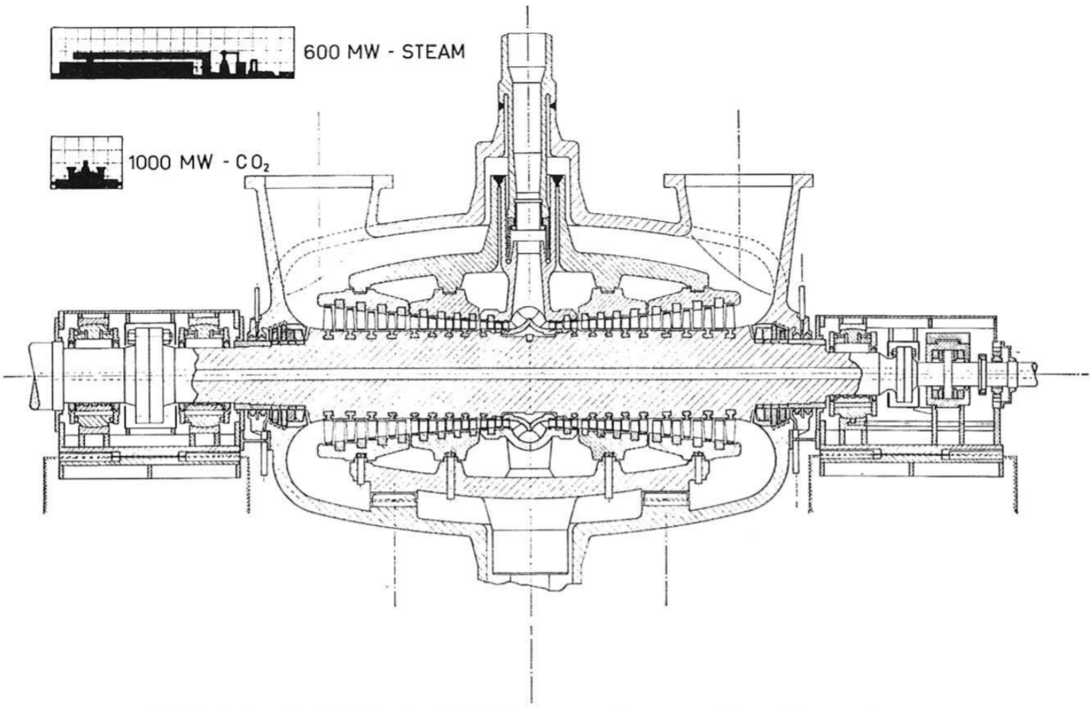


Figure 2.1: Carbon dioxide turbine for 1000 MW net output; inlet gas conditions, 300 atm, 565 deg C. From Angelino [1].

These last two research works were made a lot of improvement in the field of sCO_2 power cycles however they were never deployed in practice because of the insufficient experience in turbomachinery design and compact heat exchanger availability. However, in the recent this technology has received a lot of attention due to the seek of new thermally and cost efficient cycles. Some of the most popular applications in the present day are waste heat recovery, high temperature fuel cells, solar energy, generation IV nuclear reactors and geothermal energy. It is also significant as a solution to global problems such as fossil energy shortage and environmental issues.

Chapter 3

SCO2 cycle layout discussion

In the following chapter will take place the discussion of several cycle layouts. Then, the best option with the best performance characteristics will be selected for the optimization process in the next chapter.

3.1 Parallel and serial layouts

At first, taking advantage of having two heat sources there are two configurations that can be arranged without further components. These are the parallel and serial layouts, depending on the placement of both heat sources. In figure 3.1 is shown the the first proposed option. Although it seems reasonable, the need of cooling down both hydrogen flows until 120ºC made this option unfeasible. This is because the outlet temperature at the cold side of the first heat exchanger will be always higher than the hot side outlet temperature of the secondary heat exchanger.

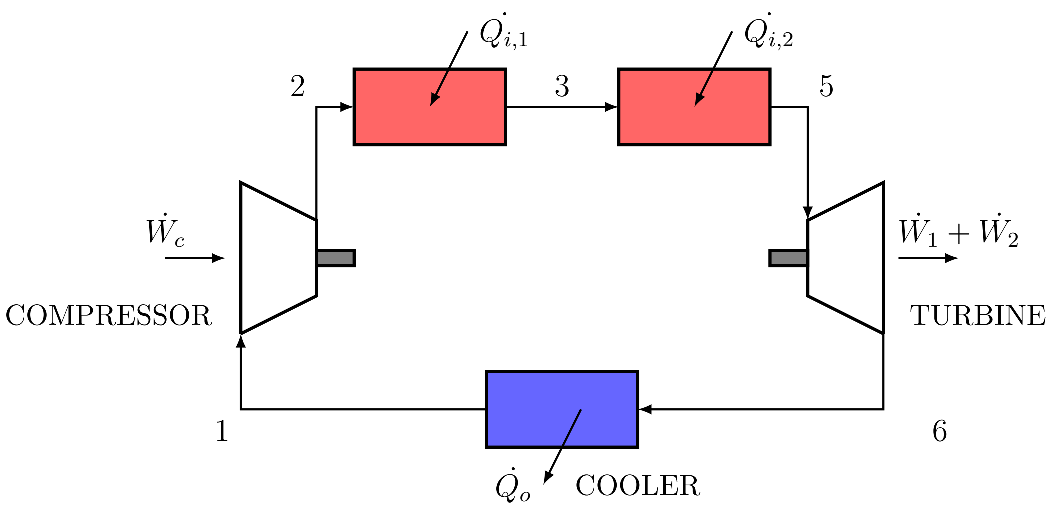
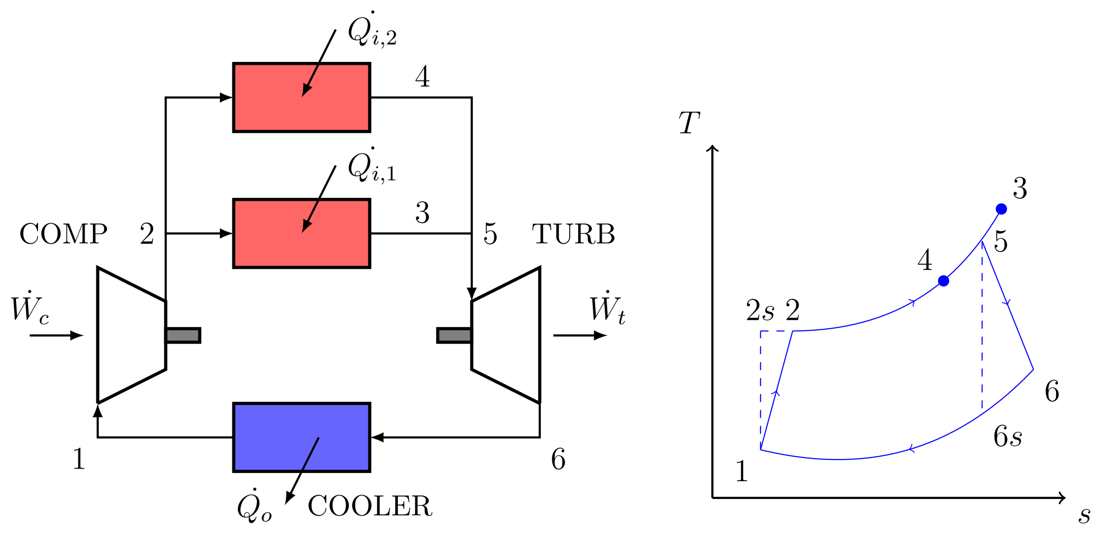


Figure 3.1: Serial layout

The other option left, is to place the heat sources in a parallel configuration (Figure 3.2) so both hydrogen flows can be fully cooled down. The only downside is that a mixing chamber would be needed at point 5 since the pressure could be not equal. However, as the pressure drop in both heaters is set to 2%, an isobar ideal mixture of both branches is assumed.



(a) Layout

(b) Temperature-entropy diagram

Figure 3.2: Parallel cycle

3.2 Recuperated cycle

In order to close the cycle loop, flow has to return to initial conditions through the utilization of a cooler by discharging heat to the atmosphere, which translates into wasted energy. To partially solve this issue use of a recuperator is highly recommended if temperature at the turbine outlet is higher than the temperature at the compressor outlet. This recuperator, another kind of heat exchanger, takes advantage of the extra energy to pre-heat the fluid flow before entering the heaters, thus increasing the cycle efficiency.

