Exergy Analysis of the Allam Cycle

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Abstract

In the generation of electricity from fossil fuels a large part of the CO_2 emissions can be omitted by employing carbon capture and sequestration. This, however, significantly reduces the overall plant efficiency. The recently developed Allam cycle presents a new approach to combine low-cost power generation with complete CO_2 separation at a high-efficiency. The cycle employs oxy-fuel combustion of hydrocarbons in a supercritical CO_2 environment with subsequent expansion in a hybrid gas-steam turbine. The resulting high CO_2 concentration in the turbine exhaust simplifies the step of CO_2 separation. As a result, the cycle is supposed to achieve efficiencies of up to 59 % (LHV) for natural gas operation. Based on the published cycle data, an exergy analysis is used to investigate the cycle performance regarding its parameters and configurations for natural gas combustion. A sensitivity analysis is used to study the effects of the most important thermodynamic parameters. The exergy concept thereby helps to identify the sources of the cycle's thermodynamic inefficiencies at the component level. This information is further used to compare the Allam cycle design with other designs incorporating CO_2 separation, in order to identify the main differences. However, based on the identified inefficiencies in the proposed cycle design, important aspects for further research and development are identified.

1 Introduction

Future outlooks by the IEA [1] and the U.S. Department of Energy [2] regarding the worldwide primary energy usage show an increasing demand and change in resources for the forthcoming years. While the relative share of oil – as the main source of energy – is probably going to decline between the years 1990 and 2040, the relative share of coal and natural gas is increasing [1,2] due to extensive secured resources and global availability. The projections further indicate that coal and natural gas will remain two of the world's main sources of energy regarding power generation.

Due to the increasing industrial competition and more stringent governmental policies worldwide, the demand for new, efficient and economically competitive processes is high, especially regarding the emission of pollutants [3,4]. As a result, an enormous effort is made in the research and development concerning the clean utilization of coal and natural gas by improving traditional and developing new technologies [3–13].

For the research and development of future power generation technologies, the feature of carbon dioxide (CO_2) capture is essential. In general, three different approaches can be used for CO_2 capture from fossil fuel based power generation: post-combustion, pre-combustion, and oxy-combustion [14, 15]. All three approaches differ greatly in technological implementation. In post-combustion capture, the CO_2 is captured from the flue gas which is dilute in CO_2 and at ambient pressure. In contrast, pre-combustion technology removes the CO_2 from a synthesis gas derived by gasification of coal or reforming of natural gas. Thus, the synthesis gas is concentrated in CO_2 and at high pressure. The synthesis gas is then burned in a gas turbine for power generation. Another possibility is the application of oxy-combustion technology, where the fuel is burned in a pure oxygen environment stream containing almost no nitrogen. As a result, the flue gas has a very high CO_2 concentration thus greatly facilitating its removal. Even though the different technological approaches have their decisive advantages, the inherent efficiency penalties due to thermodynamic limitations and the high additional costs of carbon capture equipment make it increasingly challenging to implement any of the three different technologies [4,15–19] when used with traditional power generation systems.

In this context it has to be mentioned that a carbon capture, utilization and storage (CCUS) [9, 20] pathway is a means to utilize CO_2 in the production of valuable outputs thus generating a revenue that is beneficial for offsetting the costs associated with carbon capture. From this point of view it is clear that new and innovative power cycles are required.

While commercial post- and pre-combustion CO_2 capture technologies are readily available, oxy-combustion technology is still under rapid development but not commercially available yet. Nevertheless, the main technologies for oxy-combustion, air separation for oxygen production and pure oxygen combustion itself, can be considered as state-of-the-art today [4].

Although oxy-combustion technology can be used with all kinds of fuels [4, 8, 21], the application in natural-gas-fired power plants using oxy-combustion turbines is highly attractive substituting air as the working fluid with carbon dioxide (CO_2) or water (H_2O), respectively. By eliminating the nitrogen, the combustion of natural gas with pure oxygen generates a flue gas that consists mainly of CO_2 and H_2O. This mixture can be easily separated by condensation thus simplifying the CO_2 capture step. However, an additional oxygen production unit and a large recycle stream to moderate the combustion temperature are required thus resulting in semi-closed cycle designs. Such gas turbine configurations differ greatly from conventional designs as the physical properties of CO_2 and H_2O are different from air. This is particularly important in the development of the required turbomachinery.

In the literature, different semi-closed cycles using CO_2 or H_2O as the main working fluid have been proposed. The conditions (temperature and pressure) of the working fluids differ greatly between the cycles. The designs include semi-closed oxy-combustion combined cycles (SCOC-CC) that closely resembles a conventional combined cycle but with CO_2 as the main working fluid [21–26]. Another proposed cycle by Clean Energy Systems (CES cycle) uses H_2O as the main working fluid in a subcritical cycle [24, 25, 27, 28]. Another subcritical cycle that has been proposed with either CO_2 or H_2O as the combustion temperature moderator is the Graz cycle [29–

32]. Furthermore, the CO_2-based MATIANT cycle has been proposed using different supercritical and subcritical configurations [23, 33–36]. All the different cycles are employing a conventional cryogenic air separation unit (ASU) for oxygen production.

In contrast, another option for oxygen production by employing high temperature oxygen transport membranes (OTM) has been proposed in the the AZEP [37,38]) and ZEITMOP [39,40] designs. Other options for oxy-combustion include Chemical Looping Combustion (CLC) [41] and hybridized solid oxide fuel cell and gas turbine systems (SOFC-GT) [42,43].

However, despite the innovative nature of the oxy-combustion systems, all cycles show a large efficiency penalty compared to a conventional natural-gas-fired combined cycle (NGCC) power plant as large auxiliary power requirements for oxygen production are present. With efficiencies in the range of 45-52.5 % [21,24,25,44–46], the processes will remain economically not feasible in the near future thus prohibiting their commercialization. Furthermore, technologies like oxygen transport membranes, chemical looping combustion and solid oxide fuel cells are currently not at a level of development that would allow them to become commercially available in the near future [8].

