Thermoeconomic analysis of sCO_2 cycle options for combined cycles

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ABSTRACT

The application of sCO_2-based power cycles in combination with conventional gas turbine cycles is a potential application for an sCO_2 cycle substituting the water-steam-based bottoming cycle in combined cycle processes. In the literature, many different cycle configurations have been proposed, analyzed, and optimized regarding their thermodynamic efficiency. However, as suitable data for estimating the capital costs of such cycles is of poor quality or simply not available, the question regarding the economic feasibility of the different cycle configurations remains generally unanswered or is associated with large uncertainties. For the present study, different sCO_2 power cycle configurations for waste heat recovery of state-of-the-art gas turbines are modeled and simulated using Aspen Plus. Based on the results from the simulations, thermoeconomic analyses are conducted employing a methodology that uses a non-dimensional approach based on thermoeconomic similitude providing the possibility to significantly reduce the number of required parameters to obtain robust results. It further facilitates the exploration of the design-space for decision-making taking into account the conflict between thermodynamic efficiency and capital costs. Thus the results assist in identifying the most promising cycle configurations and important aspects for further research and development of sCO_2 power cycles.

INTRODUCTION

The research and development for the application of supercritical CO_2 (sCO_2) for power generation has gained considerable momentum during the last decade. Based on the prospects for high-efficiency, low-emission power generation with favorable economics, because of smaller equipment sizes and higher operational flexibility [1], an extensive effort for future implementation and commercialization is currently made.

The potential areas of application span the whole field of different heat sources for thermal power cycles, ranging from fossil-fuel [2–4], nuclear [5–7], and concentrated-solar [8–11] to waste heat [12–15] based power generation. Compared to conventional water-steam-based power generation technologies [16], sCO_2 power cycle design follows a different design paradigm being characterized by high temperatures and pressures in combination with low-pressure ratios and a high recuperation potential [17], thereby providing the possibility to achieve higher efficiencies with a considerably smaller plant footprint [1] at comparable conditions.
To date, most of the research activities have concentrated on the identification of thermodynamically efficient sCO_2 power cycle configurations and the design of components for cycle application. However, due to the large number of different applications, it is unlikely that a single cycle configuration satisfies all the requirements for a high-efficiency design considering the trade-off between thermodynamic and economic efficiency depending on the specific use case. Within the compilation of sCO_2 power cycle by Crespi et al. [17], recompression designs have been identified as suitable for high-temperature applications like coal, nuclear, or concentrated-solar based designs. In contrast, split-expansion designs are considered favorable for waste heat recovery applications. With such a large variety of options available, it is possible to find a potentially attractive cycle design. However, the most important question regarding the economic feasibility remains largely unanswered until today. Only a limited number of studies [4, 15, 18–21] are available that investigate the economic feasibility of sCO_2-based power generation. With the intrinsic high uncertainty of the data, it is currently not possible to identify potentially economically feasible designs that provide the potential for long-term commercialization.
Experience shows that higher-efficiencies are likely to be achieved by integrating different thermodynamically-justified improvement options at the expense of making the systems more complex [22]. However, such an approach is often in conflict with the requirement of achieving high-efficiency operation at proven economic performance. In the present study on indirect (closed-cycle) sCO_2 power cycles for waste heat recovery of gas turbines for combined cycle applications [1], different cycle designs incorporating potential improvement options are analyzed regarding their economic feasibility. This is achieved by minimizing the uncertainty by reducing the number of parameters using similitude theory. Based on this approach, different cycles for waste heat recovery are compared and their potential application for high-efficiency and economically feasible power generation are discussed.

SYSTEM DESCRIPTION

Within the present study, the focus is put on sCO_2 based power cycles for waste heat recovery for gas turbine combined cycle applications. For that reason, a subset of promising designs is chosen that incorporates well-known principles for cycle improvement, e.g., recuperation (R), intercooled compression (IC), and split-expansion (SE), starting from the simple cycle design.

Cycle Configuration

For the present study, a given gas turbine design is used to analyze the potential to recover heat from the hot flue gases in order to improve the overall cycle efficiency. For that reason a GE 9E.04 gas turbine model [23] is used as an example with given ISO specifications which are adjusted using the data provided in [24] in order to accommodate for changes in fuel composition and backpressure that occurs because of the waste heat recovery heat exchangers pressure drop assuming that similar values for high-efficiency heat recovery generators are to be achieved [25, 26]. The resulting gas turbine data set is given in Table 1.
With a large set of possible sCO_2 power cycle designs being suggested in the literature [17], a hierarchically structured approach is advantageous in order to identify, analyze, and benchmark the different cycle designs according to their features. The early works of Gokhshtein and Dekhtyarev [27–29], Angelino

Table 1: Gas turbine specifications [23, 24].

Gas turbine data	
Model	GE 9E.04
Net power output	143.17 MW
Exhaust mass flow	416.00 kg/s
Exhaust temperature	545.30 °C
Exhaust pressure	1.04325 bar

Flue gas composition	
Nitrogen (N_2)	74.731 % (mol)
Oxygen (O_2)	13.364 % (mol)
Argon (Ar)	0.895 % (mol)
Carbon Dioxide (CO_2)	3.433 % (mol)
Water (H_2O)	7.577 % (mol)