An example of this modified configuration is represented in the figure 3.3. The recuperator can work in a counter-flow or cross-flow setup and pressure losses should be expected like every other heat exchanger in the system. In regard to the temperature-entropy diagram (Figure 3.4) turbine exhaust flow cools down from state 7 to state 8 while compressor outlet heats up from state 2 until state 3. Consequently, heat input from the primary heater is only needed from state 3 to state 4 and 5 for each heater. Mathematically, heat flow per mass unit is as follows,

$$\frac{\dot{Q}_i}{\dot{m}} = \frac{\dot{Q}_{i,1} + \dot{Q}_{i,2}}{\dot{m}} = (h_4 - h_3) + (h_5 - h_3) \tag{3.1}$$

Adding a recuperator does not decrease the net power output from the turbine, it only saves heat input energy. Therefore, cycle efficiency increases.

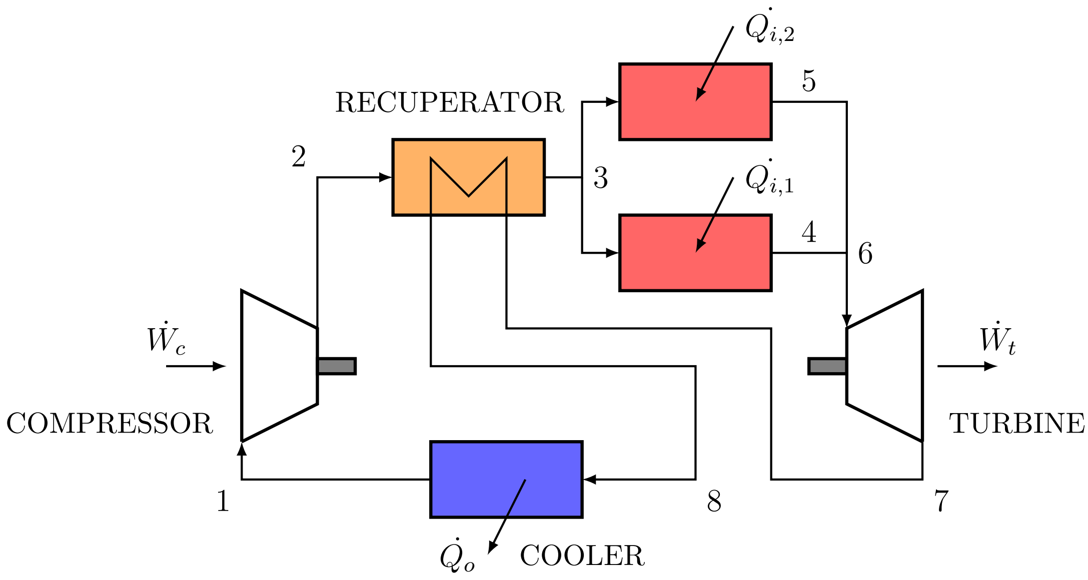


Figure 3.3: Recuperated Brayton cycle

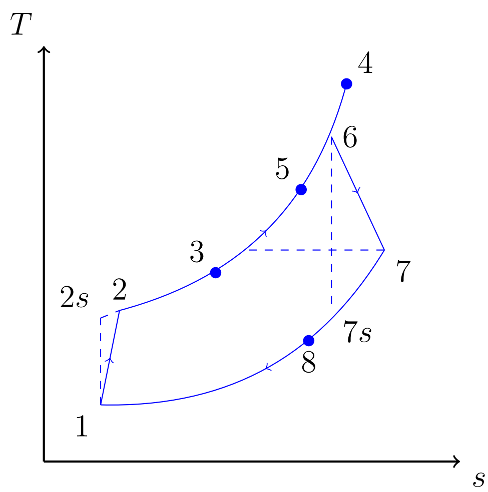


Figure 3.4: Temperature-entropy diagram for the recuperated cycle

Note, though, that a higher specific enthalpy h_3 means a lower needed heat input. This is specially relevant when saving up fuel is an issue. Thermodynamic laws states that heat can only flow from high temperature to low temperature if no work is applied. This means, that the maximum temperature value for cold outlet is the temperature at the hot inlet. In other words, h_3 would be equal to h_7 in an reversible heat exchanger. In reality, a temperature difference of zero would mean a heat exchanger of infinite surface so the goal here is to minimize the temperature difference between the cold and the hot flow while keeping the volume at a reasonable value. Besides, the capital cost that it would take would nullify the benefit of heat saving.

The efficiency of a recuperator is evaluated by the following expression

$$\eta_{rec} = \frac{h_3 - h_2}{h_7 - h_2} \tag{3.2}$$

where numerator represents the increment of enthalpy through the recuperator and denominator represents the maximum theoretical enthalphy increment. The closer h_3 and h_7 get between each other the more the efficiency value approximates to one.

However, it should not be forgotten that there is a boundary condition that requires the hydrogen flow temperature to be at maximum 120ºC at both heater outlets. This conditions, leaves almost no temperature range for the potential energy absorption in the recuperator making the recuperated cycle an unfeasible option.

3.3 Re-heated cycle

This layout is aimed at improving the cycle efficiency by increasing the equivalent Carnot efficiency for the cycle (equation 3.3). There are two ways of modifying this limit. One is to lower the cold side temperature at which the heat flow from the cooler evacuates. Often this is set by the weather conditions of the operating system so it's non modifiable, in this case this value is fixed by the air flow temperature of 25ºC which can cool down sCO_2 to a minimum temperature of 35ºC. The other choice is to vary the average temperature at which the heat is absorbed by the system. This last strategy is used by the layout, through the heating of both stages. In order to achieve the best efficiency, one should split equally the pressure ratio between both expansion stages, this way inlet and outlet temperatures remain the same. Although this is true for ideal layouts, in reality split ratio may not be exactly one due to the variation of specific heat capacity in real gases. Regarding s-CO2 this is not a big issue since at this operating temperature it behaves as a real gas.

