Exergoeconomic analyses of different sCO_2 cycle configurations

Mathias Penkuhn
Research Assistant
Technische Universität Berlin
Berlin, Germany
mathias.penkuhn@iet.tu-berlin.de

George Tsatsaronis
Professor
Technische Universität Berlin
Berlin, Germany
tsatsaronis@iet.tu-berlin.de

Mathias Penkuhn is a research assistant at the Department of Energy Engineering (group of Prof. Tsatsaronis) at Technische Universität Berlin, Germany. He received a diploma in process engineering from Technische Universität Dresden, Germany. His areas of interest include the design, analysis and optimization of energy conversion systems, and the application and further development of exergetic and exergoeconomic methods.

George Tsatsaronis is since 1994 the Bewag Professor of Energy Engineering and Environmental Protection at the Technische Universität Berlin, Germany. His areas of interest include the design, development, analysis and optimization of energy conversion systems. He contributed significantly to the fundamentals and terminology of exergy-based methods. He co-authored the book “Thermal Design and Optimisation” (Wiley, 1996), has published more than 400 papers and reports, received several international awards and recognitions, and has served as Chairman or Co-chairman of 20 international conferences.

Abstract

Supercritical CO_2 (sCO_2) power cycles have received widespread interest during the last years for future applications in fossil-fuel, nuclear, and renewable energy-based power generation. Compared to conventional water and steam based power cycles, the sCO_2-based power cycles have been found to provide numerous advantages, in particular, concerning their economic viability due to higher efficiencies and smaller equipment sizes. Many different configurations have been proposed, analyzed, optimized, and reviewed regarding their thermodynamic efficiency. However, it is currently not clear which of these configurations are the most promising ones concerning their economic feasibility. Furthermore, the strong interrelation of economic and thermodynamic efficiencies has to be investigated at both, the overall system and at the component level. These issues have to be addressed thoroughly in order to further develop the most promising cycle configurations. In the present analyses, the concept of exergoeconomic analysis, being a combination of thermodynamic and economic analyses, is used to provide thorough information on the different sCO_2 cycle configurations for comparisons at the system and the component level. Based on the results of the exergoeconomic analyses the most important aspects for further research and development are identified.

1 Introduction

The development of new technologies for the efficient conversion and use of energy is a major challenge considering an increasing demand and change in primary energy resources within the next decades [1]. In particular, the power generation sector is subject to a diversification with the continuing use of fossil-fuel and nuclear-based technologies as well as a widespread implementation of renewables. With an increased economic competition and with the environmental protection regulations becoming stricter, new highly-efficient, flexible and economically competitive processes for energy conversion are required [2, 3], especially for power generation.

Within the field of emerging power generation technologies, the application of supercritical CO_2 (sCO_2) as a working fluid in direct (open) and indirect (closed) thermodynamic cycles [4, 5] has received increased attention during the last years. Based on the work of Gokhshtein and Dekhtyarev [6–8], Angelino [9, 10], Feher [11, 12], and Schabert [13], the interest in sCO_2 for power generation applications was renewed by the works of Dostal [14–17] in the beginning of the new century. In the last years, sCO_2-based power cycles have been studied for future application in direct and indirect cycles[18–21] in fossil-fuel [22–24], nuclear [25, 26] and concentrating solar [27–30] power generation as well as for waste heat recovery [31–33]. The advantages of sCO_2-based power cycles include, e.g., higher efficiencies and increased higher flexibility compared to conventional working fluids, lower capital cost, and smaller plant footprints [4].

Although significant progress has been made in the design of sCO_2 power cycles and their constituting components, the question of economic viability remains to be addressed. Only limited data [14, 34–36] is available concerning economic analyses of sCO_2-based cycles. In particular, the question of economic feasibility must address aspects for further research and development with the long-term goal of successful commercialization. Therefore, the thermodynamic and economic performance of the overall system and its constituent components plays an important role because of their interdependencies. Consequently, the design of the system needs to be analyzed in order for it to be improved and to make it cost-efficient.

In this context, exergy and exergoeconomic analyses [37, 38] provide an objective means for the evaluation of the thermodynamic and economic performance of a system. Although, high system efficiencies are desirable from a thermodynamic point of view, only cost-efficient systems are going to be commercially successful. Based on the integrated exergetic and economic (exergoeconomic) approach, the present study is investigating important aspects in the design of indirect sCO_2-based power cycles.

2 System Descriptions

The present study concentrates on a selection of well-known basic configurations. Furthermore, for the subsequent analysis, the simulation parameters that were used are presented and discussed.

2.1 Cycle Configurations

Within the literature a great variety of different cycle layouts for different applications [20, 26] can be found. However, the most common cycle features derive from the works of Gokhshtein and Dekhtyarev [6–8], Angelino [9, 10], and Feher [11, 12]. Due to the thermodynamic properties of CO_2 [39] – different from water-steam (H_2O) – it is generally acknowledged that the reference configuration of a sCO_2 power cycle consists of a single train of compressor, turbine, heat exchangers for heat supply and removal, and a recuperator. Heat recuperation within the cycle is possible because of the low pressure ratio of a sCO_2 cycle, with a minimum pressure of 73.77 bar (critical pressure) and turbine outlet temperatures being still high and far away from the critical temperature of 30.98 °C.

The reference configuration is shown in Figure 1a and shows a single flow cycle with simple recuperation. However, in order to improve the cycle efficiency, different possibilities have been proposed that incorporate intercooling or reheating (Figure 1b) options in order to either minimize

Table 1: Environment conditions used for the simulations based on [41]. Ambient pressure and dry bulb temperature are used as the thermodynamic reference environment for exergy calculation.