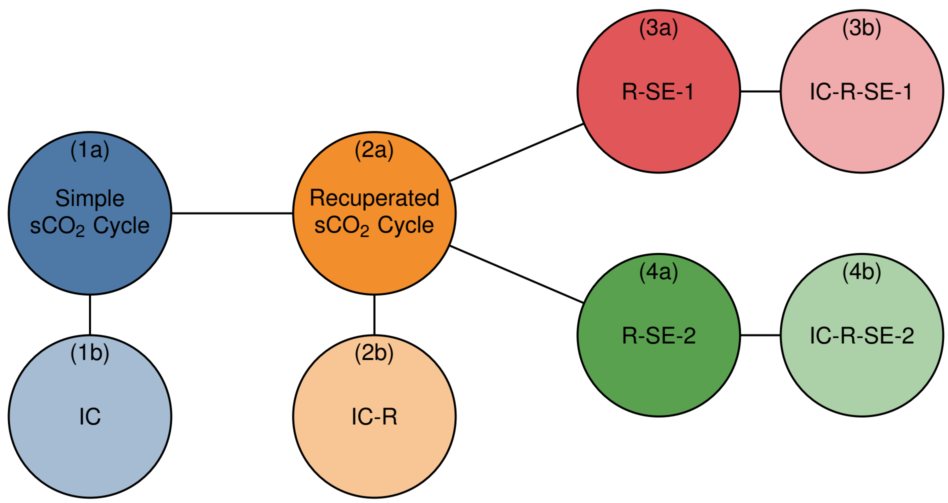
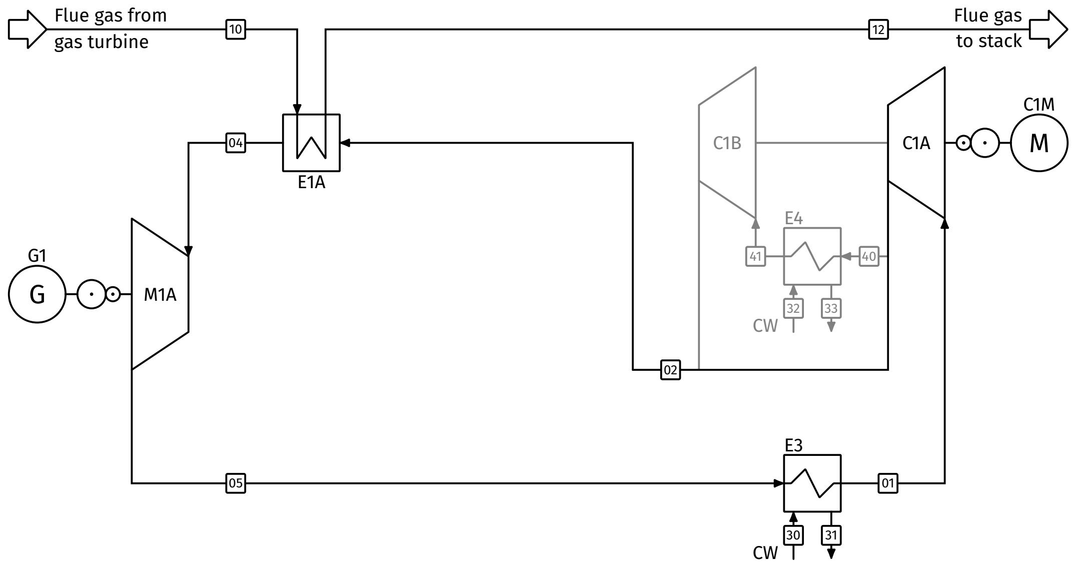
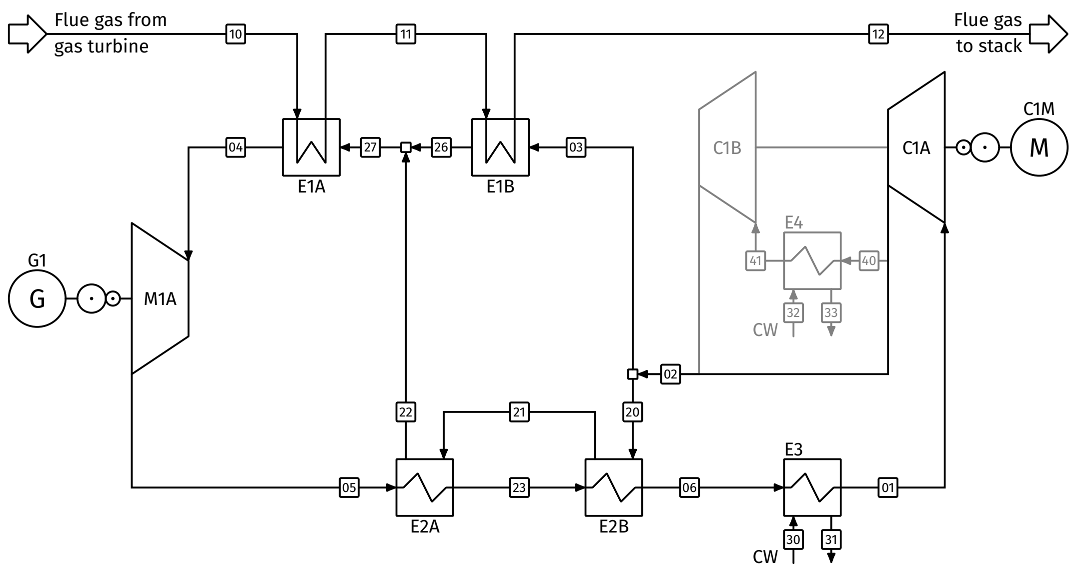


Figure 1: Hierarchical representation of the sCO_2 power cycle designs considered in this study.

[30, 31] and Feher [32] have identified the main features of a potential sCO_2 power cycle. Due to the particular properties of CO_2 which are different from that of water (H_2O), sCO_2 cycles configurations for high-efficiency power generation exhibit high-temperature, high-pressure but low-pressure ratio, and highly recuperative characteristics. The minimum configuration of such a power cycle consists of a compressor and turbine as well as heat exchangers for heat supply and removal. However, because of the low-pressure ratio and the high turbine outlet temperature, heat recuperation is necessary and an advantageous feature for achieving higher thermodynamic efficiencies.
Based on the given considerations, a systematic sCO_2 power cycle design hierarchy is used that is shown in Figure 1. The designs are chosen based on the review provided by Crespi et al. [17] and the configurations and considerations given by Kim et al. [13]. The simplest configuration is the four component simple sCO_2 power cycle, termed Design (1a), as depicted in Figure 2a. This cycle consists of a single cooler for heat removal (E3), and the high-temperature waste heat recovery heat exchanger (E1A). Furthermore, there is also the main compressor (C1A) which is connected to a motor (C1M), and the turbine (M1A) which drives the electric generator (G1).
The first improvement option investigated in this study, which is termed Design (2a) and shown in Figure 2b, is based on the integration of a recuperation (R) heat exchanger (E2) which is used for preheating purposes, thereby splitting the waste heat recovery heat exchanger into a low-temperature part (E1A) and a high-temperature part (E1B). The next option, incorporated in Design (3a) and shown in Figure 2c, employs a split-expansion (SE) concept, in which a recuperatively heated split stream drives a second turbine M1B. The last configuration option, Design (4a) which is depicted in Figure 2d, modifies Design (3a) in such a way, that the recuperator is used to preheat a split stream which then enters the high-temperature waste heat recovery heat exchanger E1A and is subsequently expanded in



(a) Design (1): Simple sCO_2 cycle with intercooled compression option.

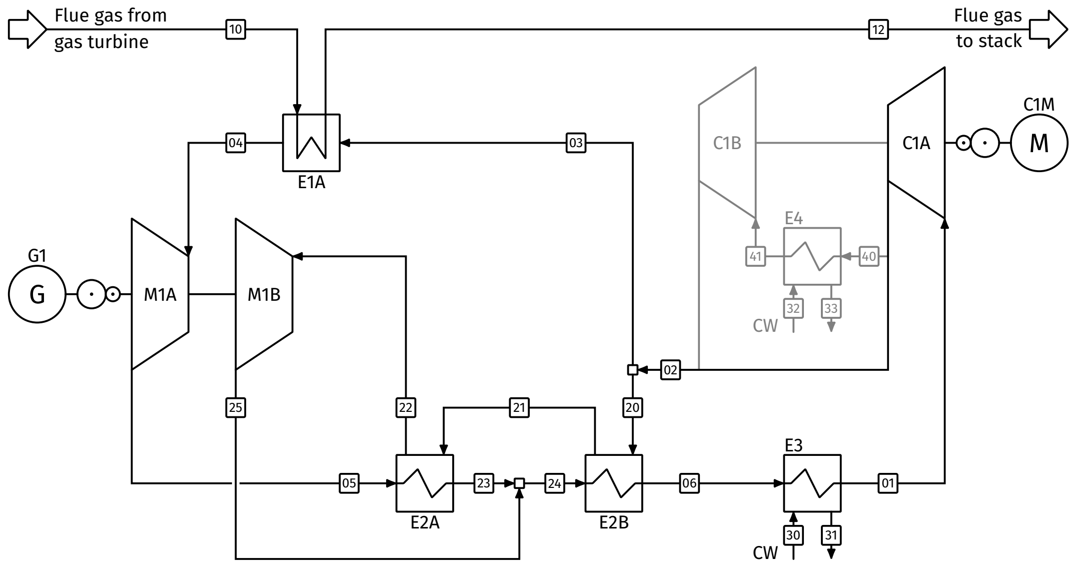


(b) Design (2): Recuperated sCO_2 cycle with intercooled compression option.