$$\eta = \frac{W}{Q_H} = \frac{Q_H - Q_C}{Q_H} = 1 - \frac{T_C}{T_H} \tag{3.3}$$

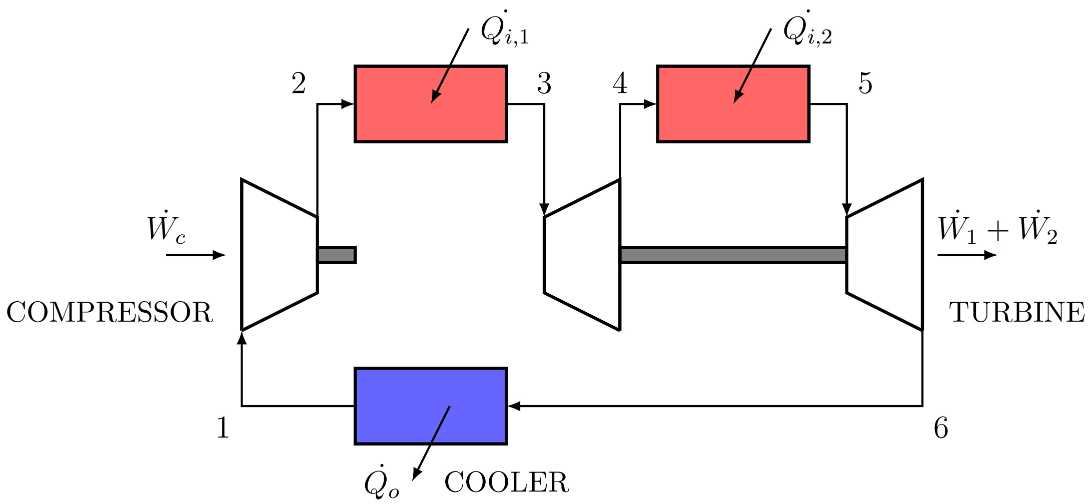


Figure 3.5: Reheated Brayton cycle

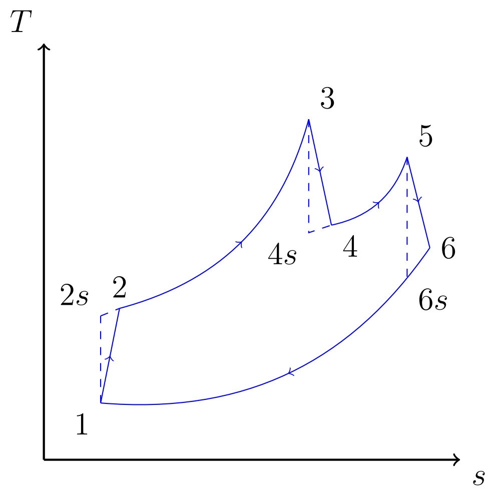


Figure 3.6: Temperature-entropy diagram of the reheated cycle

Standard configuration of a re-heated multi-stage Brayton cycle and its temperature-entropy diagram are shown in figure 3.5. After gas expansion at the first stage the gas flow re-enters the primary heater and re-heats reaching at constant pressure state 5. Then, it enters in a second stage completing expansion until state 6. Prime benefit from this configuration comes from the slightly deviation of isobaric lines when there's an increase in entropy. This causes that the total amount of work produced by the multi-stage turbine is higher than if the expansion only occurred in a single stage turbine without re-heating. Therefore, net output power is also increased. Knowing that turbine outlet temperature will be also higher this could be used in benefit of the recuperator and improving the overall cycle efficiency. It is also possible to set more than one re-heating stage, however implicates more heat exchangers and turbines, hence higher capital costs that may offset any benefit from the layout.

Chapter 4

Optimization

In this chapter standard cycle will be analysed and optimized looking for maximizing net power and minimizing the sum of heat transfer surface areas of both heat sources and the cooler as a cost-effective solution is sought. Higher and lower pressure as well as outlet temperatures at the heat sources will be the main variables to be optimized. Other dependent variables as mass flow, total equipment cost or surface area of each one of the heat exchangers present in the cycle will also be regarded.

4.1 Data analysis

This study implements a design analysis of S-CO2 cycles using the OptDesign device developed in to evaluate the performance of carbon dioxide cycles [5]. The tool performs multi-objective computer-aided optimizations for different applications and configurations of S-CO2 cycles. In this way, it allows investigating the best-operating conditions, and assists in the preliminary development of the equipment and architecture of the cycle. The program is compiled in MATLAB®, and determines the thermodynamic properties (based on Refprop® data library) for each equipment.

The results from the study show that the system can output from 18 kW to 29 kW of net power and the total heat exchanger surface area ranges from 8 m^2 to 18 m^2. Figure 4.1 shows the modelled layout in EbsilonProfessional® evaluated in the optimum conditions.

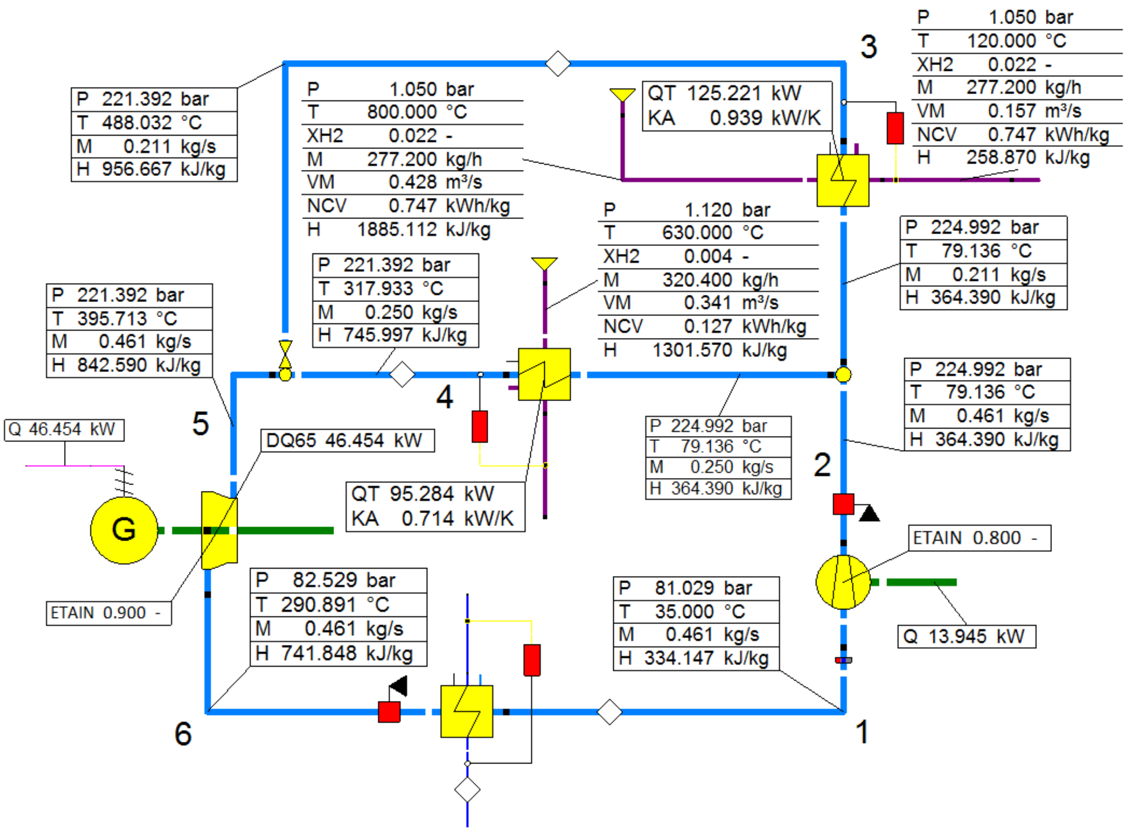


Figure 4.1: Circuit diagram at optimized conditions

4.1.1 Pressure effects

There are three variables related to pressure; higher pressure, lower pressure and pressure drop. First two ones are independent variables, in other words, it's value can be chosen. Third one, is prefixed by the boundary conditions, in this case 2% in each heat exchanger. In this section, higher and lower pressure will be focused on the analysis.

From Figure 4.3a it is seen the proportionality relation between net power output and higher pressure values. Assuming a higher pressure value means a higher pressure ratio, the net power increase is due mostly to the higher enthalpy drop in the turbine. Although a higher pressure ratio means a higher work input in the compressor as well, the growth rate is always lower than of the turbine.

However, it is not possible to increase this value over 225 bar because of boundary conditions of the hydrogen flow at heaters outlet. As pressure ratio increases the compressor outlet temperature will be also higher. If this value reaches the 120ºC boundary, hydrogen flow wouldn't be fully cooled down. Besides, a lower temperature difference would mean higher surface area required by the heaters and therefore, higher capital costs. In regard to the cooler, it is not affected by a higher pressure change since all its input and output conditions remain stable.