Site Conditions	
Model	Midwest ISO
Ambient Pressure	1.01325 bar
Ambient Dry Bulb Temperature	15.0 °C
Ambient Wet Bulb Temperature	10.8 °C
Relative Humidity	60 %
Cooling Water Temperature	15.6 °C

the compression work and maximize the expansion work, respectively. Regarding the improvement of single flow cycles, other possibilities include precompression, interrecuperation, and split expansion [26]. A common disadvantage of single flow cycles is a temperature limitation within the recuperator (E-2) because of a significant difference in thermal properties (heat capacity rate) of CO_2 between the hot low-pressure and the cold high-pressure side.

To better match the difference in heat capacity rates between the hot and cold side of the recuperator, split flow cycles [14, 26] have been proposed to further improve the cycle efficiency. Common layouts for split flow cycles are the recompression (Figure 1c) and the modified recompression cycle (Figure 1d). Other options comprise preheating and splitting of the turbine flow [26].

Considering the scope of the previous compilation of possible cycle layouts, the four configurations shown in Figure 1 were chosen for further investigation because of their high thermodynamic efficiency according to available studies [20, 26].

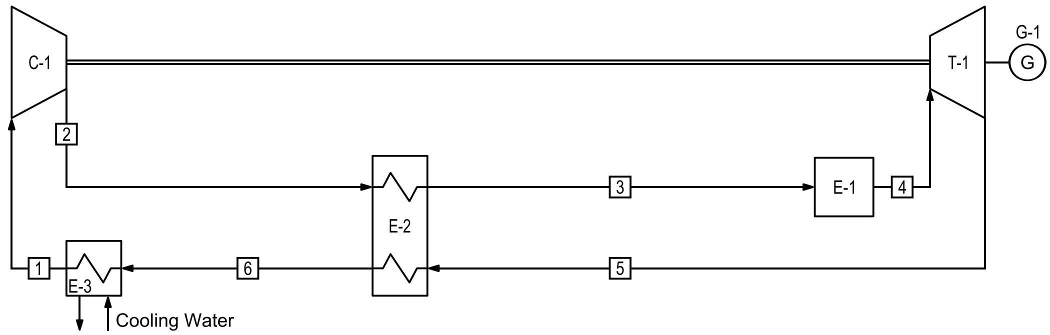
2.2 Cycle Simulation and Parameterization

When analyzing different processes in order to obtain comparable results for benchmarking [40], it is necessary to use best practice guidelines [41, 42], representing proper assumptions, in addition to potentially available process data.

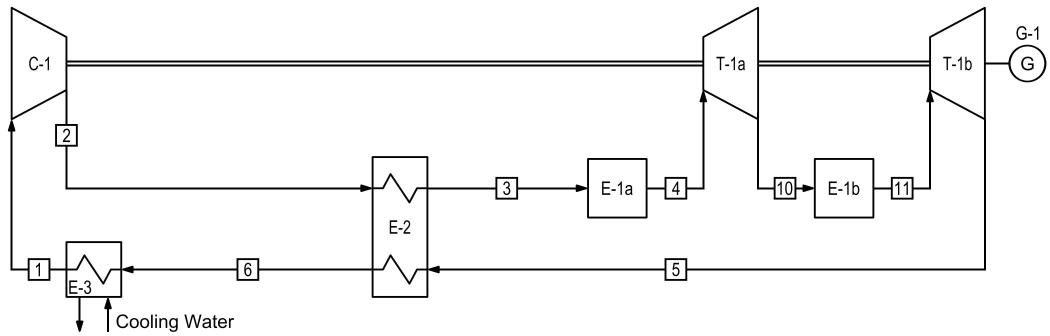
The simulated process configurations of the different cycles are based on Figure 1. These cycles have been implemented in AspenPlus using the REFPROP [43] property method that uses the Span-Wagner equation of state [39] for CO_2.

In order to obtain comparable results from the simulation model, a parameterization according to proper definitions of the technology level and the operating conditions of the process [41, 42] is required. Therefore, the environmental conditions are specified according to [41] and are presented in Table 1. The same conditions are also used for the calculation of the exergy of each process stream via embedded FORTRAN subroutines.

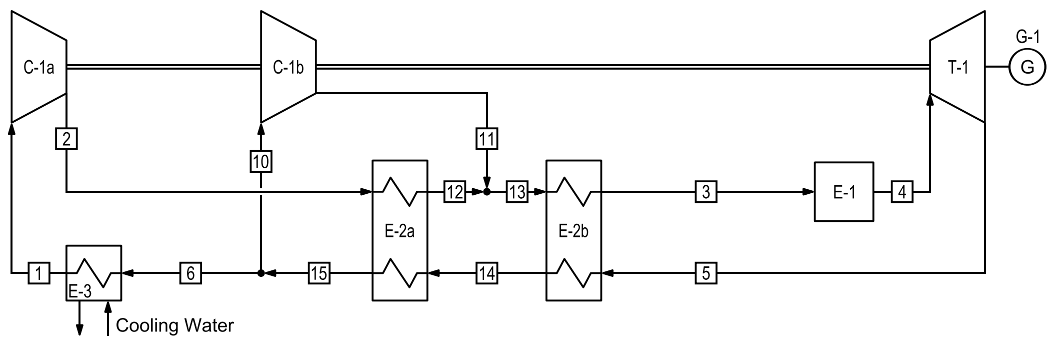
The parameters that are used for modeling the closed sCO_2 cycles have been chosen according to the discussions and data referenced in [41, 42] and are shown in Table 2. Thus, the main characteristics for comparing the different cycles are a temperature of 35 °C and a pressure of 75 bar at the compressor inlet. Moreover, at the turbine inlet the temperature is chosen to be 600 °C and the pressure to 250 bar. The simulations are conducted using a generic heat source to obtain the different cycle parameters that are subsequently used for the evaluation.