In the case of the semi-closed cycles, the auxiliary power demand for the cryogenic air separation unit (ASU) reduces the cycle performance. However, the recently proposed Allam cycle [47, 48] recovers heat from the ASU into the main power cycle to boost the overall cycle's efficiency. Combined with features such as an efficient compression/pumping configuration for CO_2 compression, and a high temperature heat recovery, a very high cycle efficiency of 59 % (LHV) is reported for natural gas operation. This figure is even higher than the baseline efficiency used by the U.S. Department of Energy National Energy Technology Laboratory's baseline efficiency [10] for conventional NGCCs.

As not much information is available for the Allam cycle [47,48], the already published data is used in the following analyses to evaluate the configuration of the natural gas oxy-combustion Allam cycle. Furthermore, exergy analyses [49,50] are used to identify the location, the magnitude, and the sources of the thermodynamic inefficiencies that occur within the process. In particular, the combination of exergy and sensitivity analyses provide a systematic framework to thoroughly understand [51–53] the process for further research and development.

2 System Description

For a rigorous analysis, the Allam cycle is first characterized considering its main features. Subsequently, the simulation model is described and the assumptions are specified.

2.1 Allam Cycle Features

In general, the Allam cycle can be classified as a single, highly recuperated, high-pressure and high-temperature but low-pressure-ratio gas turbine cycle using CO_2 as the main working fluid [47, 48]. The process diagram of the Allam cycle is shown in Figure 1 and resembles a typical oxy-combustion process [16].

Pressurized natural gas reacts with high-pressure pure oxygen that is generated in a cryogenic air separation unit (ASU). A high-pressure carbon dioxide recycle stream is used to moderate the combustion temperature. The hot gas is subsequently expanded in a turbine that is coupled to a generator. The single turbine has an inlet pressure between 200 bar and 400 bar and a pressure ratio of 6-12. The recuperator further cools the gas by reheating the recycle CO_2 stream to about 750 °C.

Afterwards, water is separated by condensation and the CO_2 stream is then again repressurized by compression and pumping and reheated in the recuperator. However, due to imbalances

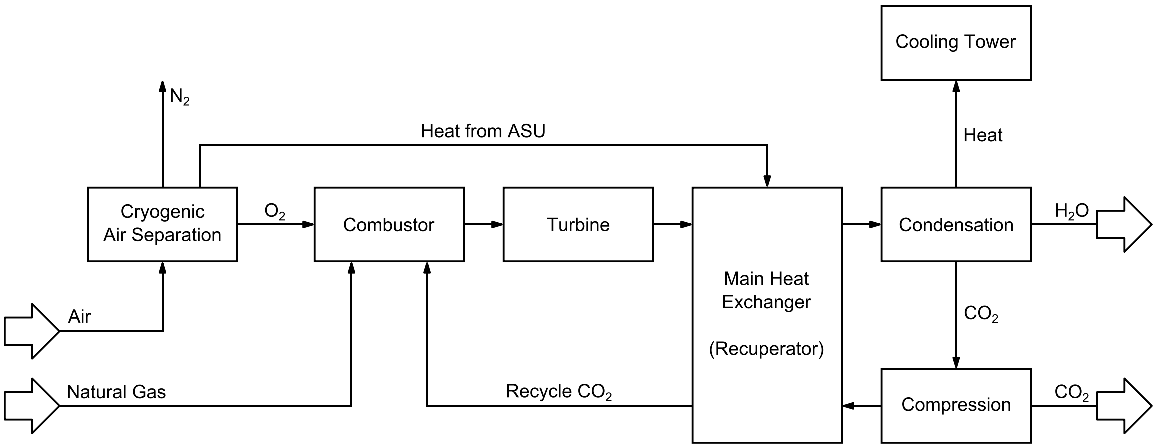

Figure 1: Process diagram of the Allam cycle

Table 1: Environment conditions used for the simulation based on [58]

Site Conditions		Air Composition	
Model	Midwest ISO	Nitrogen (N_2)	0.7732 mol/mol
Ambient Pressure	1.01325 bar	Oxygen (O_2)	0.2074 mol/mol
Ambient Dry Bulb Temperature	15.0 °C	Argon (Ar)	0.0091 mol/mol
Ambient Wet Bulb Temperature	10.8 °C	Carbon Dioxide (CO_2)	0.0003 mol/mol
Relative Humidity	60 %	Water (H_2O)	0.0100 mol/mol
Cooling Water Temperature	15.6 °C	Molar Mass	28.854 kg/kmol

in the thermodynamic properties between the hot gases leaving the turbine and the recycled CO_2, a low-temperature heat input is required where waste heat from the ASU can be effectively utilized. Instead of being rejected to the environment, it thus contributes to a higher efficiency of the overall cycle.

As the combustion of hydrocarbons introduces CO_2 into the power cycle, a small amount of it has to be constantly removed from the process. As the CO_2 content of the purge stream is very high ($\approx 97\,\%$), it can be either directly fed to a pipeline or further purified, depending on the prospective application [20, 54].

2.2 Description of the Simulation Model

As the Allam cycle is currently being commercialized by a consortium of NET Power, CB&I, Toshiba and Exelon, only limited information is publicly available [46–48, 55, 56]. In order to be able to benchmark the Allam cycle and to obtain comparable simulation results [24, 57], best practice guidelines are used in addition to the available process data.

The synthesized process flowsheet based on the data that is publicly available [46–48, 55] is shown in Figure 2.

The environmental conditions as well as the air composition is specified according to [58] and is shown in Table 1. Furthermore, the composition of the natural gas supplied to the power cycle is set using the average data that is given by [10, 59]. The molar composition as well as the supply temperature and pressure are shown in Table 2.

As the scope of this study is to analyze the Allam cycle in detail, the auxiliary air separation unit (ASU), the CO_2 purification and compression as well as the cooling tower are modeled by black boxes. A specific power demand is specified for each unit in order to calculate the power that has to be supplied by the main cycle.