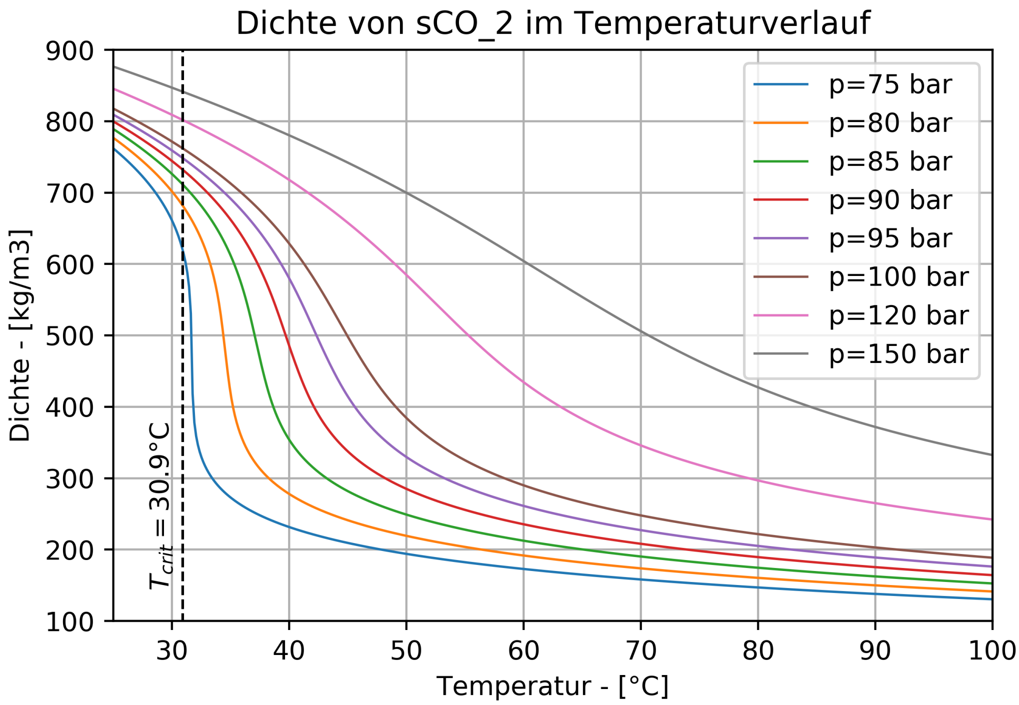
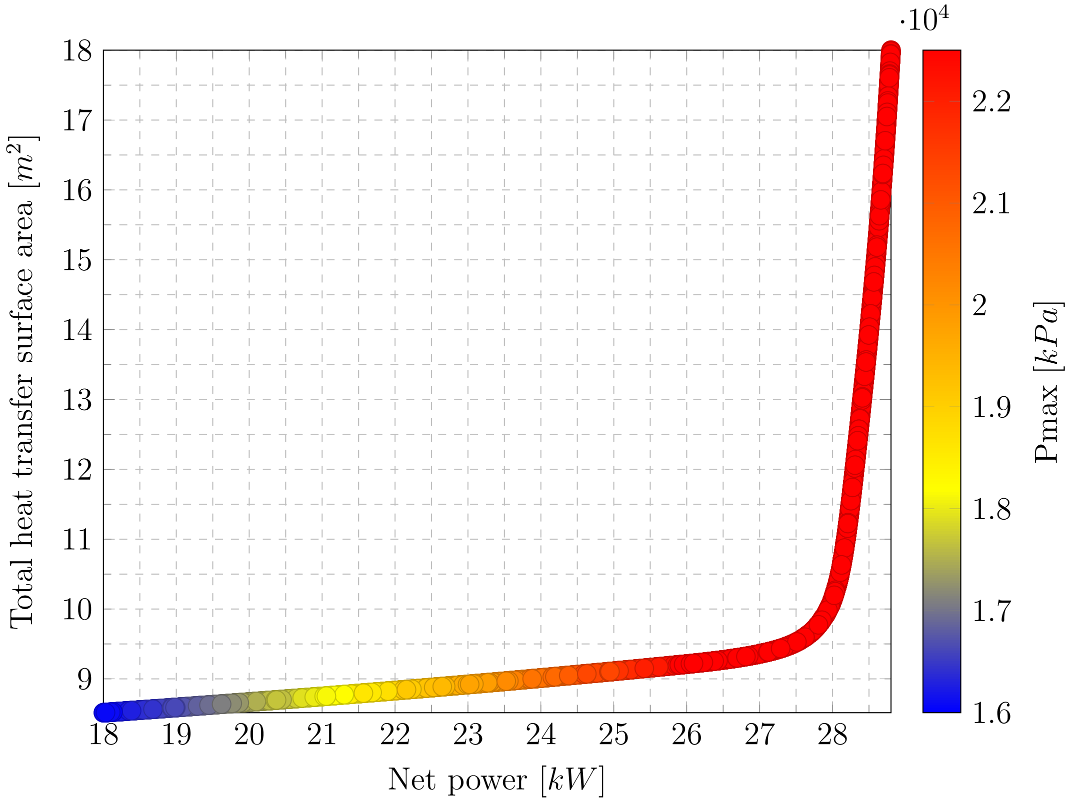
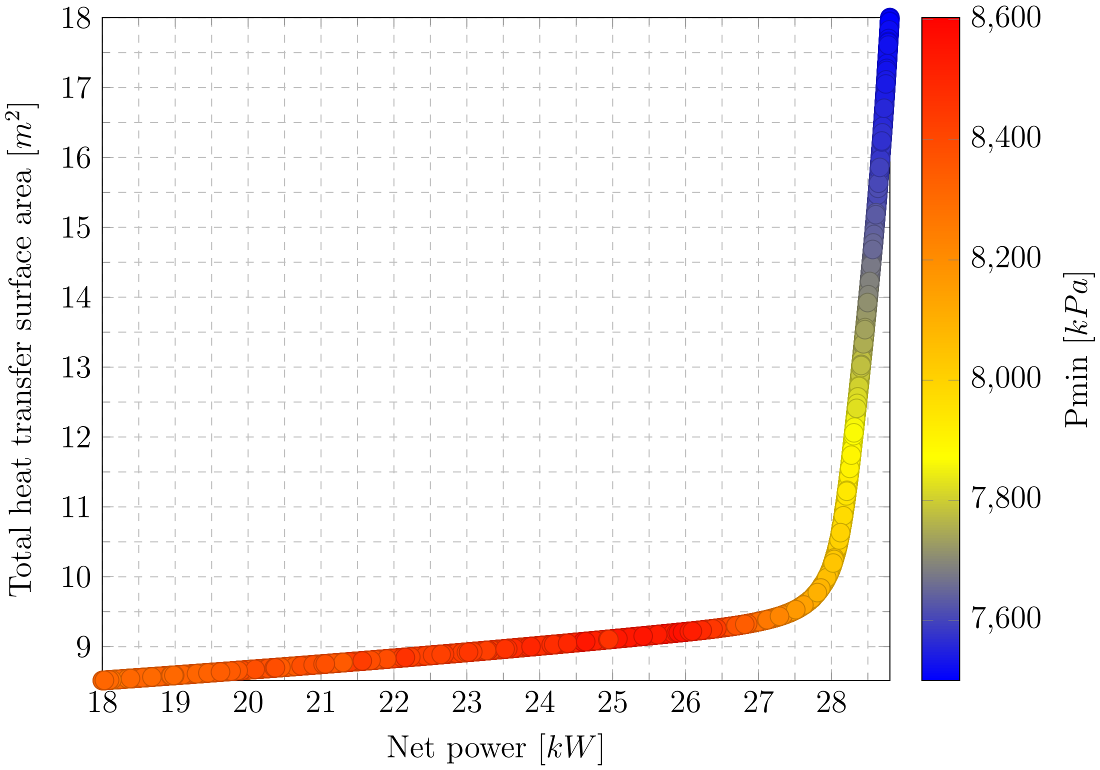


Figure 4.2: Density of CO_2 in super-critical state

On the other hand, lower pressure can also be lowered in order to achieve higher pressure ratio and therefore, higher net power output generally. Particularly, this parameter is very relevant due to the variation of the properties of sCO_2 near the critical point. The behaviour is shown on Figure 4.2. In ideal conditions, fluid temperature and pressure should be 73.8 Bar and 31ºC, respectively in order to take advantage of the low compressibility work at this point which would mean an even lower work input required by the compressor. However, it was unfeasible to reach these values, reason for this is that the air flow from the atmosphere has an inlet temperature of 25ºC and is able to cool down sCO_2 up to 35ºC which is not exactly the critical point. This causes the abnormal behaviour in Figure 4.3b at lower net power values.



(a) High pressure effects



(b) Low pressure effects

Figure 4.3: Pressure effect

4.1.2 Heater outlet temperature effect

Outlet temperatures in each heat source are independent variables as well. Since the heat flow rate from the hydrogen stream is constant in both sources, the result from changing the temperature is a variation of the overall carbon dioxide mass flow. In terms of net power, the increase in heater outlet temperature causes a higher enthalpy drop in the turbine however, it's often offset by the lower values of mass flow. Besides, the compressor work input increases as well making the performance improvement not so significant.

Furthermore, temperature parameter cannot be increased infinitely due to two reasons. First one, because the smaller the difference it gets inside the heat exchanger the higher the surface area is required. The other is about metallurgical issue since if temperature reaches values above 750-800ºC, higher heat resistant metal alloys should be used for the turbine components.