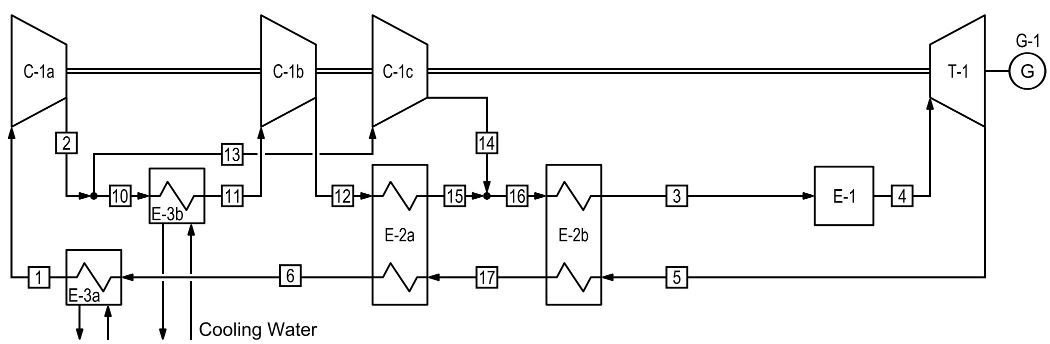
(a) Recuperated sCO_2 cycle



(b) Reheating, recuperated sCO_2 cycle



(c) Recompression, recuperated sCO_2 cycle



(d) Modified recompression, recuperated sCO_2 cycle

Figure 1: Flowsheets of the simple sCO_2 cycle configurations analyzed in this study

Table 2: Simulation parameters for the analysis of the sCO_2 cycles [41, 42]

Parameter	Value
Turbine Inlet Temperature	600 °C
Turbine Inlet Pressure	250 °C
Turbine Isentropic Efficiency	90 %
Turbine Mechanical Efficiency	99 %
Compressor Inlet Temperature	35 °C
Compressor Inlet Pressure	75 bar
Compressor Isentropic Efficiency	85 %
Compressor Mechanical Efficiency	98 %
Compressor Motor Efficiency	97 %
Electric Generator Efficiency	99 %
Primary Heat Exchanger Pressure Drop	700 kPa
Primary Heat Exchanger Temperature Difference	25 K
Recuperator Minimum Temperature Difference	5 K
Recuperator Hot-Side Pressure Drop	280 kPa
Recuperator Cold-Side Pressure Drop	140 kPa
Cooler Pressure Drop	15 kPa
Cooling Water Temperature Difference (Range)	11 K

3 Methodology

The objective of the present study is to obtain information on the performance of the different cycles regarding their thermodynamic and economic efficiency. Therefore, conventional thermodynamic and economic as well as detailed exergetic and exergoeconomic analyses are employed.

3.1 Thermodynamic Analysis

Using the AspenPlus modeling and simulation environment, the mass and energy balances for the different sCO_2 cycles are solved. Following a generally applied convention for power cycles [44], the overall efficiency is calculated as the ratio of net power obtained and heat supplied to the cycle:

$$\eta = \frac{\dot{W}_{\mathrm{Net}}}{\dot{Q}_{\mathrm{Supply}}} \tag{1}$$

However, when comparing different cycles, the selected benchmarking framework has to be taken into account. Furthermore, the overall efficiency definition based on energy-related measures only provides an incomplete set of information compared to an exergy analysis.

3.2 Exergy Analysis

By providing information that is not available by conventional thermodynamic analyses, an exergy analysis provides useful means to uncover the real thermodynamic inefficiencies within a system and its constituting components with respect to its thermodynamic environment.

In contrast to energy, representing the quantity *and* quality of energy, exergy is destroyed within each system component thereby influencing the overall system performance. As the methodology and capabilities of a conventional exergy analysis are well established and understood [37, 45–47], the parameters derived from such an analysis can be used to characterize the thermodynamic performance of a system and its different components from an unbiased point of view.

To provide a set of detailed information, the exergy, accounted by the physical exergy (PH), is split into its thermal (T) and mechanical (M) parts [47] that are related to changes in temperature and pressure, respectively.

$$\dot{E} = \dot{E}^{\mathrm{PH}} = \dot{E}^{\mathrm{T}} + \dot{E}^{\mathrm{M}} \tag{2}$$

When analyzing the process at steady-state conditions, the exergy balance for each component k can be used to calculate its exergy destruction $\dot{E}_{\mathrm{D},k}$. The exergy balance contains different terms for the transport of heat $\dot{E}_{\mathrm{q}}$ and power $\dot{W}$ as well as the transport of mass at the inlet $\dot{E}_i$ and outlet $\dot{E}_e$ of the each component k.

$$\dot{E}_{\mathrm{D},k} = \sum_j \dot{E}_{\mathrm{q},j,k} + \dot{W}_k + \sum_i \dot{E}_{i,k} - \sum_e \dot{E}_{e,k} \tag{3}$$

With the calculation of the exergy destruction $\dot{E}_{\mathrm{D},k}$ further parameters can be derived that are used to unambiguously characterize the thermodynamic performance of a component k.