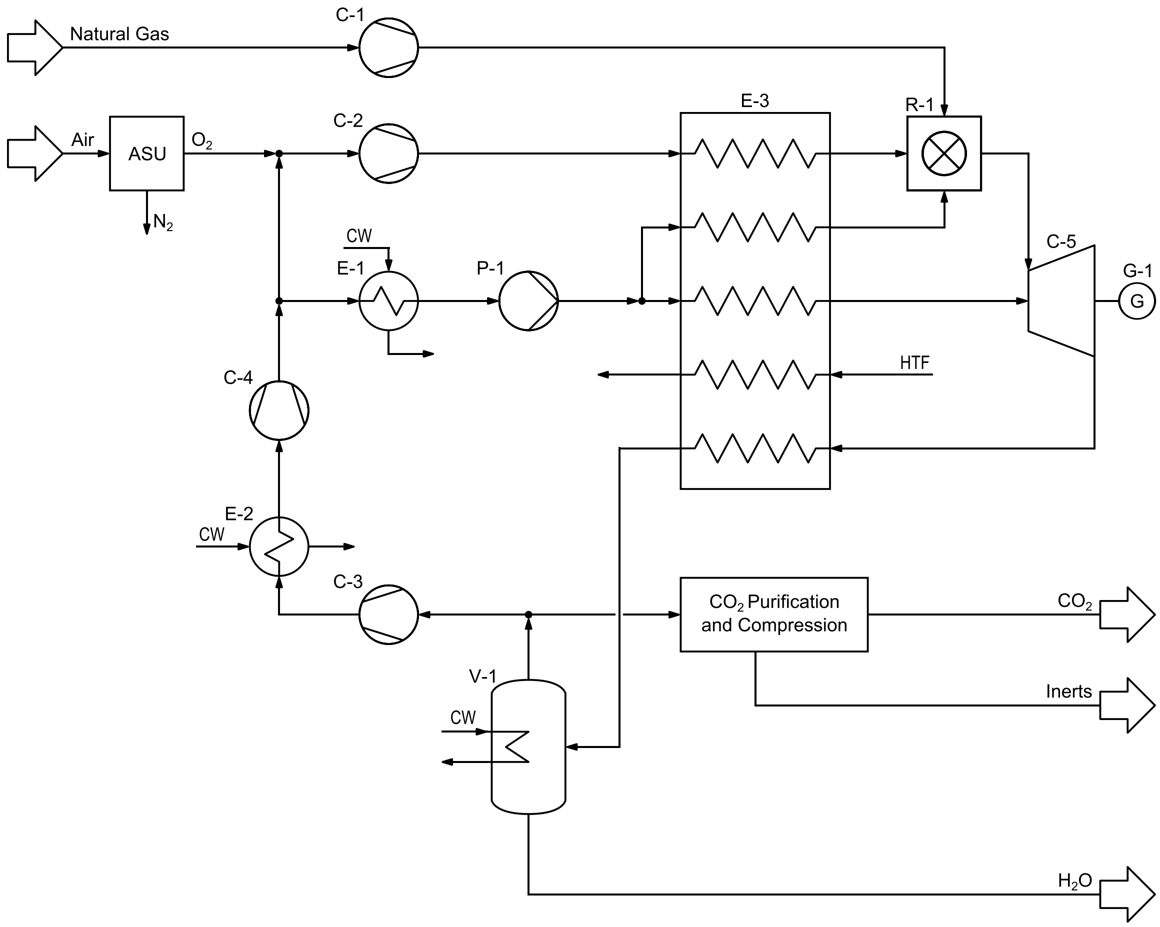

Figure 2: Flowsheet of the Allam cycle simulation model

The pressure of the natural gas that is supplied by pipeline is raised in compressor C-1 before entering the combustor R-1. To provide the required oxygen at specified purity (99.5 %, [47]), air from the environment is separated in a cryogenic ASU. The oxygen is then mixed with a part of the CO_2 recycle [47] and further compressed to the high pressure level using compressor C-2. Afterwards the stream is preheated to about 750 °C [47] in the recuperator E-3 before being fed to the combustor where it reacts with natural gas. The recuperator is a compact multi-stream plate-fin type [47]. Using this type of heat exchanger it is possible to achieve small pinch point temperature differences of 1-5 K [60, 61].

As the combustor and turbine have to be cooled, the other part of the CO_2 recycle is first cooled to near-ambient temperature [47]. The resulting high density enables the use of pump P-1 to raise its pressure to the required level. Afterwards, the stream is split into two fractions before both are also preheated in the recuperator E-3. One fraction is used for combustor cooling while the other is employed for turbine cooling purposes. The splitting of cooling flows between combustor and turbine [62] provides a higher level of detail than a simple ISO model [63]. However, the turbine is parameterized using the ISO conditions given in [47].

The hot gases from the combustor are subsequently expanded in turbine C-5 for power generation. The turbine is coupled to the generator G-1. The exiting stream is then fed to the recuperator (E-3) where it is cooled down to a temperature in the range of 60-70 °C [47]. In order to overcome the imbalance of thermodynamic properties between the recycled CO_2 and the hot gases that occurs in the low-temperature region, an additional heat source has to be supplied. For that reason,

Table 2: Natural gas supply specification according to [10, 59]

Methane (CH_4)	0.931 mol/mol	Lower Heating Value (LHV)	47457 kJ/kg
Ethane (C_2H_6)	0.032 mol/mol	Higher Heating Value (HHV)	52581 kJ/kg
Propane (C_3H_8)	0.007 mol/mol	Temperature	38.0 °C
n-Butane (C_4H_{10})	0.004 mol/mol	Pressure	30 bar
Carbon Dioxide (CO_2)	0.010 mol/mol		
Nitrogen (N_2)	0.016 mol/mol		

a heat transfer fluid (HTF) is used that transfers a portion of the heat of compression from the ASU at the required temperature level.

