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PRELIMINARY COST AND PERFORMANCE RESULTS FOR A NATURAL GAS-FIRED DIRECT SCO_2 POWER PLANT

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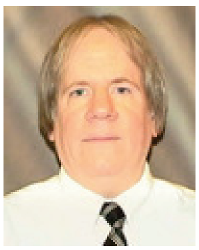

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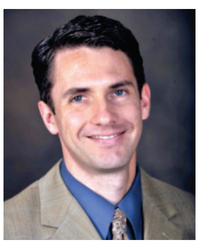

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ABSTRACT

The direct sCO_2 cycle holds significant promise as an efficient, cost-effective method of natural gas-fueled power production with the capability for carbon capture and storage (CCS). This study reports on a cost and performance baseline that has been developed at the National Energy Technology Laboratory (NETL) for a natural gas direct sCO_2 plant, enabling a preliminary assessment of this technology's ability to compete against a natural gas combined cycle (NGCC) plant with CCS. Further, the baseline direct sCO_2 model will be used as a basis for dynamic plant modeling and ongoing techno-economic improvement in the plant's design.

The steady state plant model includes turbine cooling flows that are based on a semi-empirical cooling effectiveness model, allowing for calculation of required cooling mass flows as a function of its temperature and those in the turbine's hot gas path. The model also includes detailed thermal integration with the air separation unit (ASU) and oxygen compression train, with the goal of minimizing temperature differences throughout the sCO_2 recuperation train. Discussion of the performance of the baseline plant design is followed by details on the costing methodology for the sCO_2 cycle components, the overall plant capital cost, and the resulting cost of electricity (COE). These results are compared to the efficiency and COE of a reference NGCC plant with CCS, as well as to other direct sCO_2 techno-economic analyses in the literature. Finally, preliminary analyses are performed to assess the effects of incomplete combustion on the overall plant performance.

INTRODUCTION

The direct supercritical CO_2 (sCO_2) cycle, also referred to as the Allam cycle after inventor Rodney Allam, is being developed for commercialization by 8 Rivers Capital. [1] [2] The cycle can be fueled by either gasified coal, which is under development by 8 Rivers and EERC [3], or natural gas, which will shortly be demonstrated in a 25 MWe pilot plant under the NET Power umbrella, which is a joint venture of 8 Rivers, Toshiba, Exelon, and CB&I [4].

The main strengths of the direct sCO_2 cycle lie in its inherent ability to capture carbon dioxide at high pressures typical of CO_2 storage systems, and in cycle designs that are tailored to exploit the properties of carbon dioxide in its supercritical state. In particular, direct sCO_2 cycles operate most efficiently at high cycle pressures between about 30 and 300 bars, very near to the critical pressure of CO_2 (73.7 bar), which allows for very compact turbomachinery and compact microchannel heat exchanger designs that can provide the large thermal recuperation duty required for efficient cycle operation. Further, the cold temperatures in the cycle are near the CO_2 critical temperature of 31 °C, such that cycle compression power is reduced by selecting high density compressor inlet conditions near the CO_2 critical point.

A schematic of a simple direct sCO_2 power cycle is shown in Figure 1 below. In this cycle, natural gas is burned with oxygen in a dilute sCO_2 environment, with the combustion products powering a turbine to generate electricity. The remaining thermal energy in the turbine exhaust is recovered in a recuperator to heat the CO_2 diluent return flow to the combustor. After recuperation, water is condensed out of the product stream, and a portion of the remaining, primarily CO_2 product, is drawn from the cycle for further purification and storage. The remaining fluid is compressed to a pressure near the CO_2 critical pressure, cooled, and pumped to the maximum cycle pressure for preheating in the recuperator and return to the combustor.

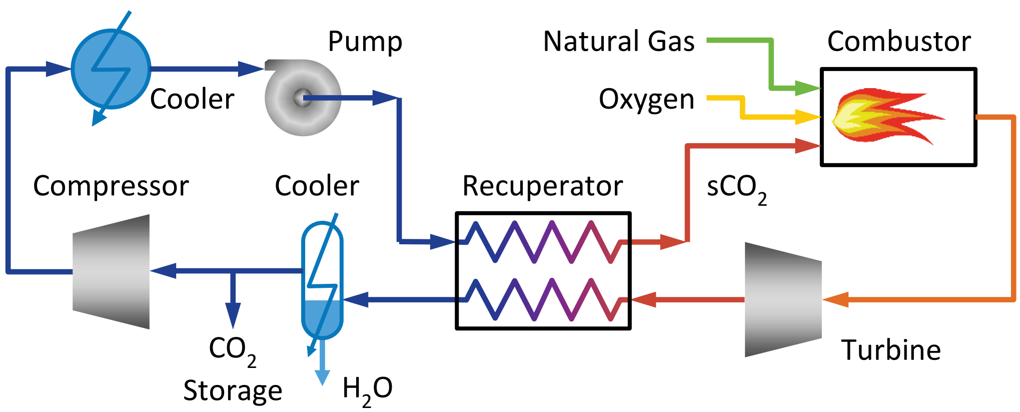

Figure 1: Schematic of a natural-gas fueled direct sCO_2 power cycle.