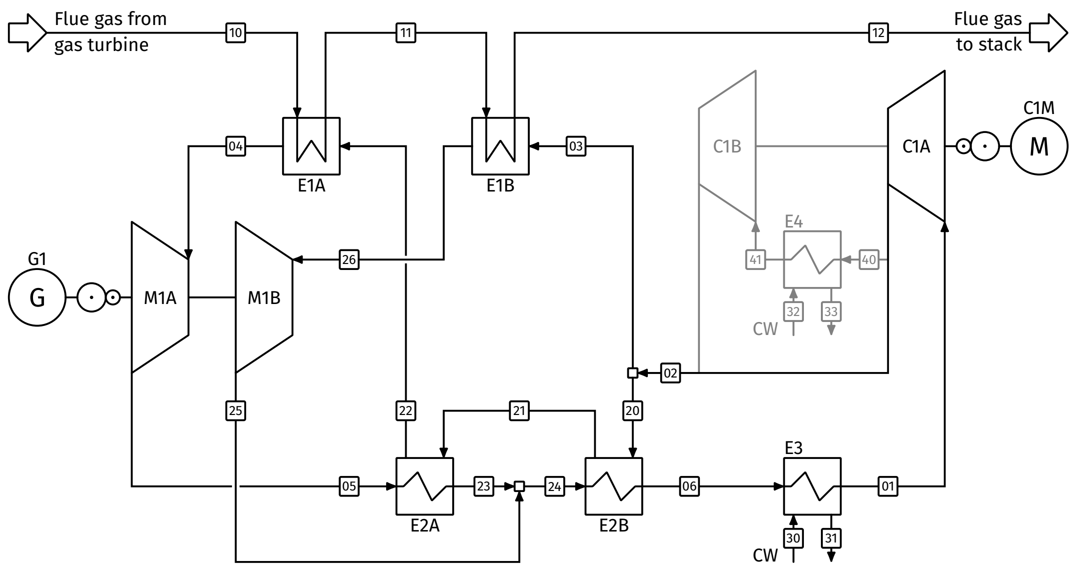
Figure 2: Flowsheets of the sCO_2 cycle configurations analyzed in this study.

turbine M1A, whereas the other split stream of the working fluid is heated by the low-temperature waste heat recovery heat exchanger E1B and then expanded in turbine M1B. In both designs, (3a) and (4a), the stream splitting of the working fluid into a low-temperature and high-temperature train provides the possibility to transfer heat more efficiently in the recuperator by compensating and accommodating for the differences in sCO_2 properties, effectively increasing the amount of waste heat transferred in the waste heat recovery heat exchanger E1.

The cycles chosen for this study are presenting the most basic features that can be used for realizing higher efficiency sCO_2 power cycle designs. In order to quantify the high potential of intercooling [20, 33] during the compression step, all Design 1–4 are also investigated employing a two-stage intercooled compression (IC) feature. These configurations are shown in Figure 2 by the intercooler E4 and the second compressor C1B, respectively.



(c) Design (3): Recuperated, dual split-expansion sCO_2 cycle with intercooled compression option.



(d) Design (4): Recuperated, split-expansion sCO_2 cycle with intercooled compression option.

Figure 2 (cont.): Flowsheets of the sCO_2 cycle configurations analyzed in this study.

Cycle Simulation and Parameterization

The comparison and benchmarking of the different cycle configurations and design options requires the use of best-practice guidelines, representing heuristics, and already available process data for modeling and simulation purposes. For the present study, Aspen Plus is used for simulation of the different cycle designs. The thermodynamic properties for the flue gas are determined by the Peng-Robinson equation of state, whereas the properties of the CO_2 are calculated using REFPROP [34, 35].

The site conditions are given in Table 2 and the composition of the natural gas that is used for the gas turbine is given in Table 2.

The different sCO_2 power cycle designs (Figure 2) are modeled and simulated using potentially viable design parameters according to [36], and [37]. For reasons of comparibility, the most efficient parameter-

Table 2: Environment and boundary conditions used for the simulations based on [38, 39].

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

Air composition

Nitrogen (N_2)	77.296 % (mol)
Oxygen (O_2)	20.727 % (mol)
Argon (Ar)	0.927 % (mol)
Carbon Dioxide (CO_2)	0.040 % (mol)
Water (H_2O)	1.010 % (mol)

Natural gas specifications

Lower Heating Value (LHV)	47201 kJ/kg
Higher Heating Value (HHV)	52295 kJ/kg
Temperature	15.0 °C
Pressure	30 bar

Natural gas composition

Methane (CH_4)	93.1 % (mol)
Ethane (C_2H_6)	3.2 % (mol)
Propane (C_3H_8)	0.7 % (mol)
n-Butane (C_4H_{10})	0.4 % (mol)
Carbon Dioxide (CO_2)	1.0 % (mol)
Nitrogen (N_2)	1.6 % (mol)

ization of each design is determined by mathematical optimization considering the net power generation of the sCO_2 cycle as the objective function. Based on the full set of parameters given in Table 3, the main characteristics of the cycles of the present study are a compressor inlet pressure of 70-120 bar with a fixed compressor inlet temperature of 32 °C. On the other hand, the turbine inlet is specified by a pressure of 300 bar, and a minimum temperature difference in waste heat recovery heat exchanger E1 [25, 26] set to 15 K. The size of the recuperator E2 is limited by an assumed maximum effectiveness of 0.9. Furthermore, cooling water is used as the cooling fluid for both recooler E3 and E4.

METHODOLOGY

The present study employs conventional thermodynamic and economic methodologies for analyzing the different cycles designs. For conducting thermoeconomic analyses both methodologies are combined which enables the evaluation of each cycle considering the trade-off between thermodynamic and economic performance.

Thermodynamic Analysis

Based on general conventions for analyzing power generation processes, the overall thermodynamic efficiency η of the gas turbine combined cycle is defined as the ratio of generated net power $\dot{W}_{\text{net}}$ and the chemical energy supplied by the fuel $\dot{m}_{\text{fuel}} \cdot LHV_{\text{fuel}}$ to the cycle [16] which is here defined by its lower heating value.

$$\eta = \frac{\dot{W}_{\text{net}}}{\dot{m}_{\text{fuel}} \cdot LHV_{\text{fuel}}} \tag{1}$$

It is however important to notice that a suitable framework [38, 40] for benchmarking different power cycles is required, in particular regarding the site-specific environment conditions, as given above, defining the cooling fluid's temperature and determining the potential cooling technology.

Table 3: Simulation parameters used for the analysis the sCO_2 power cycles based on literature data for benchmarking given by Weiland and Thimsen [36], and Crespi et al. [37].