After leaving the recuperator, the flue gases are further cooled down to condense a large portion of the water. The separator V-1 is used to remove the liquid water. The remaining vapor, that contains about 97 % CO_2, is then compressed using a two-stage inter-cooled compressor (C-3, C-4, E-2) [47] thus closing the power cycle. Nevertheless, a small portion of the vapor is purged from the cycle. This purge stream is subsequently treated in the CO_2 purification and compression unit. The scope of this unit is to provide the CO_2 concentration and pressure to meet the specifications for further usage in possible storage, enhanced oil recovery (EOR) or chemical feedstock applications [54, 64].

In order to obtain relevant results from the simulation model, it has to be parameterized defining the technology level and the operating conditions of the plant [57].

The simulations were conducted using the Aspen Plus modeling software. The thermodynamic properties are calculated using the Peng-Robinson equation of state for general property calculations whereas the Lee-Kesler-Plöcker equation is used for the CO_2 recompression modeling as recommended in [58]. Due to the fact that a detailed analysis of the Allam cycle is not available, and in order to consider the influence of the assumed technological level, three different cases are investigated. By defining a base, low and high efficiency case, the influence of the main assumptions is revealed. The component parameters that are employed in the simulations are given in Table 3. By taking uncertainty for some parameters into account, it is possible to estimate their impact on the cycle's performance using general benchmark parameters. Based on the results of the case studies, subsequent sensitivity analyses are conducted by variation of the performance of main components.

3 Methodology

In order to analyze the Allam cycle, a conventional thermodynamic analysis as well as an exergy analysis are used. The first methodology provides only general information about the thermodynamic efficiency, whereas the latter provides detailed data about the main sources of the cycle's inefficiencies.

3.1 Thermodynamic Analysis

By employing the Aspen Plus modeling software, the energy and mass balances for the simulation model are solved. Based on the methodology of general thermodynamic studies [57, 58], the efficiency of the overall power cycle is evaluated by

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

However, in order to provide a comparable benchmark, the selected modeling framework has to be taken into account [57, 58].

Table 3: Simulation parameters for the analysis of the Allam cycle

Parameter		Base Case	Variation	Reference
Turbine Inlet Temperature (ISO)	°C	1150	± 0	[47]
Turbine Inlet Pressure	bar	300	± 0	[47]
Turbine Pressure Ratio	–	0.1	± 0	[47]
Combustor R-1, Outlet Temperature	°C	1300	± 100	[46]
Combustor R-1, Pressure Drop	%	1.6	± 0	[47]
Combustor R-1, Heat Loss	%	1.0	± 0	[58]
Oxygen Purity	%	99.5	± 0	[47]
Excess Oxygen	%	2.0	± 0	[24]
O_2 Fraction (Molar) after Dilution	%	22.5	± 7.5	[47]
Pump P-1, Efficiency	%	75	± 5	[58]
Pump P-1, Mechanical Efficiency	%	98	± 0	[58]
Compressor C-1-C-4, Polytropic Efficiency	%	80	± 5	[58]
Compressor C-1-C-4, Mechanical Efficiency	%	98	± 0	[58]
Motor Efficency	%	97	± 0	[58]
Generator G-1, Efficiency	%	99	± 0	[58]
Heat Exchanger E-1, E-2, Pinch Temperature Difference	K	7.5	± 2.5	[58]
Heat Exchanger E-1, E-2, Pressure Drop (gas)	%	2	± 0	[58]
Heat Exchanger E-1, E-2, Pressure Drop (liquid)	%	4	± 0	[58]
Heat Exchanger E-3, Pinch Temperature Difference	K	3	± 2	[60, 61]
Heat Exchanger E-3, Pressure Drop	%	2	± 0	[58]
Separator V-1, Pressure Drop	%	2	± 0	[58]
Cooling Water Range	K	11	± 0	[58]
Cooling Tower Fan Power Demand	W/m^3_{Air}	197,5	± 0	[65]
ASU, Specific Power Demand	kWh/kg_{O_2}	250	± 50	[66, 67]
CO_2 Purification, Specific Power Demand	kWh/kg_{CO_2}	50	± 25	[67]

Table 4: Reference environment used for the exergetic analysis

Ambient Temperature	15 °C
Ambient Pressure	1.01325 bar
Chemical exergy model	Szargut [68]

3.2 Exergy Analysis

For the analysis of systems and their constituent components, the exergy analysis provides a useful framework to uncover the real thermodynamic inefficiencies. Exergy is defined as the maximum theoretical useful work obtainable from a thermal system as it is brought into thermodynamic equilibrium with the environment while interacting with the environment only. As exergy is destroyed in any real process, it provides information which is simply not available through conventional thermodynamic analyses. The main causes of exergy destruction are generally related to chemical reactions, heat transfer, fluid friction, and mixing of streams at different temperature, pressure and composition. The methodology and capabilities of a conventional exergy analysis are well documented [49, 50, 68, 69].

Parameters obtained from exergy analysis are used to characterize the thermodynamic performance of a system and its different components from an unbiased point of view. This is done by introducing a reference environment, that is shown in Table 4 and corresponds with the modeling framework used for benchmarking purposes.

To provide more details, the exergy is further split into its chemical and physical parts [50] for analyzing the power cycle.

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

Considering that the process operates at steady-state conditions, the exergy balance for component k is used to calculate its exergy destruction $\dot{E}_{\mathrm{D},k}$. The balance contains different terms for exergy transport by heat transfer $\dot{E}_{\mathrm{q}}$ and power $\dot{W}$ as well as with transport of matter at the inlet $\dot{E}_i$ and exit $\dot{E}_e$ of component k.

$$\sum_j \dot{E}_{\mathrm{q},j,k} + \dot{W}_k + \sum_i \dot{E}_{i,k} - \sum_e \dot{E}_{e,k} - \dot{E}_{\mathrm{D},k} = 0 \tag{3}$$

The exergy destruction $\dot{E}_{\mathrm{D},k}$ can further be used to determine the real thermodynamic performance of a system's component k as well as for the overall system. The exergetic efficiencies of the k-th component ε_k is calculated using the following equation:

$$\varepsilon_k = \frac{\dot{E}_{\mathrm{P},k}}{\dot{E}_{\mathrm{F},k}} = 1 - \frac{\dot{E}_{\mathrm{D},k}}{\dot{E}_{\mathrm{F},k}} \tag{4}$$