In addition to the studies noted above, other works can be found in the literature that use this same general cycle architecture. Foster Wheeler analyzed the direct sCO_2 cycle and compared its performance and cost with other oxy-turbine power cycles under European [5] and American [6] siting and economic assumptions. Studies by Scaccabarozzzi et al have performed sensitivity analyses and cycle optimization on the cycle originally modeled by Foster Wheeler [7], while a follow-on study expands upon the optimization variables and analyzes the cycle's part-load performance [8]. Southwest Research Institute has evaluated alternative natural gas-fired direct sCO_2 cycles with reported plant thermal efficiencies that are comparable to other studies. [9] Several studies have investigated the performance of coal-fueled direct sCO_2 cycles, and have generally found their thermal efficiency and economic performance to exceed integrated gasification combined cycle (IGCC) plants with carbon capture and storage (CCS). [10] – [15]

The influence of gaseous impurities on direct sCO_2 cycle performance has been demonstrated in studies by EPRI [12] and Foster Wheeler/IEAGHG [5], where the impact of oxygen purity from the air separation unit (ASU) on plant efficiency is investigated. In general, it was found that cases with 99.5% oxygen purity performed better than those with oxygen purities of 95% or 97%, despite the increased ASU auxiliary power required to produce higher-purity oxygen. This occurs because the lower-purity ASU includes additional argon and nitrogen in the oxygen stream, and the presence of these impurities within the sCO_2 cycle significantly increases the compression power requirements by lowering the bulk fluid density in the compression train. [12] Any gaseous species in the working fluid that do not condense out of the sCO_2 cycle with the water are therefore recirculated in the cycle with the sCO_2, and act to reduce cycle and plant efficiency. In addition to the oxygen supply, these impurities can enter the sCO_2 cycle through a variety of mechanisms: through the carrier gas used to feed dry coal to a gasifier [12], in the pipeline natural gas supply [5] [16], and through the combustion process itself [17].

The combustion process can introduce process impurities in a number of ways. First and foremost, non-stoichiometric combustion will affect the product gas composition by adding either excess oxygen or unburnt fuel, typically in the form of CO. In addition, some of these impurities may arise naturally if thermal equilibrium conditions are established at high flame temperatures within the combustion chamber, followed by rapid quenching at the combustor exit and through the turbine that prevents recombination reactions from proceeding. Theoretical [18], experimental [19] and numerical studies [20] [21] show that this may be the case in direct sCO_2 power cycles, where CO_2 dissociates into CO and O_2 at high combustion temperatures. In addition, combustor designs with poor fuel/oxygen mixing, rapid CO_2 diluent quenching, and/or insufficient residence time to complete combustion may also have high outlet concentrations of CO, O_2, and other species. The effects of incomplete combustion products on the performance of direct sCO_2 power cycles has not yet been assessed in the literature to date.

The objective of the present study is to develop a cost and performance baseline for a natural gas-fueled, direct-fired sCO_2 power plant, and to analyze the sensitivity of its net thermal efficiency and cost indicators to variations in select operating parameter and cost assumptions. The study is similar to prior techno-economic analyses by the authors on coal-fueled direct sCO_2 power plants thermally integrated with the coal gasification process. [10] [11] Relative to a reference IGCC plant with CCS and the same gasifier [22], the direct sCO_2 power cycles in these studies were shown to improve the net plant efficiency from 31.2% to 40.6% (HHV), and provided a 20% reduction in cost of electricity (COE). [11] Using a similar methodology and assumptions, the present study will investigate the techno-economic performance of a natural gas-fueled direct sCO_2 plant relative to a reference natural gas combined cycle (NGCC) plant with CCS. [23] The paper discusses the methodology used in the study, followed by a discussion of the performance and economic analysis results for the baseline direct sCO_2 case, and an analysis of the impacts of incomplete combustion on cycle performance. Finally, the results are compared to the results from other natural gas direct sCO_2 studies in the literature.

PLANT CONFIGURATIONS

Reference NGCC Plant

A simplified block flow diagram for an NGCC process with carbon capture is shown in Figure 2. This is used as a reference case for the current study, and is described in the Bituminous Baseline Study Rev 3 as Case B31B. [23]

The NGCC design is based on a market-ready technology that is assumed to be commercially available. The design consists of two state-of-the-art 2013 F-class combustion turbine generators, two heat recovery steam generators (HRSGs), and one steam turbine generator in a multi-shaft 2x2x1 configuration. The Rankine cycle portion of the design uses a single reheat 16.5 MPa/566°C/566°C (2,400 psig/1,050°F/1,050°F) sub-critical steam cycle. The steam turbine consists of an HP section, an IP section, and a double-flow LP section, all connected to the generator by a common shaft.