Considering the conversion of energy and exergy within a component k, terms for the exergy rates of fuel $\dot{E}_{\mathrm{F}}$ and product $\dot{E}_{\mathrm{P}}$ can be identified, which allows Eq. (3) to be rewritten as:

$$\dot{E}_{\mathrm{D},k} = \dot{E}_{\mathrm{F},k} - \dot{E}_{\mathrm{P},k} \tag{4}$$

This information can be used to determine the real thermodynamic performance of a system's component k.

$$\varepsilon_k = \frac{\dot{E}_{\mathrm{P},k}}{\dot{E}_{\mathrm{F},k}} = 1 - \frac{\dot{E}_{\mathrm{D},k}}{\dot{E}_{\mathrm{F},k}} \tag{5}$$

For the overall system an additional term $\dot{E}_{\mathrm{L,Tot}}$ incorporating losses to the environment has to be considered.

$$\varepsilon_{\mathrm{Tot}} = \frac{\dot{E}_{\mathrm{P,Tot}}}{\dot{E}_{\mathrm{F,Tot}}} = 1 - \frac{\sum_k \dot{E}_{\mathrm{D,k}} + \dot{E}_{\mathrm{L,Tot}}}{\dot{E}_{\mathrm{F,Tot}}} \tag{6}$$

As the assignment of exergy rates of fuel and product is not always straightforward, the SPECO methodology [48] can be employed. It provides a consistent framework that assists in defining the real thermodynamic efficiencies of a component and the overall system.

Another useful exergy-based parameter is the exergy destruction ratio $y_{\mathrm{D},k}$ that quantifies the contribution of each component's exergy destruction to the reduction of the overall system's exergetic efficiency $\varepsilon_{\mathrm{Tot}}$.

$$y_{\mathrm{D},k} = \frac{\dot{E}_{\mathrm{D},k}}{\dot{E}_{\mathrm{F,Tot}}} \tag{7}$$

The different exergy-based parameters can be used to analyze a system from an unbiased point of view and obtain possible measures for a subsequent improvement. By further combining an exergy analysis with an economic analysis, the resulting exergoeconomic analysis provides information about the cost formation process within a system.

Table 3: Assumptions of the economic analysis

Parameter	Value
System economic life (years)	20
System availability (hours per year)	8000
Average rate of return, interest (%/year)	8
Escalation rate (%/year)	0

3.3 Economic Analysis

The economic analysis is conducted considering the guidelines of the total revenue requirement (TRR) method [37]. For capital cost estimation of the cycle equipment, the information given by [36] is used under the assumption that the calculated baseline costs represent the total plant investment for each cycle.

For the sake of simplicity, inflation is neglected for this scoping study. Furthermore, the cost of the generic heat supplied to the cycle is assumed to be zero for the detailed analysis. This assumption enables the analysis of the different cycle configurations without possible interactions that have to be attributed to upstream processes providing a baseline for each cycle. However, the influence of the cost of the heat supply is subsequently assessed with a sensitivity analysis. The complete set of economic assumptions that were used for the analysis is given in Table 3.

3.4 Exergoeconomic Analysis

By combining both, the results of the exergetic and economic analyses, an exergoeconomic analysis provides additional information by revealing the cost formation process within the system. The information from such an analysis can used for a subsequent improvement and optimization of the system [38].

Based on the exergoeconomic methodology, specific costs c_i are assigned to each stream of exergy that occurs within the system.

$$\dot{C} = c \cdot \dot{E} \tag{8}$$

By using cost balances for each system component k, it is possible to calculate the costs of each stream considering costs rates related to exergy transfer and monetary expenditures.

$$\sum_i \left(c_i \dot{E}_{i,k} \right) + \dot{Z}_k = \sum_e \left(c_e \dot{E}_{e,k} \right) \tag{9}$$

The specific costs per exergy unit c_i and c_e are associated with the inlet and outlet exergy streams of a component k. The associated monetary expenditures (e.g., capital investment and operation and maintenance costs) of a component are represented by $\dot{Z}_k$. Furthermore, each stream at the inlet and outlet is also related to the definitions of each components exergy rate of fuel and product.

The most appealing aspect of exergy costing is the determination of the cost associated with exergy destruction $\dot{C}_{\mathrm{D},k}$ of each component of the system, representing the cost of additional fuel that has to be supplied to generate the same exergy rate of product.

$$\dot{C}_{\mathrm{D},k} = c_{\mathrm{F},k} \dot{E}_{\mathrm{D},k} \qquad \text{with} \qquad \dot{E}_{\mathrm{P},k} = \text{constant} \tag{10}$$

Based on these considerations, additional parameters can be obtained from an exergoeconomic analysis. At first, a comparison of the specific average exergy costs of product and fuel provides the relative cost difference r_k, revealing the importance of the different costs.

$$r_k = \frac{c_{\mathrm{P},k} - c_{\mathrm{F},k}}{c_{\mathrm{F},k}} \tag{11}$$

Another useful parameter is the exergoeconomic factor f_k that gives a relative measure of the importance of monetary and exergy destruction costs for the operation of a component k.

$$f_k = \frac{\dot{Z}_k}{\dot{Z}_k + \dot{C}_{\mathrm{D},k}} \tag{12}$$

The exergoeconomic parameters can be used to evaluate and further improve an energy conversion system, especially regarding its overall cost efficiency.

4 Results and Discussion

Based on the process models and their parameterizations, different simulations were conducted under the assumption of a net power output of each cycle of 100 MW. Based on the simulation results, the previously mentioned analyses are conducted and provide a starting point for further discussion.

4.1 Simulation Results

The results of the sCO_2 cycle simulations are shown in Table 4. The simple recuperation cycle (a) has the lowest thermal efficiency of 37.85 %. By employing different improvement options, the reheating cycle (b) exhibits a thermal efficiency of 38.31 % and the well-known recompression (c) cycle has a thermal efficiency of 40.61 %. Furthermore, the modified recompression cycle (d) has the highest thermal efficiency of 42.58 % of the cycles in this study.

Taking into account the given parameterization of the cycles. The main reason for the high efficiency of the modified recompression cycle can be found in the stream property results in Table 5.