Besides the exergy destruction that occurs within the system components, the analysis of the overall system also has to consider possible exergy losses to the environment.

$$\varepsilon_{\mathrm{tot}} = \frac{\dot{E}_{\mathrm{P,tot}}}{\dot{E}_{\mathrm{F,tot}}} = 1 - \frac{\sum_k \dot{E}_{\mathrm{D},k} + \dot{E}_{\mathrm{L,tot}}}{\dot{E}_{\mathrm{F,tot}}} \tag{5}$$

The exergy rates $\dot{E}_{\mathrm{P},k}$ and $\dot{E}_{\mathrm{F},k}$ determine the respective exergy rates of product and fuel of the examined component k. Furthermore, $\dot{E}_{\mathrm{P,tot}}$ and $\dot{E}_{\mathrm{F,tot}}$ are the exergy rates of product and fuel of the overall system. By calculating the sum of the components exergy destruction $\dot{E}_{\mathrm{D},k}$ as well as the exergy losses from the overall system $\dot{E}_{\mathrm{L,tot}}$, the system's real thermodynamic inefficiencies are determined.

In order to assign the appropriate exergy rates of product and fuel to the different components and the overall plant, the SPECO framework [70] is used. It provides a consistent methodology for defining the real thermodynamic efficiencies of a component and the overall system.

However, with the exergy destruction ratio $y_{\mathrm{D},k}$ another useful parameter can be defined. It quantifies the contribution of the exergy destruction within component k to the reduction of the overall exergetic efficiency $\varepsilon_{\mathrm{tot}}$ of the system.

$$y_{\mathrm{D},k} = \frac{\dot{E}_{\mathrm{D},k}}{\dot{E}_{\mathrm{F,tot}}} \tag{6}$$

In general, the different exergy-related parameters can be used to analyze and subsequently improve any thermal system. The exergy analysis further provides a convenient framework for benchmarking and comparing different processes from an unbiased point of view.

4 Results

This section presents the findings of the main cycle analysis to show the consequences of the assumptions that were used for the parameterization of the simulation. Furthermore, the results of the thermodynamic and exergetic analyses are the starting point for discussing the importance of the main cycle components by conducting sensitivity analyses.

Table 5: Results of the simulations

Parameter		Simulation Study		
		Base Case	High Efficiency	Low Efficiency
Fuel Mass Flow Rate	[kg/s]	9.9	9.2	11.0
Gross Power	[MW]	292.1	280.7	307.3
Net Power	[MW]	250.0	250.0	250.0
Efficiency (LHV)	[%]	53.4	57.2	47.9

4.1 Cycle Analysis

The results of the base case as well as for the low- and high-efficiency simulation studies are shown in Table 5.

Even though the cycle efficiency of 53.4 % (LHV) for the base case is below the literature figure of 59 % (LHV) [47], it can be concluded that the cycle offers a high efficiency even if unfavorable assumptions are made. Taking into account that the high efficiency case already exhibits an efficiency of 57.2 % (LHV) it can be concluded that the higher efficiencies of the turbomachinery, that is used for the recycle CO_2 as well as a lower pinch temperature difference for the cooling heat exchangers, are highly beneficial concerning the cycle's performance. Furthermore, the lower ASU and CO_2 stream purification and the compression power requirements are beneficial. This confirms the observations and discussions that are found in [46].

The efficiency of 47.9 % (LHV) that was found in the low efficiency case shows that the parameter uncertainty has a very large influence. Based on the results it can be concluded that a certain minimum turbomachinery efficiency and pinch temperature difference is needed in order to realize a high cycle efficiency.

Furthermore, the results of the low efficiency case show an interesting feature of the Allam cycle. It was found that the recuperator E-3 that is used for recuperation is highly dependent on the parameters of the recompression components. Figure 3 shows that the thermodynamic property imbalance requires an additional heat input in the temperature range between 200-300 °C. The comparison of the different cases shows that a lower efficiency of the recompression components moves the pinch point to higher temperatures.

The results of the three case studies further show that about 10-20 % of the generated power is consumed by auxiliary components.

However, the results of the exergy analysis (see Table 6) show that the main inefficiencies are actually found within the cycle. The exergetic efficiency of the base case is 51.3 %. This follows from the fact that the chemical exergy of the gaseous fuel is between its LHV and HHV. However, the figures of Table 6 indicate that the combustion reaction is the main source of irreversibilities of the system. Within the combustor, about 20 % of the supplied exergy is destroyed. Based on the absolute exergy destruction, the turbine and the recuperator rank second and third, respectively, even though their efficiencies are relatively high.

In addition, the recompression system of the recycle CO_2 has a considerable exergy destruction as it consumes about 6.3 % of the fuel exergy.

Compared to conventional gas turbine cycles [38], the distribution of the overall exergy destruction within the Allam cycle is different with the exergy destruction coefficient of the combustor being smaller, whereas the influence of the turbomachinery is stronger. This effect results from the different cycle designs.