Unit ID	Parameter	Value
M1A	Turbine Inlet Pressure	300 bar
M1A/B	Turbine Isentropic Efficiency	90 %
M1A/B	Turbine Mechanical Efficiency	99 %
C1A/B	Main Compressor Inlet Pressure	70-120 bar
C1A	Precompressor Pressure	25-75 bar
C1A/B	Compressor Isentropic Efficiency	85 %
C1A/B	Compressor Mechanical Efficiency	99 %
C1M	Motor Efficiency	97 %
G1	Electric Generator Efficiency	98.5 %
E1A/B	Primary Heat Exchanger Pressure Drop	200 kPa
E1A/B	Primary Heat Exchanger Minimum Temperature Difference	15 K
E1A/B	Minimum Flue Gas Outlet Temperature	85 °C
E2	Maximum Recuperator Effectiveness	0.9
E2	Recuperator Hot-Side Pressure Drop	280 kPa
E2	Recuperator Cold-Side Pressure Drop	140 kPa
E3, E4	Cooler Outlet Temperature	32 °C
E3, E4	Cooler Pressure Drop	15 kPa

Economic Analysis

The thermodynamic and economic evaluation, analysis, and optimization [41] of any energy-conversion system requires the comparison of annually recurrent monetary values. As each of these cost components can vary significantly over a system's economic life, levelized values are used for evaluation for the sake of comparability and simplicity.

In the literature, different methodologies [42, 43] are found, sometimes exhibiting significant methodological differences. The present study uses the total revenue requirement method (TRR) [41] which employs well-established procedures. Based on data for the total capital investment and proper assumptions regarding economic, financial, and operating parameters, the systems economic performance can be determined. The levelized cost rates are suitable input data for conducting a thermoeconomic analysis on the system's component level being related to its capital investment, fuel costs, and monetary expenses for operation and maintenance.

In case of sCO_2 power cycles, the determination of total capital investment based on capital cost estimation for the cycle equipment is still lacking substantial data. Therefore, it is only possible to use some generalized baseline data and to scale the equipment cost C accordingly using a reference value, and a suitable cost attribute X, e.g., compressor and turbine power, or heat exchanger area, and a scaling exponent n [41, 44].

$$C = C_{\text{ref}} \cdot \left(\frac{X}{X_{\text{ref}}}\right)^n \tag{2}$$

In the literature, scaling exponents n for different system components are available [4, 41, 45]. Furthermore, in case of design studies, only capital cost data from prior estimates and systems is available which requires updating employing suitable cost indices [41].

Thermoeconomic Analysis

For a conventional thermoeconomic analysis, the information provided by the economic analysis is used on the overall system level to provide a connection between of input streams representing fuels and auxiliary streams and output streams associated with useful products.

In the present study, cost rates $\dot{C}$ associated with heat and power streams are represented by the product of their thermodynamic quantities, e.g., for heat $\dot{Q}$ and power $\dot{W}$, and their associated specific cost c per unit of energy.

$$\dot{C}_\mathrm{Q} = c_\mathrm{Q} \cdot \dot{Q} \tag{3a}$$

$$\dot{C}_\mathrm{W} = c_\mathrm{W} \cdot \dot{W} \tag{3b}$$

Considering the operation of the overall system at steady state, the cost balance is used to define an input-output relationship considering the different streams streams expressing the total cost rate $\dot{C}_{i,\mathrm{out}}$ of the output streams as the sum of the input streams $\dot{C}_{i,\mathrm{in}}$ and cost streams associated with all monetary expenses $\dot{Z}$.

$$\sum_{i=1}^{m} \dot{C}_{i,\mathrm{out}} = \sum_{j=1}^{n} \dot{C}_{j,\mathrm{in}} + \sum_{k=1} \dot{Z}_k \tag{4}$$

For a gas turbine combined sCO_2 power cycle, Equation (4) contains terms for all monetary expenditures related to all the different system components $\dot{Z}$, as well as for the different energy streams related to chemical energy supplied by natural gas $\dot{C}_\mathrm{NG}$ and the heat removed by cooling fluid $\dot{C}_\mathrm{CW}$, i.e, the cooling water, and the cost rate $\dot{C}_\mathrm{W}$ for the net power output.

$$\dot{C}_\mathrm{W} = \dot{Z}_\mathrm{GT} + \dot{Z}_{\mathrm{sCO_2}} + \dot{C}_\mathrm{NG} + \dot{C}_\mathrm{CW} \tag{5}$$

Based on the presented common procedure for economic and thermoeconomic analyses, the economic analysis is subject to a large amount of highly uncertain data. This involves the cost estimation of the different components as well as the financial project parameters. For the analysis of sCO_2 cycle designs this is even more important because of the low technology readiness level (TRL) and the experimental nature of this technology.

However, in order to compare different sCO_2 power cycle designs, it is possible to employ similitude theory [46] as all the different cycles share a significant amount of common features. By choosing a proper reference sCO_2 cycle design, the different thermoeconomic parameters are effectively put into relation to each other and can be effectively compared [21, 47]. By introducing the operating and economic characteristics of the reference cycle design, e.g., the cost rates $\dot{C}_{i,\mathrm{ref}}$ and $\dot{Z}_{i,\mathrm{ref}}$, Equation (5) can be rewritten as:

$$\dot{C}_\mathrm{W} \frac{\dot{C}_\mathrm{W,ref}}{\dot{C}_\mathrm{W,ref}} = \dot{Z}_\mathrm{GT} \frac{\dot{Z}_\mathrm{GT,ref}}{\dot{Z}_\mathrm{GT,ref}} + \dot{Z}_{\mathrm{sCO_2}} \frac{\dot{Z}_{\mathrm{sCO_2,ref}}}{\dot{Z}_{\mathrm{sCO_2,ref}}} + \dot{C}_\mathrm{NG} \frac{\dot{C}_\mathrm{NG,ref}}{\dot{C}_\mathrm{NG,ref}} + \dot{C}_\mathrm{CW} \frac{\dot{C}_\mathrm{CW,ref}}{\dot{C}_\mathrm{CW,ref}} \tag{6}$$

After applying Equation (5) and introducing the dimensionless similarity numbers k_i which quantify the importance of each cost component in comparison to a choosen reference, e.g., the reference cost rate $\dot{Z}_\mathrm{GT,ref}$ of the gas turbine system, the following relationship is obtained:

$$k_\mathrm{W,ref} \frac{\dot{C}_\mathrm{W}}{\dot{C}_\mathrm{W,ref}} = \frac{\dot{Z}_\mathrm{GT}}{\dot{Z}_\mathrm{GT,ref}} + k_{\mathrm{sCO_2,ref}} \frac{\dot{Z}_{\mathrm{sCO_2}}}{\dot{Z}_{\mathrm{sCO_2,ref}}} + k_\mathrm{NG,ref} \frac{\dot{C}_\mathrm{NG}}{\dot{C}_\mathrm{NG,ref}} + k_\mathrm{CW,ref} \frac{\dot{C}_\mathrm{CW}}{\dot{C}_\mathrm{CW,ref}} \tag{7}$$

The specific constant factors k_i are thermoeconomic similarity numbers quantifying the contribution of carrying charges and fuel cost in comparison to the monetary expenses of the gas turbine, respectively. In addition, integration of Equations (3) into Equation (7) provides a direct relationship between the specific cost of electricity generated by the cycle, and its monetary expenses, fuel consumption, and cooling water demand.