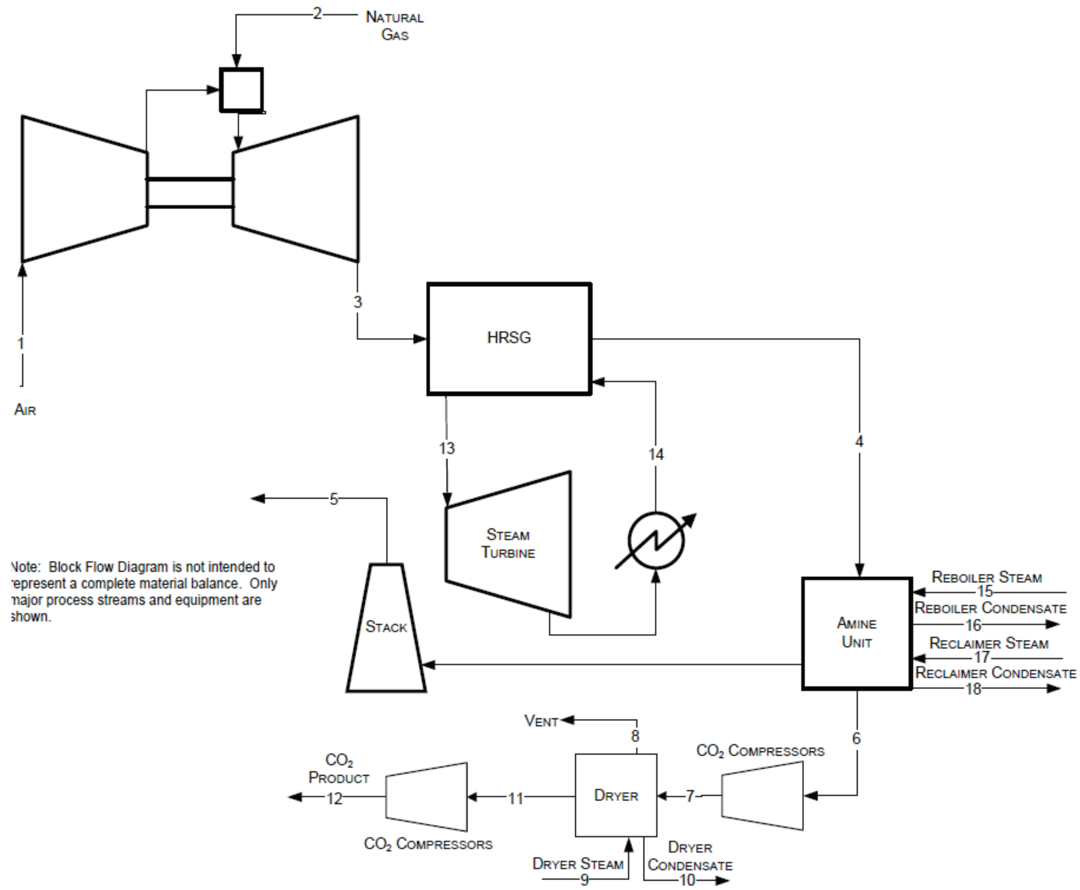

Figure 2: NGCC reference plant block flow diagram

The NGCC design includes a state-of-the-art amine-based unit to remove 90% of the CO_2 in the flue gas exiting the HRSG, dry it, and compress it to a supercritical condition. The CO_2 absorption/stripping/solvent reclamation process for this case is based on the Shell Cansolv system. [24] NETL analysis of this reference case yields a plant efficiency of 45.7% on an HHV basis, which includes an efficiency penalty of 5.8 percentage points to add CO_2 capture to the baseline NGCC case. [23] This efficiency reduction is caused primarily by the auxiliary loads of the capture system and CO_2 compression, as well as the significantly increased cooling water requirement that increases the auxiliary load of the cooling water pumps and the cooling tower fan. CO_2 capture results in a 31 MW increase in auxiliary load compared to the non-capture case.

Baseline sCO_2 Plant Configuration

Figure 3 shows a block flow diagram for the baseline natural gas-fired sCO_2 power plant modeled in this study. The evolution and state point details of this cycle design are documented in Ref. [25]. The cycle has been designed to maximize thermal recuperation and preheating of the reactants supplied to the combustor to yield a high plant thermal efficiency. The cycle's purge stream is purified in a cryogenic CO_2 processing unit (CPU, not shown), to meet CO_2 pipeline standards for oxygen, CO, water, and other contaminants. [26]

As noted in early studies of this plant, the imbalance between the specific heat and mass flow of the fluids on either side of the recuperation train requires additional heating of the high pressure sCO_2 returning to the combustor, which cannot be supplied by the hot, low pressure sCO_2 exiting the turbine, in order to maintain high cycle efficiency [1]. This is partially accomplished through thermal integration with the ASU's main air compressor (MAC) and the oxygen compressor (OC), as depicted by dashed red lines in Figure 3 and described in more detail below. Additional heat is supplied by adiabatic operation of the recycle compressor (RC), which compresses the sCO_2/oxygen mixture supplied to the combustor, in a similar manner to a recompression Brayton cycle used in indirect sCO_2 power cycles.