In general, the combustion of fuels is highly irreversible [71, 72] thus resulting in the high exergy destruction that is generally found within the combustors of gas turbines. However, differences can be found considering the compression and expansion of the working fluid. The ther-

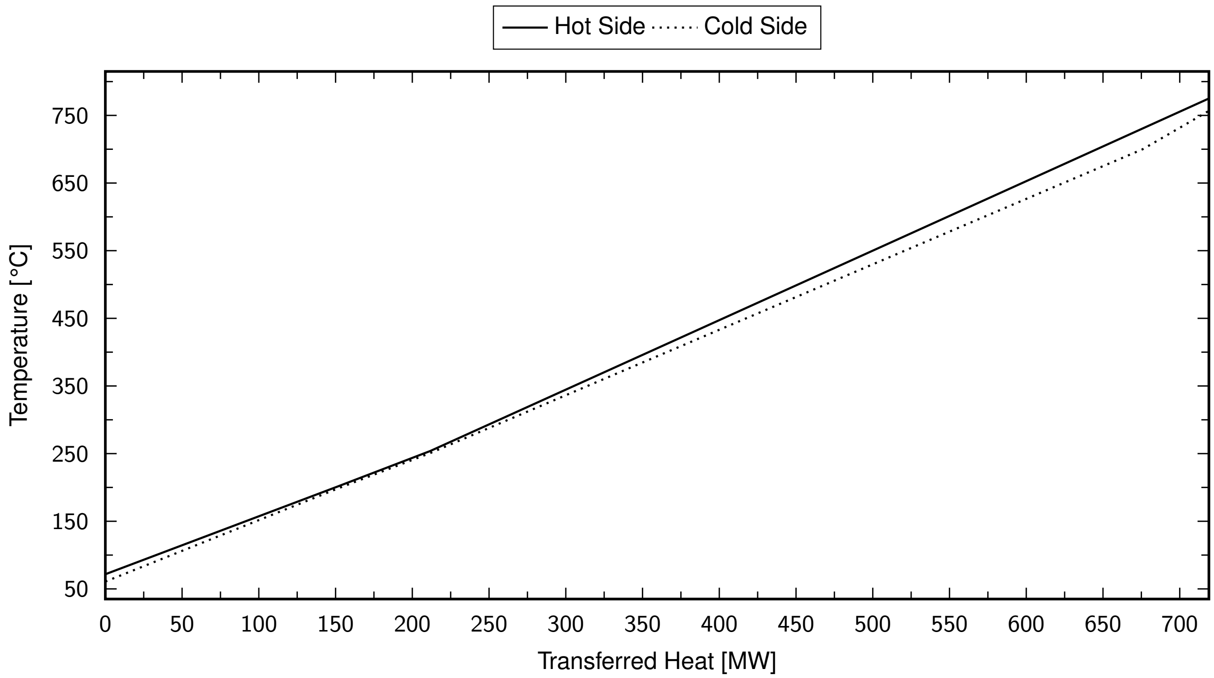

Figure 3: Composite curve of the recuperator

modynamic efficiency of these processes depends on the value of a thermodynamic parameter characterizing the respective component (e.g. isentropic, polytropic or exergetic efficiency). The design of the recompression system of the Allam cycle is more complex than that of a conventional gas turbine compressor.

The results of the cycle analysis show that the Allam cycle exhibits significant differences compared to conventional gas turbine cycles, based on the design using CO_2 as the working fluid, its recuperator and complex recompression system.

4.2 Sensitivity Analysis

The results of the thermodynamic and exergetic analyses have shown that the most important units are the combustor, turbine and recuperator as well as the recompression system. Based on this information, the main parameters of each unit are varied in order to show their influence on the cycle performance as well as the exergy destruction that occurs within other components.

4.2.1 Turbine Inlet Temperature

The turbine inlet temperature (ISO) is varied between 1100 and 1200 °C. The results are shown in Figure 4. It follows that for a difference of 50 K, the cycle efficiency changes by 1 %. The higher turbine inlet temperature improves the efficiency of combustor and turbine. Furthermore, whereas a higher temperature inlet temperature improves the efficiency of combustor R-1 and turbine C-5, there is only a small influence on other components.

4.2.2 Turbine Efficiency

The second varied parameter is the turbine isentropic efficiency, considering the same temperature inlet temperature (ISO). From the results in Figure 4 it can be concluded that the cycle efficiency changes by about 0.3 % when changing the turbine efficiency by 1 %. This is due to the