Operating between the specified temperatures and pressures, the different cycle configurations exhibit similar temperatures and pressure. However, the modified recompression cycle (d) has a much lower outlet temperature on the recuperator's (E-2) hot side. Furthermore, the required compression work is much lower compared to the other cycles. In contrast, the reheating option (b) only provides a small improvement. Moreover, the high temperature split flow recompression of the recompression cycle (c) requires a large amount of additional compression work.

Another interesting feature can be found in the temperature profiles of the recuperator (E-2) of the modified recompression cycles (d) compared to the other cycles. Comparing the inlet, interme-

Table 4: Thermal efficiencies of the analyzed cycle configurations

ID	Cycle Configuration	Thermal Efficiency (%)
a	Simple Recuperation Cycle	37.85
b	Reheating Cycle	38.31
c	Recompression Cycle	40.61
d	Modified Recompression Cycle	42.58

Table 5: Stream property results of the different cycle configurations

(a) Recuperated sCO_2 cycle

No.	t (°C)	p (bar)	e^T (kJ/kg)	e^M (kJ/kg)
1	35.00	75.00	7.59	198.46
2	123.23	258.40	34.84	218.10
3	385.88	257.00	196.84	217.95
4	600.00	250.00	362.95	217.24
5	457.14	77.95	213.96	198.80
6	128.23	75.15	33.55	198.48

(b) Reheating sCO_2 cycle

No.	t (°C)	p (bar)	e^T (kJ/kg)	e^M (kJ/kg)
1	35.00	75.00	206.05	7.59
2	123.23	258.40	252.94	34.84
3	440.16	257.00	454.15	236.20
4	600.00	250.00	580.19	362.95
5	514.44	77.95	454.86	256.06
6	128.23	75.15	232.03	33.55
10	543.67	161.82	513.11	305.04
11	600.00	154.82	556.74	349.42

(c) Recompression sCO_2 cycle

No.	t (°C)	p (bar)	e^T (kJ/kg)	e^M (kJ/kg)
1	35.00	75.00	7.59	198.46
2	123.26	258.40	34.86	218.09
3	433.63	257.00	231.28	217.95
4	600.00	250.00	362.76	217.23
5	456.95	77.95	213.80	198.80
6	128.26	75.15	33.56	198.48
10	128.26	75.15	33.56	198.48
11	264.11	257.51	116.43	218.00
12	264.11	257.51	116.43	218.00
13	264.11	257.51	116.43	218.00
14	269.11	76.94	95.93	198.68
15	128.26	75.15	33.56	198.48

(d) Modified recompression sCO_2 cycle

No.	t (°C)	p (bar)	e^T (kJ/kg)	e^M (kJ/kg)
1	35.00	75.00	7.59	198.46
2	48.08	90.71	11.27	200.26
3	386.74	257.00	197.40	217.95
4	600.00	250.00	362.95	217.24
5	457.14	77.95	213.96	198.80
6	70.26	75.15	16.49	198.48
10	48.08	90.71	11.27	200.26
11	35.00	90.56	2.48	200.25
12	65.26	258.40	8.19	218.10
13	48.08	90.71	11.27	200.26
14	123.16	257.70	34.84	218.02
15	123.16	257.70	34.84	218.02
16	123.16	257.70	34.84	218.02
17	128.19	76.55	34.01	198.64

diate, and outlet temperatures, it becomes clear, that the pinch point temperature is significantly lower, thus allowing more heat to be recovered.

In general, based on the thermodynamic simulation results, the modified recompression cycle (d) is an interesting cycle configuration, that has not received widespread interest compared to the recompression cycle (c). In order to obtain further information for the different cycles, an exergy analysis is used to compare the features of the different cycles at the component level.

4.2 Exergy Analysis

The exergy analysis is conducted using the information obtained by simulation of the different cycles. The results are given in Table 6 and show the main inefficiencies of the different cycle configurations.

Comparing the results of the different cycles. It is clear that the efficiencies of all components are comparably high. Based on the figures of Table 6, it becomes clear that the main inefficiencies are related to the heat transfer in the recuperator (E-2) and the precooler (E-3), with compressor (C-1) and turbine (T-1) having a much lower impact on cycle efficiency. This clearly shows the high importance of the heat recovery due to the very high turbine outlet temperature. This changes sig-

Table 6: Results of the exergy analysis for the different cycle configurations with a cycle output of 100 MW.

(a) Recuperated sCO_2 cycle

ID	$\dot{E}_F$ (MW)	$\dot{E}_P$ (MW)	$\dot{E}_D$ (MW)	ε (%)	y_D (%)
C-1	54.5	46.1	8.4	84.6	4.7
E-1	166.6	163.3	3.3	98.0	1.8
E-2	178.4	159.3	18.6	89.3	10.3
E-3	25.6	–	24.0	–	13.3
T-1	164.7	154.5	10.1	93.8	5.6
Total	180.1	100.0	75.4	55.5	41.9

(b) Reheating sCO_2 cycle

ID	$\dot{E}_F$ (MW)	$\dot{E}_P$ (MW)	$\dot{E}_D$ (MW)	ε (%)	y_D (%)
C-1	53.4	45.2	8.2	84.6	4.6
E-1	168.0	165.0	3.1	98.2	1.7
E-1A	124.4	122.2	2.3	98.2	1.3
E-1B	43.6	42.8	0.8	98.2	0.5
E-2	215.5	194.1	20.8	90.1	11.7
E-3	25.0	–	23.5	–	13.2
T-1	162.9	153.4	9.4	94.2	5.3
T-1A	64.7	61.0	3.7	94.3	2.1
T-1B	98.2	92.5	5.7	94.2	3.2
Total	177.9	100.0	73.3	56.2	41.2