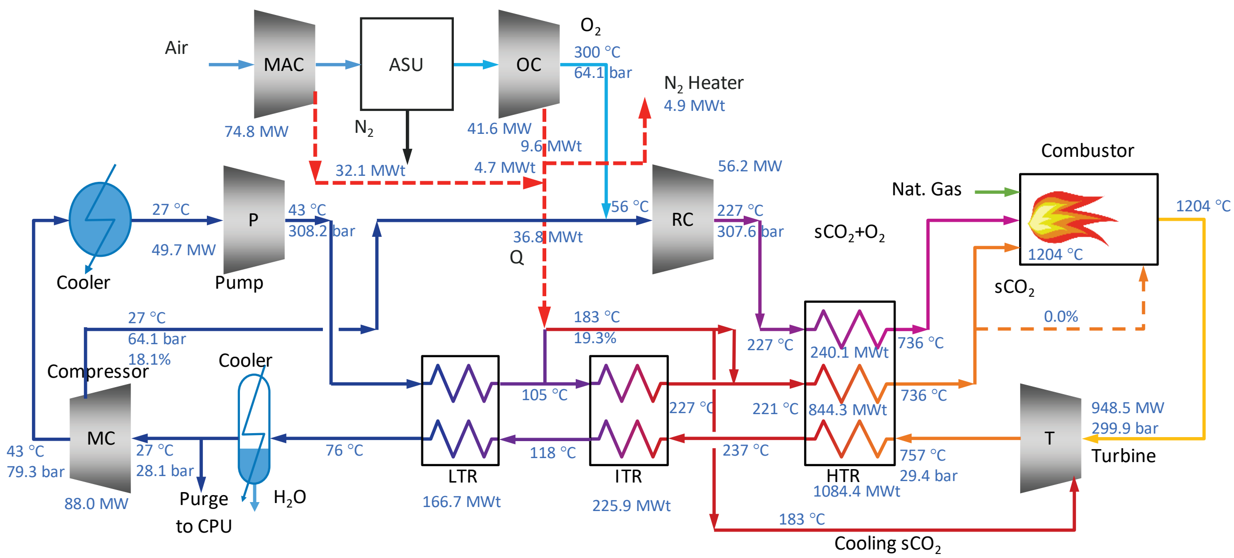

Figure 3: Block flow diagram for the baseline natural gas-fired sCO_2 power plant

Finally, the recuperation train is split into three recuperators to better manage thermal pinch points in the cycle. [27] The low temperature recuperator (LTR) is designed to handle condensation of water vapor from the hot, low pressure turbine exhaust stream. To avoid an internal pinch in the LTR, the hot-side inlet temperature is typically at or close to its dew point, and the approach temperature is fixed at about 10 °C at the LTR's hot end. Integration of process heat occurs in parallel to the intermediate temperature recuperator (ITR), where a portion of the cold stream exiting the LTR is split off and heated in the ASU MAC intercooler and the oxygen compressor intercooler, with the remainder of this stream passing through the cold side of the ITR. The flow rate of the ITR bypass stream is determined by a pinch analysis based on the minimum temperature approach (10 °C) in the MAC and oxygen compressor intercoolers. For simplicity, this stream is the source for the turbine blade cooling flow, with any excess flow mixing with the cold side ITR effluent before entering the HTR cold side. The remaining recycle streams undergo final preheating in the high temperature recuperator (HTR), with an overall goal of achieving a minimal approach temperature at the HTR's hot end in order to increase the effectiveness of the recuperators, maximize reactant preheating, and thus increase cycle and process efficiency.

ANALYSIS METHODOLOGY

The performance of the reference and conceptual sCO_2 plant designs were established using steady-state models developed using Aspen Plus® (Aspen). The direct sCO_2 plant modeling approach is described below, followed by a brief description of the economic analysis methodology.

sCO_2 Plant Modeling and Assumptions

Accurate modeling of sCO_2 power cycles requires high accuracy in determining the physical properties of CO_2, particularly near its critical point of 31 °C (88 °F) and 7.37 MPa (1069 psi). Due to difficulties in utilizing the preferred Span-Wagner Equation of State (EOS) for sCO_2 fluid mixtures, the Lee-Kesler-Plöcker EOS was used in modeling the sCO_2 cycle in this study, based on prior comparative analyses identifying this EOS as the most accurate for use in direct-fired sCO_2 cycle modeling. [28] [29]

Table 1 summarizes the sCO_2 cycle operating conditions and assumptions. Additional details and assumptions incorporated in the Aspen model regarding unit operation and cycle performance are described in the following sub-sections.

Table 1: Baseline sCO_2 power cycle configuration

Section	Parameter	Final Baseline
Combustor	O_2 purity	99.5%
	Excess O_2	1%
	Pressure drop	689 kPa (100 psid)
	Oxidant Preheating	To HTR exit temperature
	Nat gas Preheating	Via adiabatic compression
	Heat loss	Zero
Expander	Inlet temp	1204 °C (2200 °F)
	PR, P_{exit}	10.2, 2.94 MPa
	Blade Cooling	4.7% turbine inlet
Recuperator	Max temp	760 °C (1400 °F)
	Min T_{app}	10 °C (18 °F)
CO_2 cooler	Cooler/condenser	26.7 °C (80 °F)
	Cooling source	Process cooling water/Cooling tower
Recompression	CO_2 bypass	18.1%
CO_2 pre-compressor	P_{inlet}	2.81 MPa
	P_{exit}	7.93 MPa
	Isentropic efficiency	0.85
	Stages	4
	Intercooling stages	3
CO_2 pump	Exit pressure	30.82 MPa
	Isentropic efficiency	0.85
	Stages	2
	Intercooling stages	0
CPU	Presence	H_2O condenser, deoxy
	Impurities	H_2O, CO, O_2, Ar, N_2
	Streams	Working fluid/Seq CO_2

Oxy-combustor

For the baseline natural gas-fired direct sCO_2 plant configuration, the oxy-combustor is modeled in Aspen with a series of combustion reactions for the oxidizable components of the fuel and assuming 100% conversion of these fuel components. The amount of excess oxygen is calculated based on the stoichiometric amount needed for complete combustion of the fuel stream entering the process, without regard to any oxidizable components in the recycle sCO_2 stream.