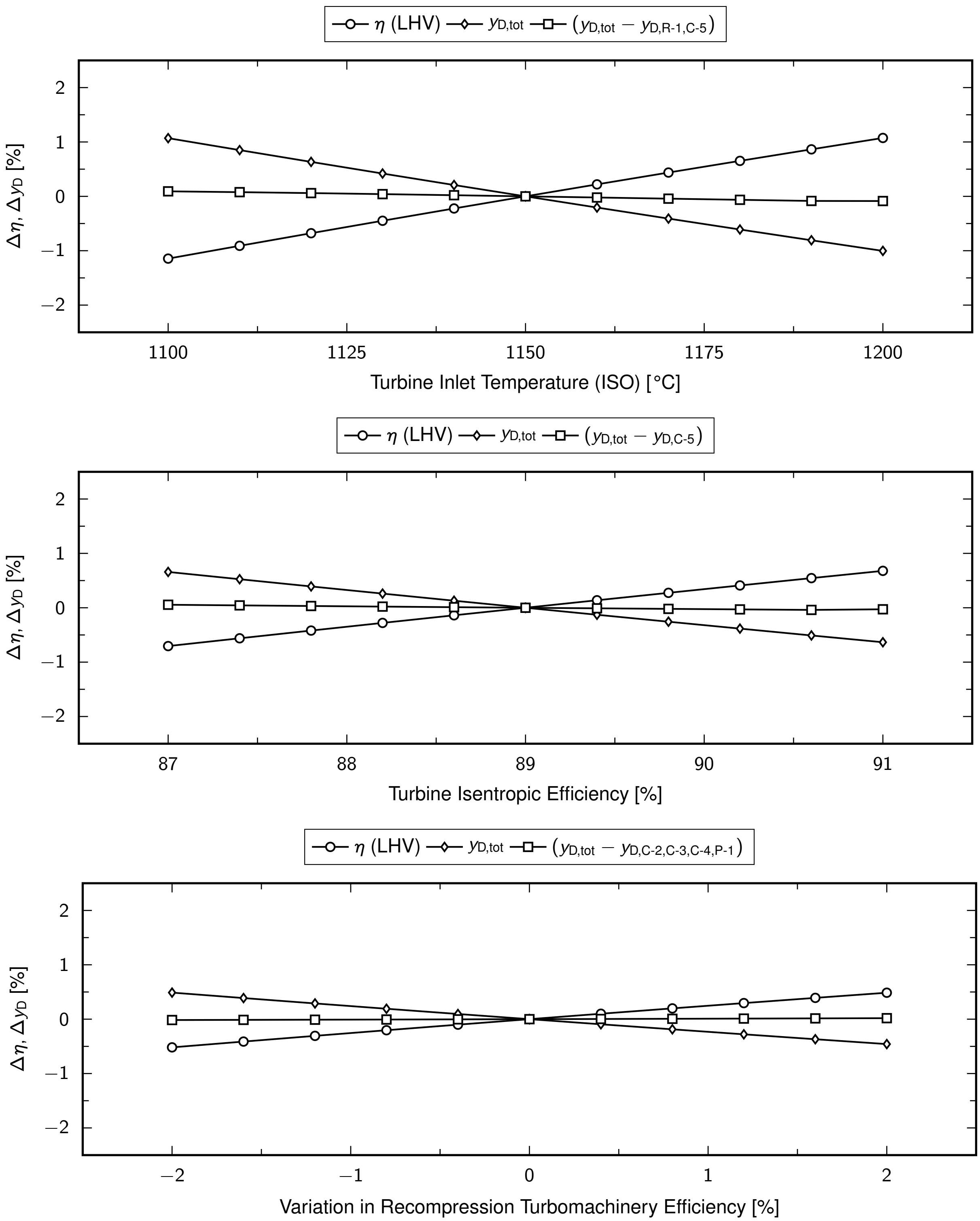

Figure 4: Sensitivity analysis for variations in turbine inlet temperature, turbine isentropic efficieny and the recompression turbomachinery efficiency. These diagrams show the cycle efficiency η and the exergy destruction coefficient of the overall cycle y_{tot} as well as the effect that the exergy destruction of the respective component(s) has on the exergy destruction coefficient of the other components.

Table 6: Results of the exergy analysis

Simulation	Base Case			High Efficiency			Low Efficiency		
	$\dot{E}_D$	ε	y_D	$\dot{E}_D$	ε	y_D	$\dot{E}_D$	ε	y_D
Component	[MW]	[%]	[%]	[MW]	[%]	[%]	[MW]	[%]	[%]
C-1	1.0	82.7	0.2	0.7	85.7	0.2	1.5	79.8	0.3
C-2	11.0	56.9	2.3	8.2	50.1	1.8	15.4	67.2	2.8
C-3	4.0	83.5	0.8	2.2	89.0	0.5	6.5	78.3	1.2
C-4	5.0	77.9	1.0	3.4	82.0	0.7	7.3	73.8	1.3
C-5	29.0	93.3	6.0	28.7	92.8	6.3	30.2	93.8	5.6
P-1	10.5	63.6	2.2	8.1	68.2	1.8	12.4	59.6	2.3
E-1	5.4	32.2	1.1	4.9	34.8	1.1	5.5	29.5	1.0
E-2	3.8	20.0	0.8	3.0	22.2	0.7	4.9	18.3	0.9
E-3	11.8	96.9	2.4	11.4	96.8	2.5	16.7	96.1	3.1
R-1	105.4	78.0	21.6	97.1	78.3	21.3	118.9	77.7	21.9
V-1	3.8	71.8	0.8	1.0	89.6	0.2	12.5	48.4	2.3
G-1	4.0	99.0	0.8	3.7	99.0	0.8	4.5	99.0	0.8
Auxiliary Units	19.7	53.1	4.0	11.2	63.5	2.5	30.6	46.6	5.6
Overall Cycle	214.4	51.3	44.0	183.6	54.9	40.4	266.9	46.0	49.1

fact that the turbine outlet temperature decreases thus limiting the ability to recuperate heat in heat exchanger E-3. This is also shown by the fact, that the exergy destruction coefficient decreases. However, the turbine efficiency has no significant influence on the exergy destruction coefficient of the other components.

4.2.3 Recompression Turbomachinery Efficiency

The CO_2 recompression system is of high importance to the cycle performance. In the sensitivity analysis, the efficiencies of the compressors C-2, C-3 and C-4 as well as the efficiency of pump P-1 are varied simultaneously. The cycle efficiency (see Figure 4) changes by 0.25 % for each percentage point change in the compressor and pump efficiency. Additionally, the analysis shows that even the exergy destruction coefficient of the overall system is reduced, the exergy destruction coefficients of the other components remain constant.

4.2.4 Oxygen Fraction after Dilution

Based on the comparison of the base and the high efficiency case, it can be concluded that a higher oxygen content after the dilution is beneficial to the cycle efficiency. This is further shown by the results of sensitivity analysis as the oxygen fraction is varied between 15-30 % (mol). In Figure 5 it is indeed shown that the cycle efficiency increases. This is due to the fact that a larger fraction of CO_2 is pumped instead of being compressed. The reduced power demand causes a rise in efficiency. However, by approaching a fraction of 30 % the increase in efficiency diminishes. The reason is the larger portion of CO_2 at low temperature limits the efficiency of the recuperator.

4.2.5 Heat Exchanger Pinch Temperature Difference

The heat exchanger pinch temperature differences are important to the recuperator (E-3) and the coolers (E-1, E-2, V-1) in the CO_2 recycle. The results of the sensitivity analyses are shown in Figure 5.