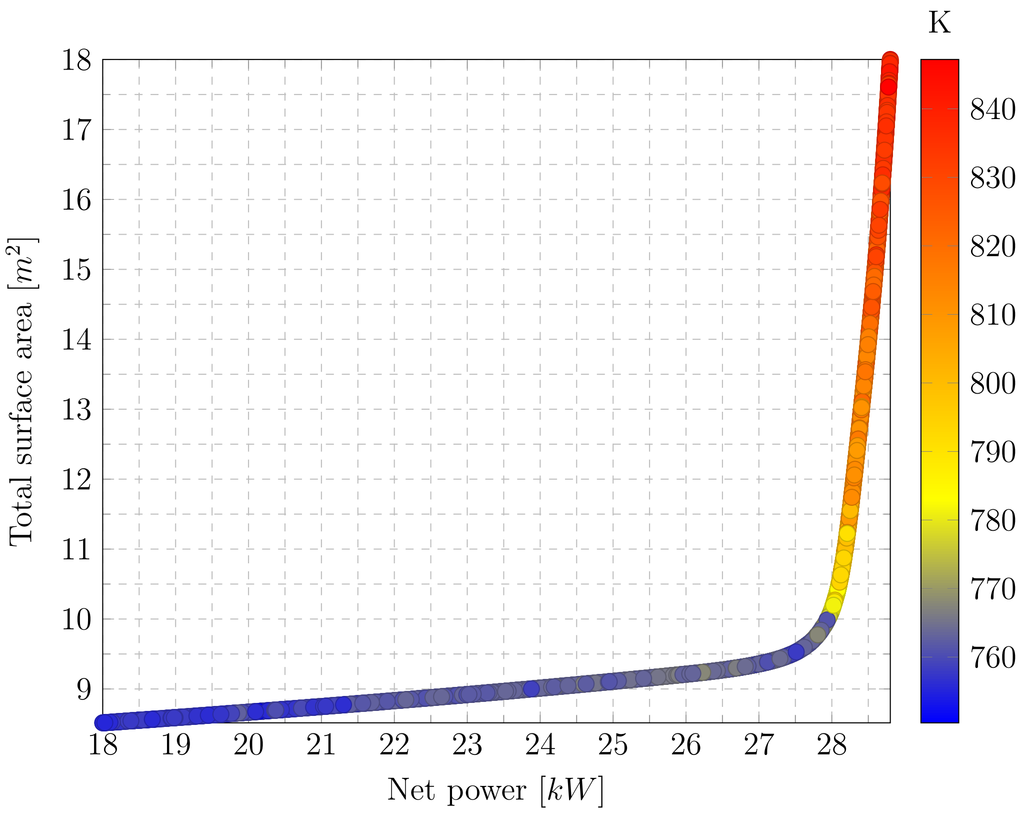


Figure 4.4: Main heater

Unlike net power, total heat transfer surface area is significantly affected by temperature, mainly because of the decrease in temperature difference of the heat exchanger which leads to higher effectiveness values, higher surface area require and therefore, higher capital costs. In regard to the cooling system, it's affected by the consequent higher turbine outlet temperature which leads to higher surface area required. Regarding the results an optimum point would be around 650ºC for the main heater and 500ºC for the secondary one. This setup will result in 26 kW of net power and a value of heat transfer area of 4.5 m^2.

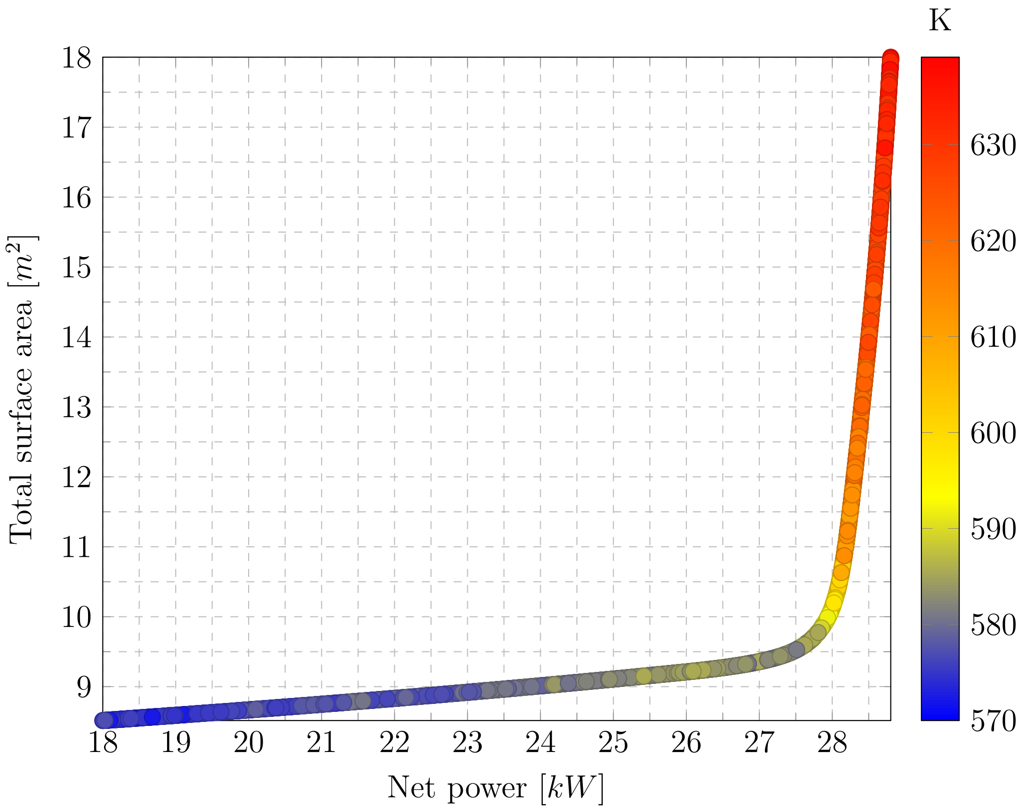


Figure 4.5: Secondary heater

4.1.3 Mass flow rate effect

Variation of mass flow is inversely proportional to both heat transfer surface area and net power (Figure 4.6). From the heat source it can be stated that if mass flow increases, enthalpy difference must decrease and therefore flow temperature (heat input is constant). Then, if at the turbine inlet, the fluid has a lower enthalpy value, outlet enthalpy must be lower as well. One could think that lower enthalpy difference compensates the decrease in mass flow making turbine work output remain constant. However, this is not the case since the increase in mass flow has more influence. Same reasoning can be applied to the compressor.

$$W_t = \dot{m} \cdot (h_5 - h_6) \tag{4.1}$$

In terms of heat transfer surface area, lower mass flow rate values causes a higher outlet temperature at the cold side of any of the heat sources. Taking a look at equation (4.2), it is deducted that logarithmic mean temperature difference is inversely proportional to cold side outlet temperature which, from equation (4.3) makes surface area to decrease since the overall heat transfer coefficient remains almost constant.

$$\Delta_{LMTD} = \frac{\Delta_{T1} - \Delta_{T2}}{\ln \frac{\Delta_{T1}}{\Delta_{T2}}} \tag{4.2}$$

where $\Delta_{T1} = T_{h,i} - T_{c,o}$ and $\Delta_{T2} = T_{h,o} - T_{c,i}$.