An alternative three-stage combustion model is also used to explore the impact of incomplete combustion on the power cycle and plant performance. The feed to the combustor consisted of three streams: the fuel, the oxidant partially prediluted with recycle sCO_2 to a specified oxygen mole fraction of 0.3, and the main recycle sCO_2 stream. The sCO_2 recycle stream is further split with one portion sent to the first stage (the feed diluent portion) and the bypass portion sent to the third stage. The first stage fully oxidizes the fuel components to CO and H_2O by reaction (1), while the second stage equilibrates the products from the first stage with respect to the CO oxidation reaction (2):

$$C_xH_y + (x/2+y/4)\ O_2 \rightarrow x\ CO + y/2\ H_2O \tag{1}$$

$$CO + \tfrac{1}{2} O_2 \leftrightarrow CO_2 \tag{2}$$

The stream exiting the second stage is taken to represent the temperature and equilibrium products of the primary flame zone, consistent with full flame equilibrium modeling. [17] The third stage dilutes the second stage combustion products with the bypass stream to simulate a quenching process with no further chemical reactions occurring.

This simplified chemistry model is not intended to describe the full combustion mechanism, but rather provides a simple basis for examining the system performance as a function of the degree of incomplete oxidation with respect to the CO component. A bypass fraction of zero corresponds to sending all the fuel, oxidant, and recycle sCO_2 to the primary combustion zone and produces a combustion product stream composition and temperature that is essentially identical to that obtained from the complete combustion model due to the low degree of CO_2 dissociation at the low resulting flame temperatures. A bypass fraction of 100% produced the maximum flame temperature (subject to the constraint on the oxidant composition) and the maximum concentration of CO due to increased CO_2 dissociation.

sCO_2 Turbine

The sCO_2 turbine entrance temperature in the baseline case is 1204 °C (2200 °F), which is high enough to require blade cooling. An empirical turbine cooling model was developed and implemented in prior studies of coal-fueled direct sCO_2 cycles [10], based on variations in turbine cooling flow for turbine inlet temperatures of 1100 °C, 1150 °C, and 1200 °C, taken from the NET Power cycle analysis in the IEAGHG study [5]. This model assumed a turbine coolant temperature of 400 °C and a maximum blade metal temperature of 860 °C [5], but is enhanced in this work to account for variations in the required turbine coolant flow as a function of coolant temperature.

The revised turbine cooling model is based on establishing a correlation between the Gross Cooling Effectiveness and a Heat Loading Parameter. [30] The Gross Cooling Effectiveness is defined as:

Gross Cooling Effectiveness = $(T_{gas} - T_{bulk\ metal}) / (T_{gas} - T_{coolant})$

where:

T_{gas} = average hot gas temperature at the blade

$T_{bulk\ metal}$ = average metal temperature of the entire airfoil, with end walls

$T_{coolant}$ = temperature of the coolant entering the airfoil

The Heat Loading Parameter ratios the overall hot gas heat flux delivered to the component against the overall coolant capability to accept heat flux, and is defined as:

$$\text{Heat Loading Parameter} = (m_{coolant} \times Cp_{coolant}) / (2 \times H_{gas} \times A_{gas})$$

where:

$m_{coolant}$ = coolant mass flow rate to the airfoil
$Cp_{coolant}$ = coolant specific heat
H_{gas} = average external gas heat transfer coefficient
A_{gas} = external gas wetted surface area (of the blade)

In this implementation, A_{gas} is assumed to be a constant. The heat transfer coefficient is approximated using calculated gas properties entering each turbine stage at an assumed Mach number of 0.6 and a Nusselt number correlation for a 2.5 cm diameter cylinder in crossflow, roughly corresponding to an assumed blade width for the turbine modeled in the IEAGHG report [5].

The resulting cooling technology map correlation, based on the IEAGHG report data, is shown in Figure 4. This correlation is used to determine the Heat Loading Parameter, and hence the coolant mass flow required, for a given set of gas, metal, and coolant temperatures in the Aspen Plus model. The cooling bleed flow to each stage was based on the temperature of the stream entering the stage, T_n, where the ratio of cooling bleed at stage $n+1$ to the cooling bleed at stage n was set equal to $(T_{n+1} - T_{max}) / (T_n - T_{max})$. T_{max} was assumed to be 860 °C (1580 °F), the same as for the IEAGHG report. [5] Based on this analysis, only the first five turbine stages are cooled in the Aspen model of the sCO_2 turbine, which is roughly consistent with Toshiba's sCO_2 turbine cooling scheme [31].