Varying the recuperator pinch temperature difference between 1-5 K, it is found that the cycle only benefits little from a further decrease in temperature difference. This is further shown by the

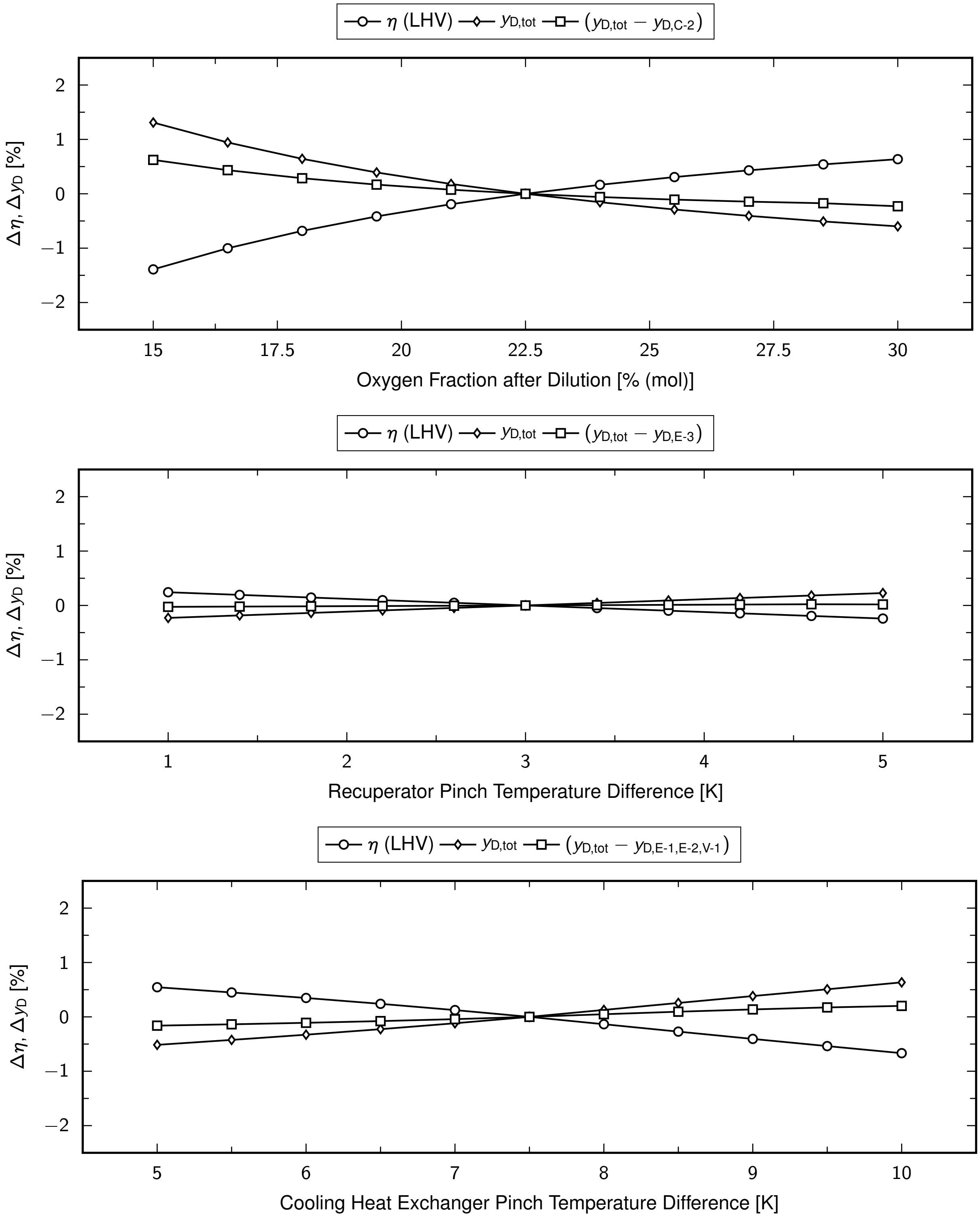

Figure 5: Sensitivity analysis for variations in oxygen fraction after dilution and pinch temperature difference in the recuperator and the coolers. These diagrams show the cycle efficiency η and the exergy destruction coefficient of the overall cycle y_{tot} as well as the effect that the exergy destruction of the respective component(s) has on the exergy destruction coefficient of the other components.

small decrease in the exergy destruction coefficient of the overall cycle. However, as the exergy destruction within the recuperator becomes smaller, the exergy destruction coefficients of the other components remain almost constant.

In contrast to the recuperator (E-3) that can realize very small temperature differences, the pinch temperature difference of the conventional heat exchangers (E-1, E-2, V-1) is larger. By varying the pinch temperature difference between the hot side and the cooling water between 5-10 K, it is found that a decrease of the temperature difference by 1 K leads to an increase in the cycle efficiency by about 0.25 %. Furthermore, it is shown that a smaller temperature difference increases the efficiency in other cycle components. The effect is mainly found in the CO_2 recompression system as the required pumping and compression power is reduced.

4.3 Discussion

The analyses of the Allam cycle has shown that the cycle is a highly innovative configuration as it features a high efficiency even at reasonable component parameterizations. By further improving the main cycle parameters its efficiency can further be increased.

The configuration of the Allam cycle is insofar special as the exergy analyses suggest that the recuperator effectively decouples the high and low pressure levels. The consequences of this effect have to be studied in more detail in the future.

Comparing the Allam cycle with similar oxy-combustion cycles [38] it is found that the efficiencies of the main system components like combustor, turbine and CO_2 recompression are similar. The same is found for the exergy destruction coefficients, respectively. Nevertheless, as the Allam cycle has less components, the simplicity results in a higher overall efficiency than for comparable cycles. Furthermore, it can be concluded that the Allam cycle is considerably different compared to conventional gas turbine cycles from a thermodynamic perspective.

Regarding the improvement of the Allam cycle, it is shown that the combustor, turbine, recuperator and CO_2 recycle recompression show a certain potential. To be able to quantify this potentials, the simulation black boxes of the auxiliary units for air separation and CO_2 purification and compression have to be replaced by detailed models.

5 Conclusion

In the present paper a model of the Allam cycle was synthesized and subsequently analyzed at the component level. The different analyses have shown that the process has a considerable higher efficiency compared to other suggested oxy-combustion cycles and exhibits a further improvement potential. The Allam cycle has several distinct features that are advantageous considering oxy-combustion cycles.

The information that was obtained from this study provides a good starting point for further studies of the Allam cycle and its features. To be able to understand the cycle even better, subsequent analyses preferably using exergy-based methods should be conducted. In particular, the exergoeconomic and advanced exergy analysis [38, 69] frameworks have the potential to provide additional useful information.

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