Another feature of gas-turbine-based combined cycles can be used for further simplifying Equation (7). As the monetary expenses of the gas turbine and the cost rate of the natural gas do not change in comparison to the design and configuration of the sCO_2 bottoming cycle, it can be easily reduced using the parameters of the standalone gas turbine system. Furthermore, by assuming that the thermoeconomic cost similarity number for the cooling water is significantly smaller than the monetary expenses for the gas turbine system, it can be neglected. Thereby, Equation (7) reduces to a very practical format.

$$k_{\mathrm{W,ref}}\frac{\dot{C}_{\mathrm{W}}}{\dot{C}_{\mathrm{W,ref}}} = k_{\mathrm{W,ref}}\frac{c_{\mathrm{W}}}{c_{\mathrm{W,ref}}}\frac{\dot{W}_{\mathrm{GT,ref}}+\dot{W}_{\mathrm{sCO_2}}}{\dot{W}_{\mathrm{GT,ref}}+\dot{W}_{\mathrm{sCO_2,ref}}} = 1 + k_{\mathrm{sCO_2,ref}}\frac{\dot{Z}_{\mathrm{sCO_2}}}{\dot{Z}_{\mathrm{sCO_2,ref}}} \tag{8}$$

This equation finally shows the conflicting objectives for the thermoeconomic improvement of power generation technologies in terms of monetary expenses, cycle complexity, and thermodynamic efficiency. Compared to a conventional thermoeconomic analysis, the amount of inherent uncertainty is significantly reduced due to the considerable reduction in parameters being now fully incorporated in and represented by the thermoeconomic factors k_i.

As the scope of the current study is the evaluation of different sCO_2 power cycle designs incorporating general improvement options, it is convenient to further identify the contributions and changes of each power cycle component. The cost rate $\dot{Z}$ of the overall power cycle is defined as the sum of the cost rates $\dot{Z}_k$ of each component k.

$$\dot{Z} = \sum_{i=1}^{n} \dot{Z}_k \tag{9}$$

In case of the different sCO_2 power cycles considered in this study, the following components are effectively considered as a single item, as they share common features, for the cost estimation procedure based on the flowsheet shown in Figure 2.

- Cooler E3/4: E3, E4
- Recuperator E2: E2A, E2B
- Waste Heat Recovery Heat Exchanger E1: E1A, E1B
- Compressor C1: C1A, C1B
- Turbine M1: M1A, M1B

Under the premise that the different cycle designs are compared using the same financial and economic parameters as discussed above, the cost rate of each component are related to the cost rate of the reference cycle design, respectively.

$$\begin{aligned}\frac{\dot{Z}_{\mathrm{sCO_2}}}{\dot{Z}_{\mathrm{sCO_2,ref}}} = {} & x_{\mathrm{C1,ref}}\frac{\dot{Z}_{\mathrm{C1}}}{\dot{Z}_{\mathrm{C1,ref}}} + x_{\mathrm{C1M,ref}}\frac{\dot{Z}_{\mathrm{C1M}}}{\dot{Z}_{\mathrm{C1M,ref}}} + x_{\mathrm{M1,ref}}\frac{\dot{Z}_{\mathrm{M1}}}{\dot{Z}_{\mathrm{M1,ref}}} + x_{\mathrm{G1,ref}}\frac{\dot{Z}_{\mathrm{G1}}}{\dot{Z}_{\mathrm{G1,ref}}} \\ & + x_{\mathrm{E1,ref}}\frac{\dot{Z}_{\mathrm{E1}}}{\dot{Z}_{\mathrm{E1,ref}}} + x_{\mathrm{E2,ref}}\frac{\dot{Z}_{\mathrm{E2}}}{\dot{Z}_{\mathrm{E2,ref}}} + x_{\mathrm{E3/4,ref}}\frac{\dot{Z}_{\mathrm{E3/4}}}{\dot{Z}_{\mathrm{E3/4,ref}}}\end{aligned} \tag{10}$$

With the assumption that the same economic parameters, e.g, availability, share in operation and maintenance costs, hold true for the different components, Equation (10) further reduces to a form that allows for its evaluation using Equation (2) for each component. Therefore, it is possible to account for different component designs and changes in cycle parameters in terms of cost ratios and adjusted degression and scaling exponents. Assuming that the design of the different components does not change significantly, the cost estimation relationships are established using the heat transfer capacity UA for heat exchangers, and the power $\dot{W}$ for compressors, motors, turbines, and generators.

$$\begin{aligned}\frac{Z_{\mathrm{sCO_2}}}{Z_{\mathrm{sCO_2,ref}}} = x_{\mathrm{C1,ref}}\left(\frac{\dot{W}_{\mathrm{C1}}}{\dot{W}_{\mathrm{C1,ref}}}\right)^{n_{\mathrm{C1}}} &+ x_{\mathrm{C1M,ref}}\left(\frac{\dot{W}_{\mathrm{C1M}}}{\dot{W}_{\mathrm{C1M,ref}}}\right)^{n_{\mathrm{C1M}}} + x_{\mathrm{M1}}\left(\frac{\dot{W}_{\mathrm{M1}}}{\dot{W}_{\mathrm{M1,ref}}}\right)^{n_{\mathrm{M1}}} \\ &+ x_{\mathrm{G1,ref}}\left(\frac{\dot{W}_{\mathrm{G1}}}{\dot{W}_{\mathrm{G1,ref}}}\right)^{n_{\mathrm{G1}}} + x_{\mathrm{E1,ref}}\left(\frac{UA_{\mathrm{E1}}}{UA_{\mathrm{E1,ref}}}\right)^{n_{\mathrm{E1}}} \\ &+ x_{\mathrm{E2,ref}}\left(\frac{UA_{\mathrm{E2}}}{UA_{\mathrm{E2,ref}}}\right)^{n_{\mathrm{E2}}} + x_{\mathrm{E3/4,ref}}\left(\frac{UA_{\mathrm{E3/4}}}{UA_{\mathrm{E3/4,ref}}}\right)^{n_{\mathrm{E3/4}}}\end{aligned} \tag{11}$$

The scaling exponents n_k used for cost estimation, and the share in cycle cost x_k of each cycle component are estimated using the baseline data given in [4, 19, 45]. In case no absolute value for the reference system's cost is available, the share in cycle costs x_k for each component can be determined based on heuristics or simply estimated based on experience. For that reason, the cost relationships provided by Reference [4] are used for estimating the cost structure of the sCO_2 reference cycle design which is compared with data given for conventional combined cycle power plant steam cycles [48] for a plausibility check.

As the complete cost estimation work of the thermoeconomic analysis reduces to the estimation of cost shares and degression coefficients x and n, and as the sum of the cost shares always has to equal unity, a sensitivity analysis assuming a 50 % error for the equipment costs using a uniform distribution is conducted. For reasons of convenience, a Latin hypercube sampling scheme with 20 times the number of variables [49] is used for exploring the parameter space.

For the following analyses, Equations (8) and (11) can be used conveniently to evaluate the different sCO_2 power cycle designs considered in this study regarding their potential to provide an economically advantageous design.

RESULTS AND DISCUSSION

Based on the data provided by the simulations and calculations, the results of the thermodynamic and thermoeconomic studies for the different sCO_2 power cycle designs and improvement options are analyzed and discussed.