(c) Recompression sCO_2 cycle

ID	$\dot{E}_F$ (MW)	$\dot{E}_P$ (MW)	$\dot{E}_D$ (MW)	ε (%)	y_D (%)
C-1	85.3	73.1	12.1	85.8	7.2
C-1A	47.6	40.2	7.3	84.6	4.4
C-1B	37.7	32.9	4.8	87.3	2.9
E-1	158.0	155.1	2.9	98.2	1.7
E-2	213.1	205.4	7.6	96.4	4.6
E-2A	73.9	70.0	3.9	94.8	2.3
E-2B	139.2	135.4	3.8	97.3	2.2
E-3	22.3	–	20.9	–	12.5
T-1	197.4	185.3	12.2	93.8	7.3
Total	167.7	100.0	63.6	59.6	37.9

(d) Modified recompression sCO_2 cycle

ID	$\dot{E}_F$ (MW)	$\dot{E}_P$ (MW)	$\dot{E}_D$ (MW)	ε (%)	y_D (%)
C-1	37.8	31.6	6.2	83.5	3.9
C-1A	5.9	4.8	1.0	82.2	0.7
C-1B	15.2	12.6	2.6	82.8	1.6
C-1C	16.8	14.2	2.6	84.6	1.6
E-1	148.1	145.2	2.9	98.0	1.8
E-2	173.6	156.8	16.8	90.3	10.5
E-2A	15.6	14.2	1.4	91.3	0.9
E-2B	158.1	142.6	15.5	90.2	9.7
E-3	12.5	–	10.9	–	6.8
E-3A	7.8	–	7.1	–	4.4
E-3B	4.7	–	3.8	–	2.4
T-1	146.9	137.8	9.0	93.8	5.7
Total	159.7	100.0	55.7	62.6	34.9

nificantly for the recompression cycle (c), exhibiting a significantly improved recuperator efficiency (E-2), in contrast to increased exergy destruction within compressor (C-1) and turbine (T-1) due to the higher mass flow rate. In comparison, the modified recompression cycle (d) shows a smaller exergy destruction for compression (C-1) as well as heat transfer in recuperator (E-2) and particularly for the precooler (E-3).

Compared to the results of the conventional thermodynamic analysis, the exergy analysis reveals the most important features of a sCO_2 cycle. It is clearly shown that compression and heat recovery are the most important aspects of a sCO_2 cycle and seem to exhibit a very high interaction. This leads to the conclusion that most of the compression has to be realized at the lowest temperature possible. However, by using a split flow configuration, the pinch point temperature of the recuperator can be effectively lowered, therefore significantly improving the cycle efficiency.

Table 7: Results of the economic analysis for the different cycle configurations with a cycle output of 100 MW and heat supplied at zero cost.

(a) Recuperated sCO_2 cycle (COE = $ 0.0215/kWh)

ID	Z ($)	Z/Z_{tot} (%)	$Z/\dot{W}_{Net}$ ($/kW$_e$)
C-1	36,641,694	30.6	366.4
E-1	36,987,181	30.9	369.9
E-2	11,001,614	9.2	110.0
E-3	7,333,066	6.1	73.3
T-1	27,658,320	23.1	276.6
Total	119,621,876	100.0	1196.2

(b) Reheating sCO_2 cycle (COE = $ 0.0214/kWh)

ID	Z ($)	Z/Z_{tot} (%)	$Z/\dot{W}_{Net}$ ($/kW$_e$)
C-1	36,068,350	30.2	360.7
E-1	36,536,508	30.6	365.4
E-1A	27,093,795	22.7	270.9
E-1B	9,442,713	7.9	94.4
E-2	12,222,226	10.2	122.2
E-3	7,187,489	6.0	71.9
T-1	27,525,631	23.0	275.3
T-1A	10,937,230	9.1	109.4
T-1B	16,588,401	13.9	165.9
Total	119,540,205	100.0	1195.4

(c) Recompression sCO_2 cycle (COE = $ 0.0262/kWh)

ID	Z ($)	Z/Z_{tot} (%)	$Z/\dot{W}_{Net}$ ($/kW$_e$)
C-1	52,090,694	32.1	520.9
C-1A	29,065,430	17.9	290.7
C-1B	23,025,265	14.2	230.3
E-1	34,466,642	21.2	344.7
E-2	38,204,642	23.5	382.0
E-2A	24,389,962	15.0	243.9
E-2B	13,814,680	8.5	138.1
E-3	6,398,035	3.9	64.0
T-1	31,315,257	19.3	313.2
Total	162,475,270	100.0	1624.8

(d) Modified recompression sCO_2 cycle (COE = $ 0.0196/kWh)

ID	Z ($)	Z/Z_{tot} (%)	$Z/\dot{W}_{Net}$ ($/kW$_e$)
C-1	27,497,767	23.5	275.0
C-1A	4,254,808	3.6	42.5
C-1B	11,011,405	9.4	110.1
C-1C	12,231,554	10.4	122.3
E-1	32,878,218	28.1	328.8
E-2	18,653,397	15.9	186.5
E-2A	9,311,439	8.0	93.1
E-2B	9,341,958	8.0	93.4
E-3	12,442,879	10.6	124.4
E-3A	5,677,140	4.9	56.8
E-3B	6,765,740	5.8	67.7
T-1	25,580,072	21.9	255.8
Total	117,052,332	100.0	1170.5

4.3 Economic Analysis

In selecting a cycle layout, the economic efficiency is the most important aspect. Consequently, the results of the thermodynamic and exergetic analysis have to be put into the context of an economic one. The results of the economic analysis are depicted in Table 7.