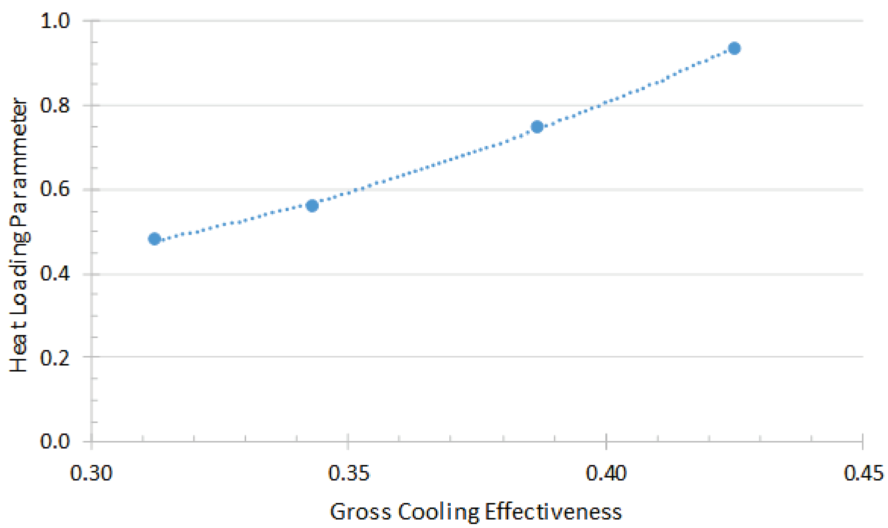

Figure 4: Cooling technology map for direct-fired sCO_2 turbine

The approach employed here is simplistic and largely empirical. A physical sCO_2 turbine blade cooling model is currently under development, and will be used in future analyses of direct sCO_2 systems to improve model accuracy.

Heat Exchangers

The sCO_2 cycle recuperators were modeled as adiabatic countercurrent heat exchangers with perfect thermal mixing in the transverse direction and negligible heat transfer in the axial direction. The minimum temperature approach throughout the recuperator train is limited to 10 °C (18 °F). Because of the sCO_2 flow rate constraints imposed by the cycle configuration, this minimum temperature approach generally occurred at the cold end of the HTR (hot end of the ITR). An additional constraint imposed on the LTR was to try to keep the hot side inlet stream at the dew point temperature to avoid an internal temperature approach. In some cases, that could not be attained but a pinch analysis was performed to make certain that the internal minimum temperature approach was at least 10 °C (18 °F).

During the preliminary model runs to establish the cycle configuration, the total recuperator train pressure drop was assumed to be 0.14 MPa (20 psid) each for the high-pressure and low-pressure sides. For the final model, the pressure drops were calculated as a fraction of the inlet stream pressure and in proportion to the duty of the individual recuperator. The pressure drop factor was set separately for the high-pressure and low-pressure sides and set such that the total pressure drop per side was approximately 0.14 MPa (20 psid) in the baseline case.

The two CO_2 coolers were modeled as adiabatic countercurrent heat exchangers. Process cooling water was used as the cold sink. Perfect thermal mixing in the radial direction was assumed and heat transfer in the axial direction was neglected. The minimum temperature approach through the coolers is limited to 11.1 °C (20 °F). The assumed pressure drop through both coolers was 0.10 MPa (15 psid).

Compressors

To maximize heat integration between the ASU and the sCO_2 power cycle, the multi-stage water intercooled main air compressor (MAC) model typically used in oxy-combustion system studies was replaced with an adiabatic compression train with aftercoolers. The first aftercooler uses recycle sCO_2 exiting the cold side of the LTR as the cold sink, with a flow rate set to yield a 10 °C (18 °F) minimum temperature approach in the aftercooler. The second aftercooler uses process cooling water as the cold sink and cools the air to the feed temperature of the air pre-purifier.

The oxygen from the ASU is compressed to the oxy-combustor feed pressure in a 2-stage compression train designed to allow process and cycle heat integration while limiting the maximum temperature of the oxygen stream to 300 °C (572 °F). The temperature limit was imposed for safety considerations but is not based on literature or vendor data. Three intercoolers are placed between the two compression stages. The first used a nitrogen stream from the ASU as the cold sink, which is then used in the adsorbent regenerator for the ASU pre-purifier. The second intercooler uses recycle sCO_2 exiting the cold side of the LTR as the cold sink, with a flow rate set to yield a 10 °C (18 °F) minimum temperature approach in the aftercooler. The third intercooler uses process cooling water as the cold sink and cools the oxygen to 26.7 °C (80 °F) before entering the second compression stage.

The CO_2 pre-compressor was modeled as a four-stage intercooled compressor having an isentropic efficiency of 0.85 for each stage. The three intercoolers used process cooling water as the cold sink and had an assumed pressure drop of 0.014 MPa (2 psid) each. [32]

A takeoff stream of sCO_2 exits the pre-compressor after the third intercooler for mixing with the oxygen stream, and is then compressed in the oxidant compressor, an adiabatic compressor with an assumed isentropic efficiency of 0.85. The sCO_2 takeoff flow is fixed to yield a 30% oxygen stream (by volume) in the oxidant supply to the combustor, inclusive of the oxygen content in the sCO_2 recycle stream. This is deemed to be the maximum safe oxygen concentration for this hot, high pressure stream, though no literature or vendor data has been found to date to support this high oxygen concentration. Additional research is needed to determine the actual oxygen safety limits for these operating conditions.