Thermodynamic Analyses

The results of the simulations of the different sCO_2 power cycle designs are given in Table 4. The cycle designs differ in terms of massflow rates of the sCO_2 working fluid and operating temperatures of the recuperator, whereas the pressures are generally similar, with the exception of Design (1b). In general, it is found that the cycle designs with intercooling allow for a higher expansion ratio in the turbine section thus providing the potential to achieve higher efficiencies.

The simulation results concerning the power output and the thermodynamic efficiency of the different sCO_2 cycles are presented in Table 5. The highest power output is obtained by the intercooled split-expansion cycle of Design (4b) with 65.50 MW and a cycle efficiency of 31.18 %. The second highest power output and efficiency is obtained for the same Design (4a) without intercooled compression, with

Table 4: Stream parameters of the different sCO_2 cycle simulations.

Design (1a): Simple sCO_2 Cycle

Stream-No.	Temperature (°C)	Pressure (bar)	Massflow (kg/s)
01	32.00	76.23	328.97
02	80.22	302.00	328.97
04	510.30	300.00	328.97
05	354.25	76.38	328.97

Design (1b): Intercooling, Simple sCO_2 Cycle

Stream-No.	Temperature (°C)	Pressure (bar)	Massflow (kg/s)
01	32.00	29.70	315.82
02	78.95	302.00	315.82
04	530.30	300.00	315.82
05	281.82	29.85	315.82

Design (2a): Recuperated sCO_2 Cycle

Stream-No.	Temperature (°C)	Pressure (bar)	Massflow (kg/s)
01	32.00	78.41	698.46
02	73.55	303.40	698.46
03	73.55	303.40	249.18
04	377.57	300.00	698.46
05	241.96	81.36	698.46
06	94.07	78.56	698.46

Design (2b): IC-R sCO_2 Cycle

Stream-No.	Temperature (°C)	Pressure (bar)	Massflow (kg/s)
01	32.00	66.26	630.09
02	71.78	303.40	630.09
03	71.78	303.40	232.68
04	395.30	300.00	630.09
05	242.87	69.21	630.09
06	90.54	66.41	630.09

Design (3a): R-SE1 sCO_2 Cycle

Stream-No.	Temperature (°C)	Pressure (bar)	Massflow (kg/s)
01	32.00	77.31	658.38
02	75.84	302.00	658.38
03	75.84	302.00	314.84
04	530.30	300.00	314.84
05	377.12	80.26	314.84
06	102.66	77.46	658.38
22	307.22	300.00	343.54
25	177.65	80.26	343.54

Design (3b): IC-R-SE1 sCO_2 Cycle

Stream-No.	Temperature (°C)	Pressure (bar)	Massflow (kg/s)
01	32.00	63.62	599.31
02	74.55	302.00	599.31
03	74.55	302.00	314.44
04	530.30	300.00	314.44
05	357.91	66.57	314.44
06	102.44	63.77	599.31
22	314.48	300.00	284.86
25	167.94	66.57	284.86

Design (4a): R-SE2 sCO_2 Cycle

Stream-No.	Temperature (°C)	Pressure (bar)	Massflow (kg/s)
01	32.00	78.17	690.10
02	74.00	303.40	690.10
03	74.00	303.40	271.44
04	502.44	300.00	418.65
05	353.41	81.12	418.65
06	89.10	78.32	690.10
22	300.69	302.00	418.65
25	172.75	81.12	271.44
26	300.69	300.00	271.44

Design (4b): IC-R-SE2 sCO_2 Cycle

Stream-No.	Temperature (°C)	Pressure (bar)	Massflow (kg/s)
01	32.00	67.24	642.46
02	72.09	303.40	642.46
03	72.09	303.40	270.82
04	528.22	300.00	371.64
05	361.46	70.19	371.64
06	85.90	67.39	642.46
22	300.53	302.00	371.64
25	160.11	70.19	270.82
26	300.53	300.00	270.82

Table 5: Results of the gas turbine sCO_2 combined cycle design simulations.

Design	$\dot{W}_{sCO_2,net}$ (MW)	x_{sCO_2} (%)	η_{sCO_2} (%)	$\dot{W}_{net}$ (MW)	η_{ov} (%)	Heatrate (kJ/kWh)
(1a)	39.17	35.84	18.96	182.34	46.60	7725.8
(1b)	47.43	43.41	22.90	190.61	48.71	7390.8
(2a)	57.81	52.91	27.60	200.99	51.36	7009.1
(2b)	58.96	53.95	28.65	202.13	51.65	6969.5
(3a)	60.02	54.92	28.79	203.19	51.92	6933.2
(3b)	61.37	56.16	29.33	204.55	52.27	6887.2
(4a)	64.35	58.89	30.75	207.53	53.03	6788.3
(4b)	65.50	59.94	31.18	208.68	53.33	6750.9

$x_{CO_2} = \dot{W}_{sCO_2} / \dot{W}_{sCO_2,max}$

Table 6: Main results of the sCO_2 cycle design simulations.

Design	$\dot{Q}_{E1}$ (MW)	$\dot{Q}_{E2}$ (MW)	$\dot{Q}_{E3/4}$ (MW)	$\dot{W}_{C1}$ (MW)	$\dot{W}_{C1M}$ (MW)	$\dot{W}_{M1}$ (MW)	$\dot{W}_{G1}$ (MW)	$\dot{W}_{net}$ (MW)
(1a)	206.53	–	165.52	12.85	13.23	53.74	53.20	39.17
(1b)	207.09	–	156.37	31.46	32.40	81.87	81.05	47.43
(2a)	209.46	121.33	148.47	25.76	26.53	86.49	85.63	57.81
(2b)	205.78	109.12	143.50	27.35	28.18	89.35	88.46	58.96
(3a)	208.46	129.83	145.27	24.72	25.46	87.66	86.78	60.02
(3b)	209.02	111.12	144.28	27.28	28.10	91.75	90.83	61.37
(4a)	209.26	156.13	141.57	25.56	26.33	92.99	92.06	64.35
(4b)	210.10	139.97	141.11	27.43	28.25	96.14	95.18	65.50

an sCO_2-cycle power output of 64.35 MW and an efficiency of 30.75 %. Designs (3), (2), and (1) are ranked in consecutive order, thereby showing that a cycle's very design configuration has the largest influence on its power output and efficiency. Furthermore, it has to be noted that the intercooling option always provides a small but tangible benefit. In general, it can be seen that the simple yet efficient Designs (3) and (4) provide a significant improvement in terms of the combined cycle efficiency, almost achieving the same efficiency as complex water-steam-based cycles with two-pressure heat recovery steam generators.

Furthermore, it is interesting to see that the different cycle parameters vary considerably in terms of heat transfer, compression, and expansion equipment with regard to the different improvement options. It can be seen in Table 6 that the largest amount of compression power it required in Design (1b) because of its very low turbine outlet pressure which is a significant deviation from all other cycle designs. Moreover, it found that higher efficiency cycles exhibit a larger amount of heat transferred in the waste heat recovery heat exchanger and recuperator and also feature compressors, motor, turbines, and generators with a higher power rating. From the perspective of a better heat utilization and high thermodynamic efficiency, the split-expansion cycles designs seem to be the best options.