The specific cycle costs show that the simple recuperation (a), the reheating (b) and the modified recompression (d) cycles have similar specific investment costs. The specific investment costs for the recompression cycle (c) are significantly higher, due to the expensive high temperature compressor (C-1B). The baseline cost of electricity (COE) for the simple recuperation (a) and reheating (b) cycles are about $ 0.0215/kWh. Due to the higher investment cost, the COE for the recompression cycle is $ 0.0262/kWh. In contrast, the modified recompression cycle exhibits a COE of $ 0.0196/kWh.

It is further shown that the simple recuperation (a) and the reheating (b) cycle exhibit a similar cost structure for compression, expansion and heat recovery equipment, which is related to their

Table 8: Results of the exergoeconomic analysis for the different cycle configurations with a cycle output of 100 MW and heat supplied at zero cost.

(a) Recuperated sCO_2 cycle

ID	c_F (\$/GJ)	c_P (\$/GJ)	$\dot{C}_D$ (\$/h)	r (–)	f (–)
C-1	5.97	9.87	180.59	0.65	0.72
E-1	0.04	0.84	0.49	19.35	1.00
E-2	4.40	5.18	293.99	0.18	0.32
E-3	7.82	–	675.51	–	0.12
T-1	5.01	5.97	182.83	0.19	0.66
Total	0.97	5.97	262.18	5.19	0.85

(b) Reheating sCO_2 cycle

ID	c_F (\$/GJ)	c_P (\$/GJ)	$\dot{C}_D$ (\$/h)	r (–)	f (–)
C-1	5.93	9.84	175.89	0.66	0.72
E-1	0.08	0.87	0.91	9.53	1.00
E-1A	0.05	0.84	0.44	14.49	1.00
E-1B	0.16	0.95	0.47	4.81	1.00
E-2	4.24	4.93	318.17	0.16	0.33
E-3	7.82	–	662.10	–	0.12
T-1	4.99	5.93	169.56	0.19	0.67
T-1A	5.42	6.38	72.11	0.18	0.66
T-1B	4.71	5.64	97.32	0.20	0.68
Total	0.96	5.93	252.85	5.19	0.86

(c) Recompression sCO_2 cycle

ID	c_F (\$/GJ)	c_P (\$/GJ)	$\dot{C}_D$ (\$/h)	r (–)	f (–)
C-1	7.27	10.99	317.29	0.51	0.68
C-1A	7.27	11.15	191.77	0.53	0.66
C-1B	7.27	10.80	125.53	0.49	0.70
E-1	0.04	0.85	0.45	18.36	1.00
E-2	5.68	6.55	156.03	0.15	0.76
E-2A	5.69	7.24	79.40	0.27	0.80
E-2B	5.67	6.19	76.70	0.09	0.70
E-3	7.82	–	589.45	–	0.12
T-1	6.26	7.27	273.88	0.16	0.59
Total	0.90	7.27	207.13	7.03	0.91

(d) Modified recompression sCO_2 cycle

ID	c_F (\$/GJ)	c_P (\$/GJ)	$\dot{C}_D$ (\$/h)	r (–)	f (–)
C-1	5.45	9.61	122.41	0.76	0.74
C-1A	5.45	9.76	20.44	0.79	0.73
C-1B	5.45	9.69	51.06	0.78	0.73
C-1C	5.45	9.48	50.90	0.74	0.75
E-1	0.05	0.71	0.48	14.52	1.00
E-2	3.88	4.72	234.95	0.22	0.50
E-2A	3.93	6.63	19.27	0.69	0.86
E-2B	3.87	4.53	215.66	0.17	0.36
E-3	13.99	–	547.90	–	0.22
E-3A	12.94	–	330.35	–	0.18
E-3B	15.76	–	214.54	–	0.29
T-1	4.50	5.45	146.53	0.21	0.69
Total	0.82	5.45	164.59	5.64	0.90

single flow configuration. However, the split flow configurations show a significantly different cost share for the main components. In the case of the recompression cycle, the compressor and the recuperator cost shares are significantly higher. In contrast, the cost share of the precooler in the modified recompression cycle (d) is significantly higher.

In summary, the economic analysis shows that the cycle configuration has a substantial impact on the economic efficiency due to the high cost of turbomachinery.

4.4 Exergoeconomic Analysis

Based on the results of the exergetic and economic analyses, an exergoeconomic analysis is further used to provide information on the cost formation process on the component level. The main results are shown in Table 8.

Comparing the results of the different cycles, it is clear that despite the assumption that the generic heat is supplied at zero cost, some distinct features can be seen. Whereas the turbomachinery is characterized by the high importance of capital investment, the exergy destruction cost