$$Q = UA_{ref} \cdot \Delta_{LMTD} \tag{4.3}$$

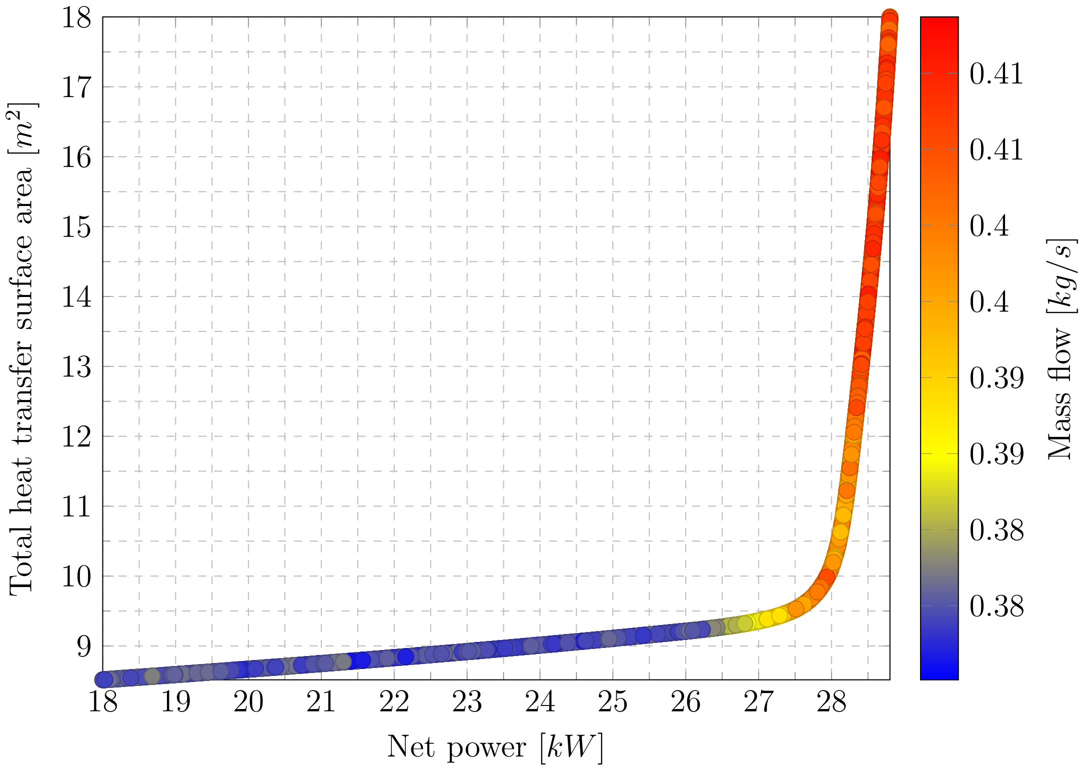


Figure 4.6: Mass flow effect

4.1.4 Heat transfer surface area

Heat transfer surface area is a dependent variable which is directly proportional to capital costs since heat exchangers are the most expensive components among the equipment. Therefore, this parameter should be carefully optimized. Figure 4.7 and 4.8 shows the reduced Pareto distribution for different transfer surface areas in both main and secondary heater which behave in a similar way.

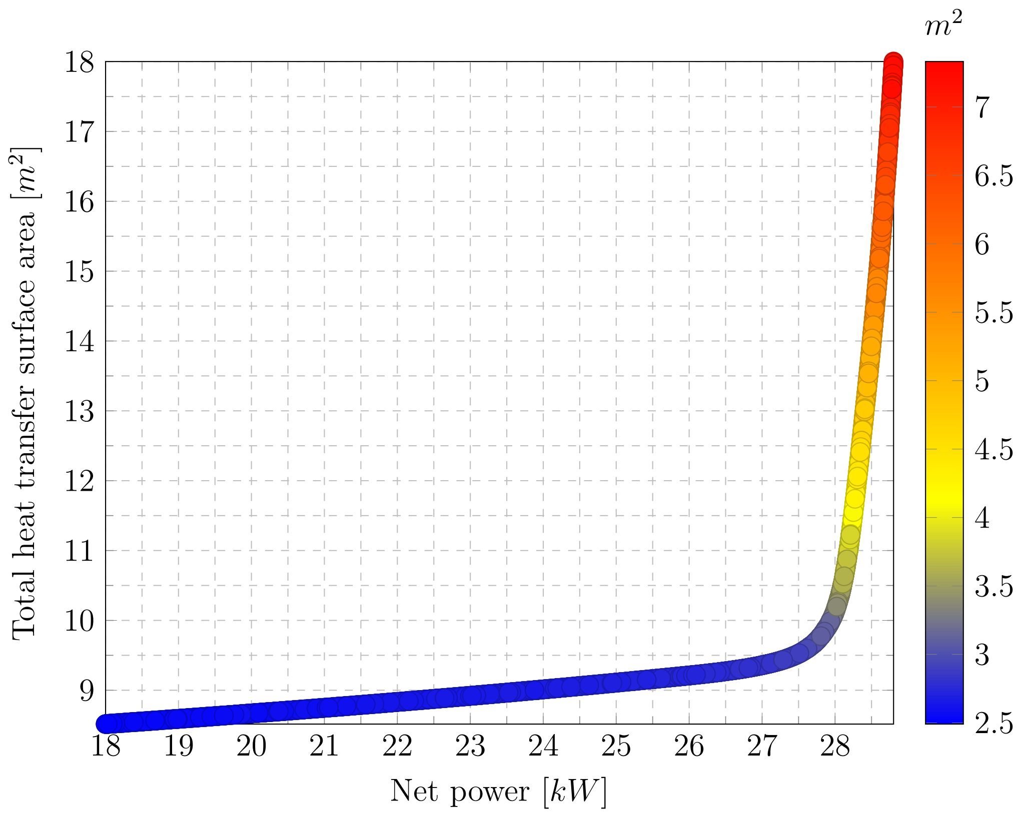


Figure 4.7: Main heater

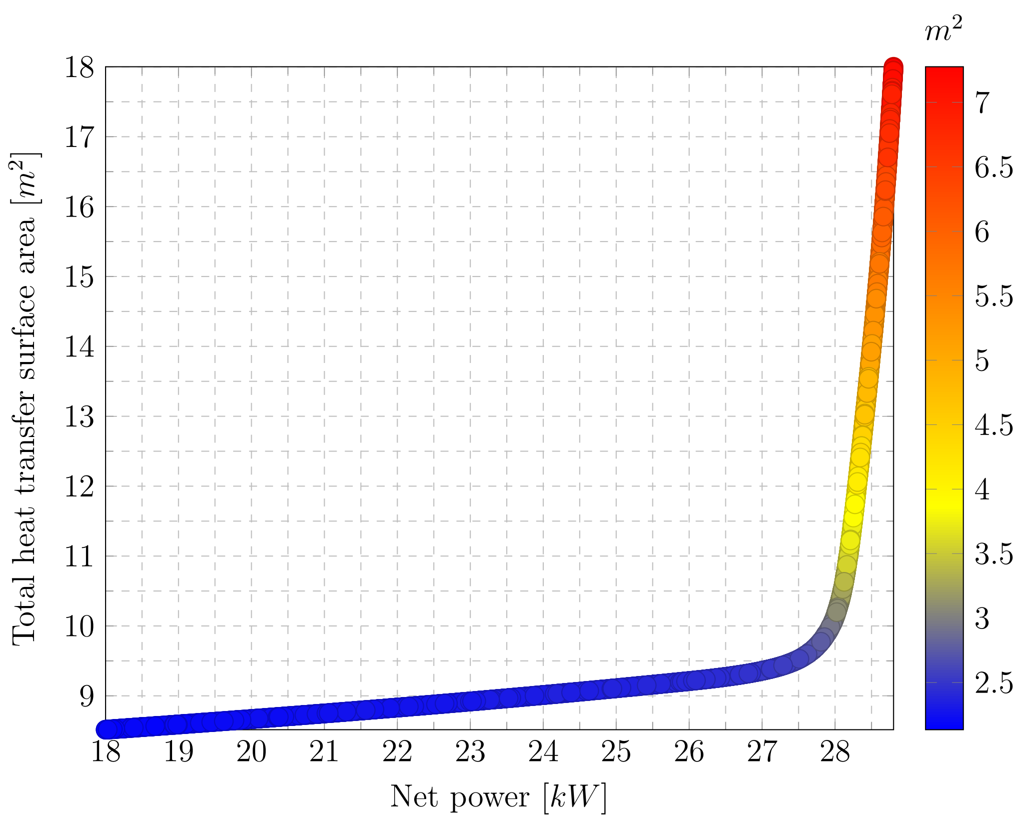


Figure 4.8: Secondary heater

Looking at the graph it is clear how until reaching 27 kW of net power, heater surface area remains around 3 and after this it drastically increases m^2. There are two reasons for this behaviour, one is the variation of cold side outlet temperature and thus, mass flow rate. Comparing the Figures 4.4 and 4.5 it can be stated that above 780K and 590K in the main and secondary heater, respectively, the heat transfer surface area increase in both heat is huge.

On the other hand, high pressure also plays an important role in the used surface area of the heaters since higher values causes an increase in heater cold side inlet temperature which translates into more surface area required. Taking a glance at figure 4.3a high pressure values above 210 bar have a drastic increase in used surface area in both heaters. Thus, an optimum value for both heat exchangers would be around 3 m^2 where net power reaches 27-28 kW.