Thermoeconomic Analyses

Thermoeconomic analyses are conducted in order to identify the economic feasibility of each cycle design using the methodology presented earlier.

Based on the available general relationship between thermodynamic efficiency and monetary expenditures, detailed thermoeconomic analyses for the different sCO_2 power cycle designs are conducted.

Table 7: Results of the thermoeconomic analyses for the different sCO_2 cycle designs.

Parameter			Designs							
			(1a)	(1b)	(2a)	(2b)	(3a)	(3b)	(4a)	(4b)
$\dot{W}_{C1}/\dot{W}_{C1,ref}$			0.499	1.221	1.000	1.062	0.960	1.059	0.992	1.065
$\dot{W}_{C1M}/\dot{W}_{C1M,ref}$			0.499	1.221	1.000	1.062	0.960	1.059	0.992	1.065
$\dot{W}_{M1}/\dot{W}_{M1,ref}$			0.621	0.947	1.000	1.033	1.013	1.061	1.075	1.112
$\dot{W}_{G1}/\dot{W}_{G1,ref}$			0.621	0.947	1.000	1.033	1.013	1.061	1.075	1.112
$UA_{E1}/UA_{E1,ref}$			0.862	1.111	1.000	0.882	1.104	1.103	1.444	1.859
$UA_{E2}/UA_{E2,ref}$			0.000	0.000	1.000	0.906	0.881	0.808	1.442	1.278
$UA_{E3/4}/UA_{E3/4,ref}$			0.390	0.649	1.000	1.522	0.909	1.424	0.999	1.558
	$x_{i,ref}$	n_i	(1a)	(1b)	(2a)	(2b)	(3a)	(3b)	(4a)	(4b)
$Z_{C1}/Z_{C1,ref}$	0.125	0.3992	0.095	0.135	0.125	0.128	0.123	0.128	0.125	0.128
$Z_{C1M}/Z_{C1M,ref}$	0.080	0.6062	0.052	0.090	0.080	0.083	0.078	0.083	0.080	0.083
$Z_{M1}/Z_{M1,ref}$	0.060	0.5561	0.046	0.058	0.060	0.061	0.060	0.062	0.062	0.064
$Z_{G1}/Z_{G1,ref}$	0.035	0.5463	0.027	0.034	0.035	0.036	0.035	0.036	0.036	0.037
$Z_{E1}/Z_{E1,ref}$	0.440	0.7000	0.397	0.474	0.440	0.403	0.472	0.471	0.569	0.679
$Z_{E2}/Z_{E2,ref}$	0.170	0.7544	0.000	0.000	0.170	0.158	0.155	0.145	0.224	0.205
$Z_{E3/4}/Z_{E3/4,ref}$	0.090	0.7500	0.044	0.065	0.090	0.123	0.084	0.117	0.090	0.126
Z/Z_{ref}	1.000		0.661	0.857	1.000	0.992	1.007	1.042	1.186	1.321

The results are presented in Table 7. Based on the calculation of the design parameters for each cycle component determining its relative size and cost, the economic scaling laws are applied employing a suitable scaling coefficient. The resulting ratio of each cycle design's actual cycle cost rate in comparison to the reference cycle cost rate is thus determined.

It is shown in Table 7 that the cost ratios for all cycle designs are comparable in case of the intercooling and non-intercooling option. In contrast, as mentioned before, a more efficient waste heat utilization requires larger components, therefore increasing the cost compared to the reference design. Based on the determined cost distribution for the components of each cycle, it can be seen that the main cost drivers are the waste heat heat exchanger, recuperator, and compressor. The share in overall cycle cost is significantly larger for all three components in case of the split-expansion Designs (3) and (4) compared to the reference case of Design (2a). In contrast, the shares in overall cycle cost for the cooling and turbine components change marginally because of favorable scaling of the components' associated costs.

A major advantage of the methodology used for the thermoeconomic analysis is the possibility to determine the maximum allowable ratio of monetary expenses for each cycle design in comparison to the reference case of single gas turbine operation for obtaining equal specific cost of electricity. The recuperated, non-intercooled Design (2a) is chosen as the reference design for each sCO_2 cycle as it exhibits all the features that are found within all other cycle designs. The results are shown in Figure 3. It is found that for a design to become economically advantageous compared to the reference cycle design, there is a maximum allowable difference in capital cost to justify less or more efficient designs. This correctly allows for the identification of the well-known conflict between higher thermodynamic efficiencies and increased capital investment subject to a distinct economic case represtend by the thermoeconomic factor k.. Due to the very nature of the problem of waste heat recovery, higher cost can be justified by a more efficient cycle design. In general, the Designs (3) and (4) can be more expensive than the reference Design (2) because of their higher efficiencies, whereas the Design (1) is required to be significantly cheaper.

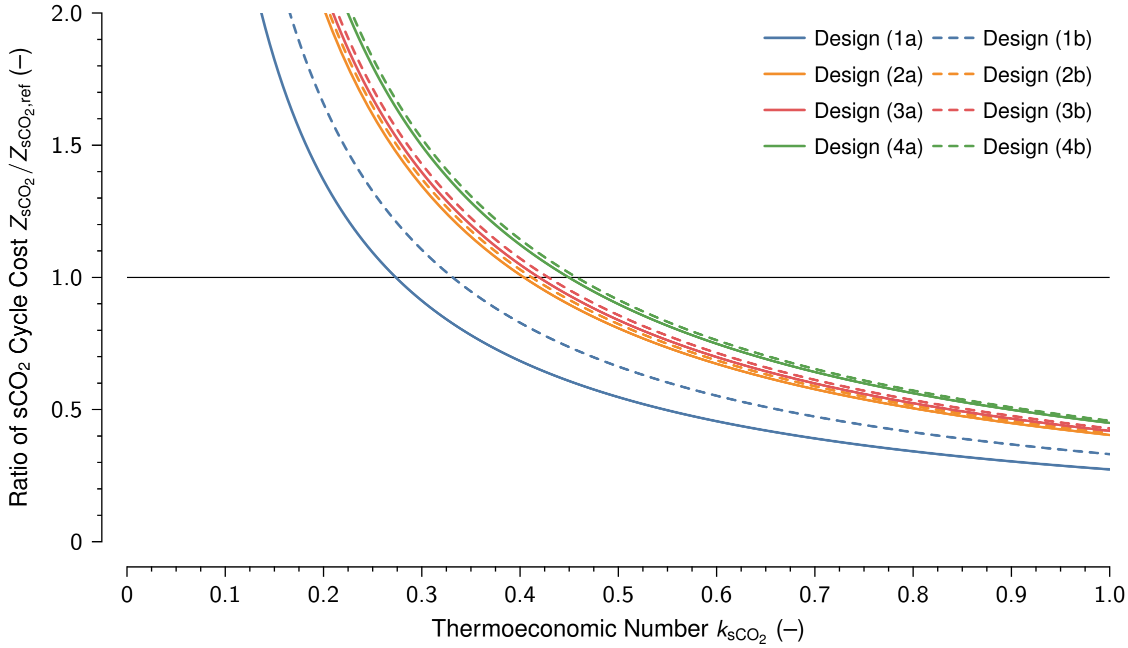


Figure 3: Isolines determining the maximum ratio of cost rates for the overall sCO_2 cycle design resulting in the same specific cost of electricity as for a standalone gas turbine.