Although Aspen reports that the stream entering the CO_2 pump has a vapor fraction of zero, it is so close to the critical point that it was treated as a compressible fluid. Therefore, the CO_2 pump was modeled as an adiabatic compressor with an isentropic efficiency of 0.85.

Economic Analysis Methodology

Based on the steady-state modeling results, pertinent operating conditions of cycle and balance of plant components are used to estimate costs for equipment, installation, contractor fees, and contingencies. Following a standard cost estimating methodology [33] [34], these are summed to arrive at a total plant cost (TPC), expressed in June 2011 dollars for consistency with other NETL studies. [10] [23] Using a consistent methodology and set of assumptions [34], owner's costs are also calculated and added to the TPC to arrive at the total overnight cost (TOC), which is used in the cost of electricity (COE) calculation.

sCO_2 Cycle Component Cost Estimates

The components in the sCO_2 power cycle are not present in the reference plant and new cost algorithms had to be derived for these units. The component cost algorithms developed under prior work on coal-fueled direct sCO_2 power cycles were used in this study, and are briefly outlined here. [10] [25]

The most novel component of the cycle is the combustor and turbine, for which no cost estimates exist in the public domain at any scale. The approach taken with this component is to combine the cost of a similarly-sized gas turbine (without the compressor) with the cost of a high pressure outer casing similar to those used for HP steam turbines. Costs for these components are well-known, and combine to constitute a cost estimate for a mature, n^{th}-of-a-kind (NOAK) direct sCO_2 turbine and combustor, albeit with a high degree of uncertainty. In contrast, CO_2 pumps and compressors are more commercially-available, thus their costs are estimated from scaling relationships applied to vendor quotes for utility-scale compressors for indirect sCO_2 recompression cycles.

Cost estimates for the recuperators and coolers are derived from "UA-type" cost scaling relationships that include the thermal duty of the heat exchanger and the average temperature differential between its hot and cold sides. The cost estimates for the LTR, ITR, CO_2 pre-cooler and the CO_2 condenser are based on a specific cost of \$0.294/(W/K), while those for the HTR are divided into low and high temperature sections to account for material variability. When the hot side fluid is at or below 600 °C, the HTR cost is \$0.253/(W/K) and for temperatures above 600 °C the HTR cost is \$1.318/(W/K), with units assumed to be installed in series.

Capital Cost Estimate

Aside from the thermal power cycle, the balance of plant unit operations and equipment items are analogous to those found in the reference NGCC plant. The cost estimates for these items were based on a combination of vendor data, estimates from Worley-Parsons, power law scaling, and correlations that were fit to historical cost estimates published in previous NETL reports. [22] [23]

Cost scaling algorithms with reference costs and reference process variable data were taken from the reference NGCC plant analysis, which are consistent with published cost scaling guidelines. [33] The same BOP cost accounts as for the reference NGCC plant were used except that there was no steam turbine generator or HRSG and some of the sub-accounts from the NGCC case did not apply, including the Feedwater & Miscellaneous BOP Systems and the steam turbine building.

Following this methodology, capital cost estimates are expected to have an accuracy in the -15% to +30% range [34], however, a -15% to +50% accuracy range is more appropriate since direct sCO_2 power cycles are only in the demonstration stage, consistent with an AACE Class 4 Feasibility Study designation. [35]

Cost of Electricity (COE) Calculation

The COE consists of five components representing normalized contributions from capital costs, fixed operations & maintenance (O&M) costs, variable O&M costs, fuel costs, and the transportation and storage (T&S) costs for captured CO_2. These are calculated following a consistent methodology used in all NETL studies [34]. The financial assumptions used in the COE calculation for the sCO_2 plant are detailed elsewhere [25], but are identical to those used for the reference NGCC plant [23], including the capacity factor (85%), natural gas fuel price ($6.13/MMBtu), plant construction period (3 years), operating lifetime (30 years), staffing (6 operators/shift), and other plant financing assumptions.

TECHNO-ECONOMIC ANALYSIS RESULTS

Performance Results Summary

Table 2 shows the power summary for the baseline sCO_2 plant and reference NGCC plant. It is assumed that the CO_2 pre-compressor, pump, and recycle compressor are all connected to the CO_2 turbine shaft. Table 3 shows the performance summary for these two cases. Compared to the NGCC plant, the sCO_2 plant shows a 2.5 percentage point higher HHV efficiency, 31 MW higher net power output, 3.5 times higher auxiliary power requirement, 7.5% more CO_2 captured, and 17% less water withdrawal.

Table 2: Power and auxiliary load summary for the baseline sCO_2 plant and the reference NGCC plant

	NGCC plant	sCO_2 plant
Power Summary		
Combustion Turbine Power, MWe	428	949
Steam Turbine Power, MWe	182	---
CO_2 Pre-compressor Power, MWe	---	-88
CO_2 Pump Power, MWe	---	-50
CO_2 Recycle Compressor Power, MWe	---	-56
Generator Loss, MWe	-9	-16
Total Gross Power, MWe	**601**	**738**
Auxiliary Load Summary		
Feedwater Pumps, kWe	3,550	---
SCR, kWe	2	---
CO_2 Capture/Removal Auxiliaries, kWe	13,000	---
ASU MAC, kWe	15,010	74,780
Natural Gas Compressor Power, kWe	---	12,710
sCO_2 Oxygen Compressor Power, kWe	---	41,620
CPU & CO_2 Compression, kWe	---	8,680
Miscellaneous Balance of Plant[A], kWe	500	500
Combustion Turbine Auxiliaries, kWe	700	700
Steam Turbine Auxiliaries, kWe	100	---
Condensate Pump, kWe	130	---
Circulating Water Pumps, kWe	4,310	4,180
Ground Water Pumps, kWe	360	320
Cooling Tower Fans, kWe	2,230	2,160
Transformer Losses, kWe	1,840	2,570
Total Auxiliaries, MWe	**42**	**148**
Net Power, MWe	**559**	**590**