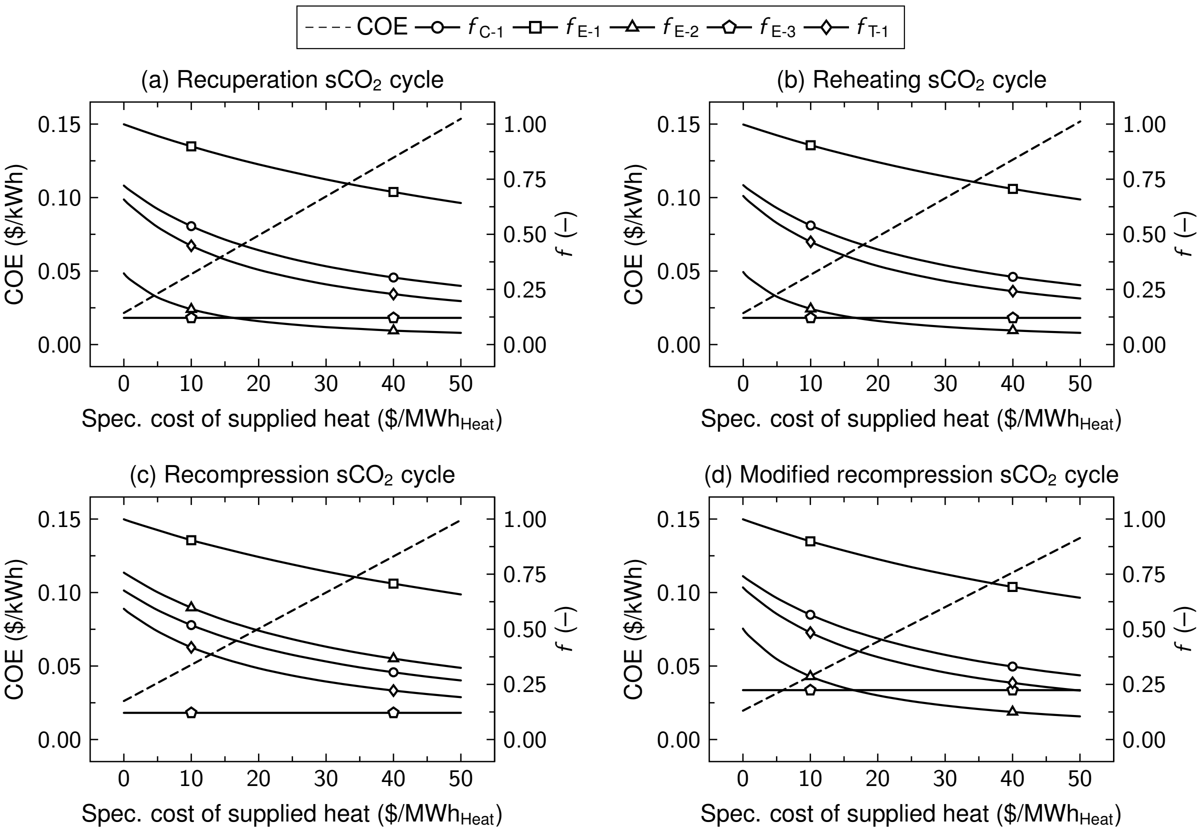


Figure 2: Sensitivity analyses of the simple sCO_2 cycle configurations regarding the influence of the cost of supplied heat.

associated with heat transfer in the recuperator and the precooler is more important compared to their monetary expenditures.

In order to gain additional information on the cost formation process within the different cycle layouts, the specific cost of the heat supplied to the cycles is increased. The results of the sensitivity analysis based on the parameter set of the detailed analysis are shown in Figure 2.

The COE for all cycle configurations constantly increases, with the modified recompression cycle generally exhibiting the lowest COE. Furthermore, the COE for the recuperation (a), reheating (b), and the recompression (c) cycle become equivalent for a specific cost of heat of about $30/MWh. It means that a higher specific cost of heat favors the recompression cycle design with its higher thermal efficiency and investment costs compared to the simple recuperation and reheating cycle.

The variation of the exergoeconomic factors for the cycle components reveals another important aspect. As all exergoeconomic factors continuously decrease except for the dissipative precooler (E-3), it can be concluded that each component's cost of exergy destruction and thus its efficiency becomes increasingly important. In particular, this applies to the recuperator (E-2) showing its importance for the cycle efficiency. Comparing the exergoeconomic factors of the different cycle configurations, it is also shown that the simple recuperation, reheating, and modified recompression cycle exhibit similar properties as the sequence of the different exergoeconomic factors and their variation is similar. In contrast, the recuperator (E-2) of the recompression cycle has a much higher exergoeconomic factor than of its higher investment costs.

In conclusion, the results of the exergoeconomic analysis thus support the results from the exergy analysis. From the exergoeconomic point of view, the following implications can be drawn for the design of sCO_2 cycles. It is necessary to chose a cycle layout where the compression is realized at low temperatures to minimize the cost of the compressors. Furthermore, the recuperator and precooler have to be designed to give the highest component efficiency possible, in order to minimize the exergy destruction in these components. From this point of view, the design of the modified compression cycle (d) seems to be a very promising one that incorporates all of these features. It thus provides a good starting point for further studies.

5 Conclusion

The present paper discussed several promising cycle configurations that have been suggested for sCO_2 cycles. Whereas the thermodynamic and economic analyses have shown that the cycle efficiency and its capital costs are interrelated, the integrated methods of exergy and exergoeconomic analysis have demonstrated how the different design features affect the economic efficiency of each cycle.

From this perspective it has to be concluded that at the current stage of cycle development, an integrated approach is required, in order to synthesize and realize promising cycle configurations for commercialization. From this perspective, the modified compression cycle should be further investigated.

Future work should therefore be concentrated on the integrated analysis of other cycle configurations that have not been incorporated in this study. Therefore, this study only presents a starting point for subsequent mapping of cycle features with respect to other cycles and its parameterizations. It is further necessary to include models for the upstream heat source in order to get more detailed information, particularly considering its cost. And for a better understanding of each cycle layout, advanced exergy-based methods can be used. In particular, the advanced exergy-based framework has to potential the provide additional useful information.

Nomenclature

Symbols

c	Specific costs	$\dot{C}$	Exergy related cost rate
e	Specific exergy	$\dot{E}$	Exergy rate
f	Exergoeconomic factor	$\dot{W}$	Power
p	Pressure	$\dot{Z}$	Monetary expenditure related cost rate
r	Relative cost difference	T	Absolute temperature
y	Exergy destruction ratio	ε	Exergetic efficiency

Subscripts and superscripts

D	Destruction	P	Product
F	Fuel	PH	Physical
L	Loss	T	Thermal
M	Mechanical		

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