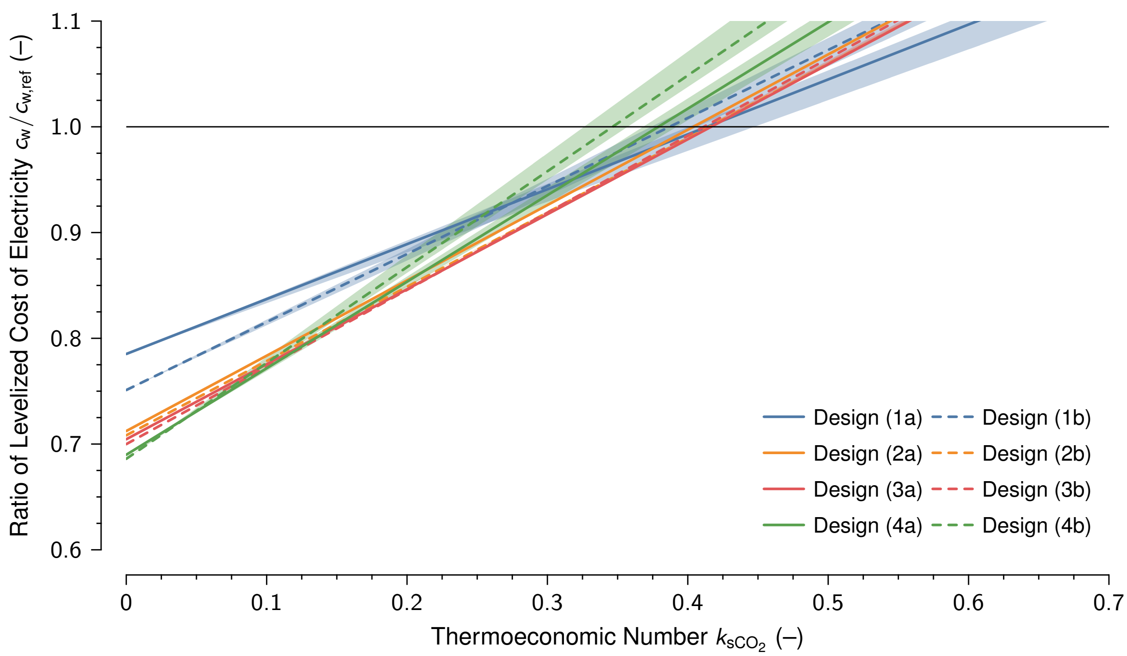


Figure 4: Actual ratio of levelized cost of electricity for the sCO_2 power cycle designs considered in this study. The sensitivity of the calculations for each design with respect to the equipment cost estimation is represented by the highlighted area.

Based the ratio of cost rates determined for the different cycle designs, it is possible to identify the actual ratio of the levelized cost of electricity for each power cycle design. The results are shown in Figure 4. It is found that the recuperated, non-intercooled sCO_2 cycle Design (4a) provides a the highest benefit in terms of the smallest levelized cost of electricity over the first segment of the thermoeconomic number k_{sCO_2} representing situations where the costs for the gas turbine system and the natural gas are much more important than for the sCO_2 cycle. In contrast, the ratio of levelized cost of electricity for the recuperated Design (3a) becomes favorable at thermoeconomic number k_{sCO_2} larger than 0.15. At thermoeconomic numbers exceeding 0.4, the simple cycle Design (1a) is the best option. However, at such high thermoeconomic numbers, the sCO_2-based power cycles for waste heat recovery in gas turbine combined cycle applications are rendered economically infeasible, as the ratio of levelized costs of electricity crosses the reference line for a standalone gas turbine system.

Another result from this discussion shows that the cycle Designs (2) and (3) are very similar in terms of their thermoeconomic features, with Design (3) being, in general, the thermodnamically and economically more efficient configuration.

The sensitivity study of the results is also shown in Figure 4, indicated by the colored areas. It is shown that, in comparison, the Designs (2) and (3) are less sensitive to the assumptions and the previous discussion remains largely valid. However, due to the more significant changes in the Designs (1) and (4) compared to the reference Design (2a), uncertainty increases but is not contrasting the previous results and discussions.

In general, the results of this study can be explained mostly by the scaling relations for estimating each component cost. Due to the unfavorable scaling for the size of the heat exchanger components and their large share in the overall cost of the sCO_2 bottoming cycle designs, a significant increase in heat exchange capacity *UA* is highly detrimental for the cycles economic performance if the technology is expensive in general. However, this is sometimes compensated by the increased efficiency in case of very expensive heat generation technologies and fuels. On the other hand, with a more favorable cost-size degression, an increase in compression and expansion equipment can be economically justified if significantly higher efficiencies can be realized and the heat transfer components can be reduced in size.

CONCLUSIONS

The present study discussed several promising cycle configurations that have been suggested for sCO_2 cycles for gas turbine combined cycle applications. Whereas the thermodynamic and economic analyses have shown that the cycle efficiency and its equipment costs are strongly interrelated, the integrated concept of the thermoeconomic analysis has demonstrated how the different design features affect the economic efficiency of each cycle.

A dimensionless approach for the thermoeconomic evaluation of the sCO_2 power cycle designs has been used and applied for comparing the economic viability of a simple cycle design and three different options for improving the cycle efficiency employing recuperation, intercooled compression, and split-expansion. The approach presented in this study has significantly reduced the number of parameters required for the evaluation of the different cycle designs and provided informative results.

The results for a particular system study, using a benchmarking parameterization, have shown that an increase in heat exchanger capacity is economically justifiable only if the monetary expenses of the sCO_2-cycle are small compared to the gas turbine and fuel costs because of an unfavorable cost degression with respect to the heat exchangers. On the other hand, configurations with larger compressors and turbines can be economically justified if a higher efficiency is achieved and the heat transfer equipment is reduced in size. The recuperated split-expansion Designs (3, 4) seem to be a good starting point for conducting further analyses and studies.

In the future, a more detailed study taking into account baseline data should be conducted in order to show the viability of the different cycle design and to determine which designs should be further developed. Based on the simplicity of the new approach, the different sCO_2 power cycle designs should can be further refined in the future considering other efficiency improvement options. Moreover, in combination with sensitivity and optimization studies on the cycle parameters, the suggested approach is able to provide a systematic and robust basis for the design of sCO_2 power cycles. The integration of an exergoeconomic analysis offers even more information on the different cycle designs by revealing the actual cost formation process within each power cycle providing tangible information which design decisions provide the possibility of obtaining even more cost-effient designs.

NOMENCLATURE

Symbols

C	Cost
$\dot{C}$	Cost rate
$\dot{W}$	Power
UA	Heat transfer capacity
$\dot{Z}$	Cost rate
c	Specific costs
k	Thermoeconomic number
n	Degression, scaling exponent
x	Fraction
η	Efficiency

Superscripts and subscripts

ref	Reference
CW	Cooling water
GT	Gas turbine
NG	Natural gas
W	Power

Acronyms

C	Compressor
E	Heat exchanger
G	Generator
IC	Intercooled compression
M	Turbine, motor
R	Recuperation
SE	Split expansion

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