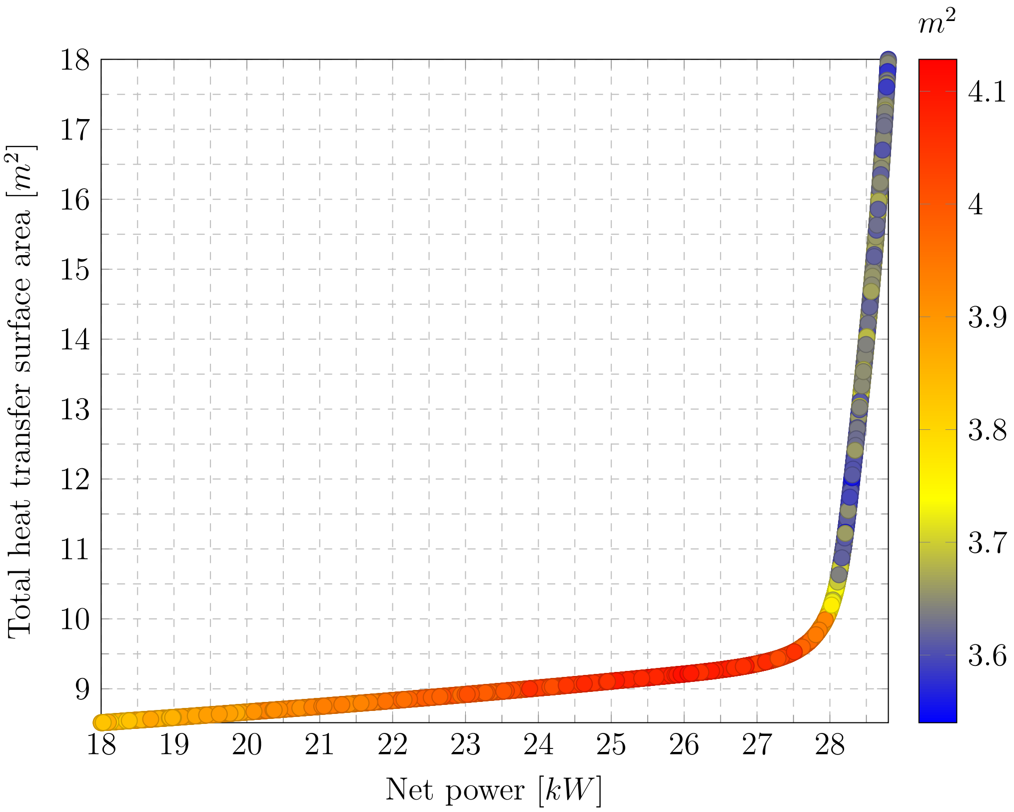


Figure 4.9: Cooler heat transfer surface area effect

However, there is another heat exchanger in the system which is often neglected from the analysis, the cooler. The cooler is mainly affected by the lower pressure parameter, specially when working near the super-critical point. In Figure 4.9 this abnormal behaviour is seen which is caused by the variation of the properties of the carbon dioxide along with its temperature. It is important to denote that because the cooler de-heating capacity is limited by the coolant fluid, in this case air, since 25ºC of inlet temperature is not enough to cool down the fluid to critical point at reasonable heat transfer surface areas values.

4.2 Economical analysis

Although nowadays studies have focused more on on the performance of sCO_2 power cycles rather than in economic analysis, an estimation of the overall cost of the system has been evaluated. This study is based on the paper [8] where a compilation of component vendors quotes across different U.S. Department of Energy National Laboratories was used to obtain cost scaling relationships for almost every sCO_2 power cycle component.

However, this study aimed at high power cycles like nuclear or fossil electrical energy generation applications which have a higher net power values compared to a wasted-heat application. For this reason, the results from the basic analysis should be taken with care. From figure 4.10 a total amount of 400.000 € should be expected for the whole setup, including heat exchangers, compressor, turbine and generators.

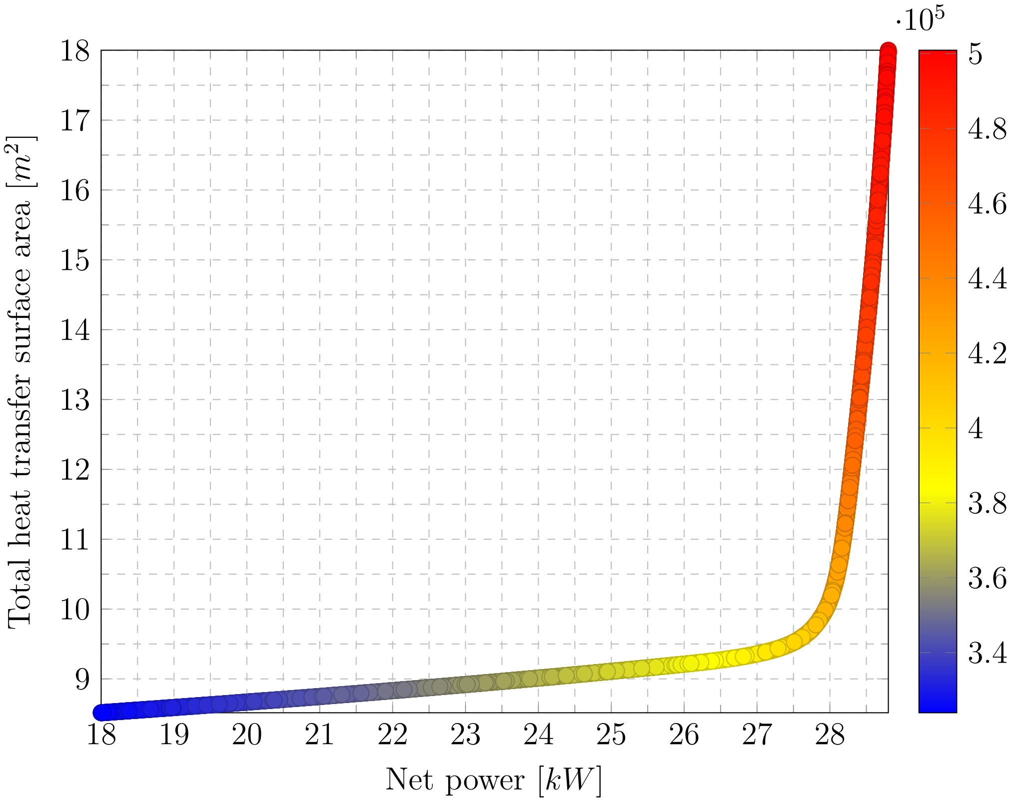


Figure 4.10: Capital costs

4.3 Model validation

Model validation is a vital part of every problem like this. Without it, it is not possible for the readers to trust the results from the study and therefore it will have no impact on the scientific field. For this reason, one set of conditions will be compared for OptDesign and EbsilonProfessional models. Further comparisons of different set of conditions will be attached to the appendix.

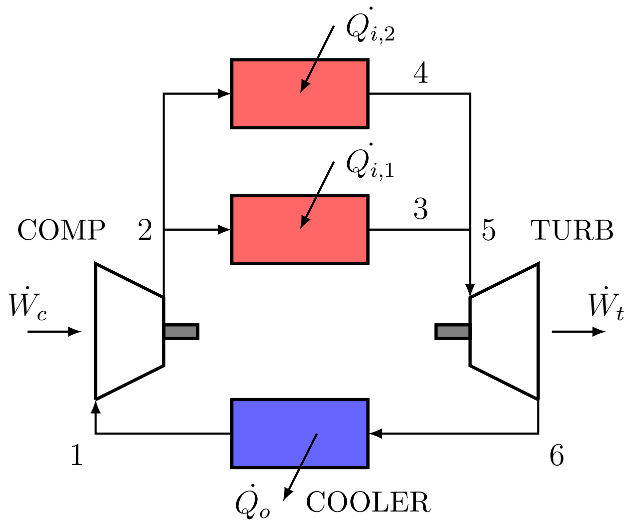


Figure 4.11: Layout

P_2	P_6	T_5	T_6	T_1
174.610 Bar	85.019 Bar	759.69 K	577.14 K	308.15 K

Table 4.1: Validation point conditions

The randomly chosen set of conditions are shown in Table 4.1 which are referenced to the layout from Figure 4.11. Most values are reasonably low, below 3% of relative error, however the highest error resides in the overall heat transfer coefficient from the cooler, although still assumable it could be due to the different properties database used by both models. Overall, these are good results from which it can be assured that OptDesign model is valid for the study.

Parameter	Unit	OptDesign	Ebsilon	Abs. error	Rel. error [%]
Wc	kW	6.66	6.62	0.038	0.57
Wt	kW	27.90	27.76	0.140	0.50
Wn	kW	20.2281	20.13	0.097	0.48
T2	K	333.06	333.06	0.000	0.00
T5	K	674.86	675.23	0.371	0.05
T6	K	599.63	599.98	0.350	0.06
UAhx1	W/m^2K	0.82	0.82	0.002	0.19
UAhx2	W/m^2K	0.61	0.60	0.006	1.01
UAcool	W/m^2K	2.12	2.08	0.039	1.87

Table 4.2: Results comparison for conditions given in Table 4.1

Chapter 5

Summary and conclusion

This research aimed to identify the best solution to de-heat an external flow made up by a mixture of hydrogen and water. Firstly, an inert gas based gas turbine was proposed but soon it was discovered that compressor outlet temperature was to high to cool down the heat input flow. Then, the study focused on a sCO_2 gas turbine which had far more advantages, like compactness, simplicity and efficiency when used at super-critical conditions.

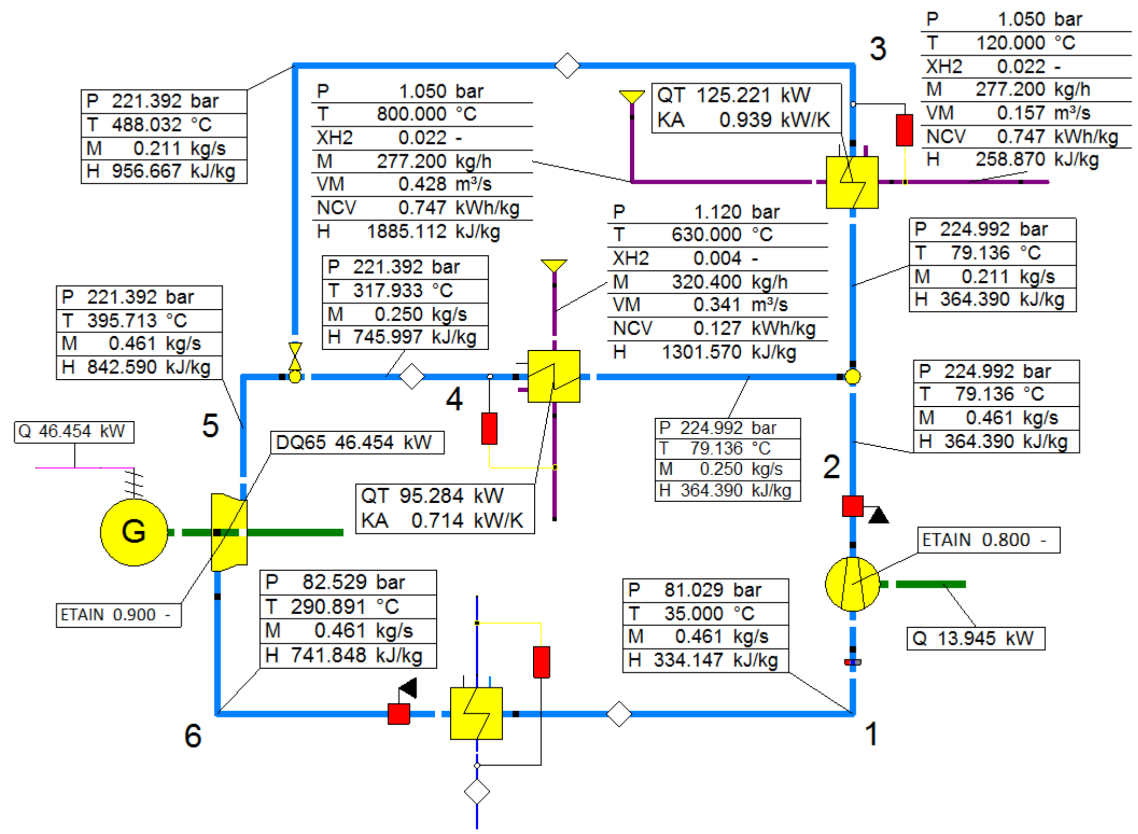


Figure 5.1: Circuit diagram at optimized conditions

Parameter	Unit	Value
Higher pressure, P_{max}	[Bar]	224.991
Lower pressure, P_{min}	[Bar]	82.529
Mass flow rate, $\dot{m}$	[kg/s]	0.399
Main H.X. temperature, T_1	[K]	761.182
Secondary H.X. temperature, T_2	[K]	591.083
Main H.X. surface area, A_1	$[m^2]$	3.040
Secondary H.X. surface area, A_2	$[m^2]$	2.818
Cooler surface area, A_{cooler}	$[m^2]$	3.925
Total surface area, A_t	$[m^2]$	9.782
Compressor work, W_c	[kW]	12.194
Turbine work, W_t	[kW]	40.976
Air fan work, W_{fan}	[kW]	0.970
Net power, W_n	[kW]	27.812
Estimation cost, C_T	[€]	410001.54

Table 5.1: Optimization results

In the cycle layout discussion chapter, parallel configuration proved to be the best performing option thanks to its simplicity. Serial configuration, was impossible to be carried out because of the temperature boundary condition at the hot side outlet of both heat sources. Same reason goes for the recuperated cycle, it was not feasible since the potential recuperating temperature range was too low because of the hot side outlet temperature limitation (120ºC) at the heat sources. The other alternative, the reheated cycle, was not feasible either as the low improvement in net power was offset by the capital cost of the extra components. Other alternative layout were not considered because the complexity and the cost increase would have offset the improvement in performance.

Once the parallel layout was selected, the model was optimized looking for the best net power output while maintaining low heat transfer surface area values for both heat sources and the cooler (Figure 5.1). High and low pressure values had a significant effect in net output, first one because of the increase in pressure ratio and therefore, higher enthalpy drop in the turbine although the maximum value for higher pressure was not to high (225 bar) since it ended in higher compressor outlet temperature which increased the required heater surface effectiveness and therefore their area. On the other hand, lower pressure played an important role as well due to the rapid variation of properties near the critical point and the fixed cooling water temperature which was 4ºC higher than the supercritical temperature of sCO_2.

In regard to heater outlet temperatures, it had influence on net power because of the higher turbine inlet temperature, though it was offset by the consequent drop in mass flow, and on heat transfer surface area of the heaters since the required heat exchanger effectiveness increased as well.

An enthalpy-entropy diagram is shown in Figure 5.2 which stands for a standard Brayton cycle except from the points 3 and 4. These conditions represent the ideal pressure and temperature mixture of the two heater outlets at point 5. Results from the analysis which are described in Table 5.1 where overall cost estimation of the gas turbine is around 400.000 € for a net power output of 27.81 kW. Although this may seem expensive for such a low value of power output, it's important to note that the cost estimation is based in high power level applications, so the value may be overestimated.

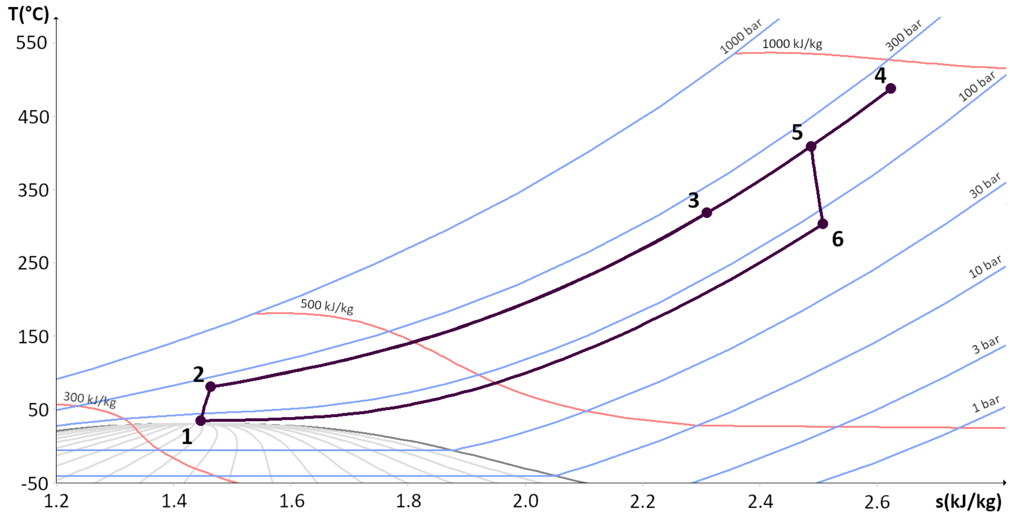


Figure 5.2: Enthalpy-entropy diagram

List of Figures

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