A-Includes plant control systems, lighting, HVAC, and miscellaneous low voltage loads

Table 3: Performance summary for the baseline sCO_2 plant and the reference NGCC plant

	NGCC plant	sCO_2 plant
Natural Gas Feed Flow, kg/hr (lb/hr)	84,134 (185,484)	84,134 (185,484)
HHV Thermal Input, kWt	1,223,032	1,223,052
LHV Thermal Input, kWt	1,105,162	1,105,180
Total Gross Power, MWe	601	738
Total Auxiliaries, MWe	42	148
Total Net Power, MWe	**559**	**590**
HHV Net Plant Efficiency, %	45.7	48.2
HHV CT/sCO_2 Cycle Efficiency, %	34.5	58.6
LHV Net Plant Efficiency, %	50.6	53.4
LHV CT/sCO_2 Cycle Efficiency, %	38.1	66.8
Steam Turbine Cycle Efficiency, %	43.5	---
Condenser/sCO_2 Cooler Duty, GJ/hr (MMBtu/hr)	888 (842)	1,978 (1,875)
Raw Water Withdrawal, $(m^3/min)/MW_{net}$ (gpm/MW_{net})	0.027 (7.2)	0.023 (6.0)
Raw Water Consumption, $(m^3/min)/MW_{net}$ (gpm/MW_{net})	0.020 (5.4)	0.016 (4.3)
Carbon Capture Fraction, %	90.7	98.2
Captured CO_2 Purity, mol%	99.93	100.00

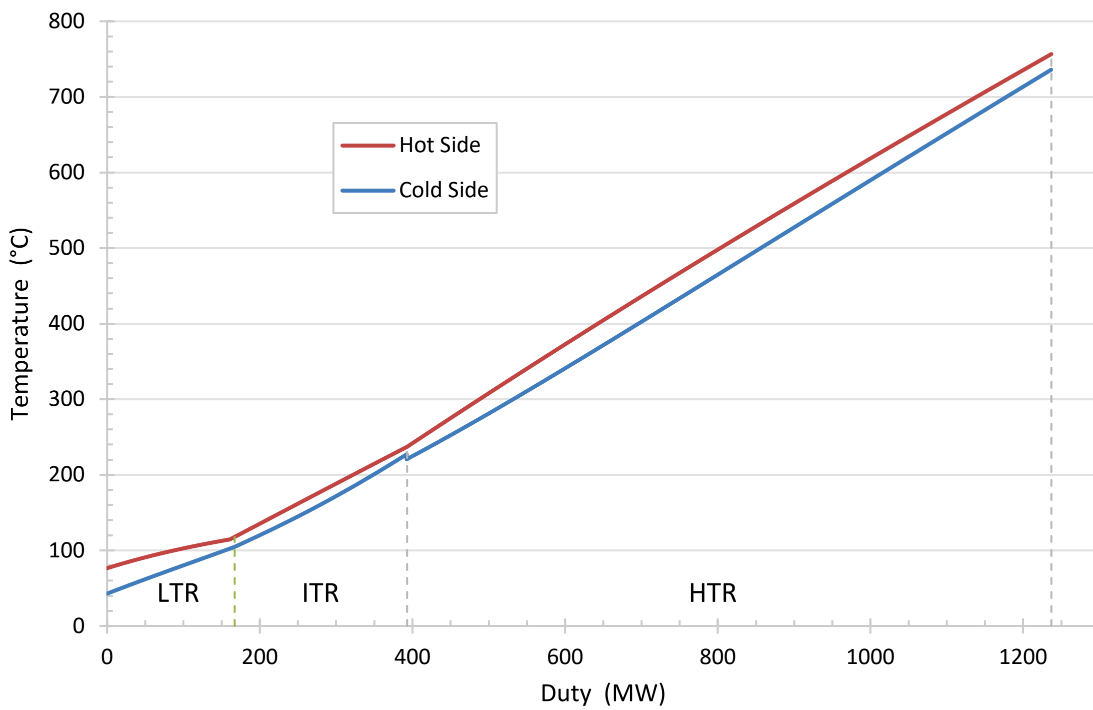

Figure 5: Recuperator pinch analysis - T-Q diagram

Figure 5 shows the temperature-heat duty (T-Q) diagram for the recuperator train in the baseline sCO_2 power cycle. The vertical dashed lines show demarcation between the LTR, ITR, and HTR. The minimum approach temperature occurs at the hot end of the ITR. There is a large approach temperature at the cold end of LTR due to condensation on the hot side. The average recuperator approach temperature is just above 25 °C and the average recuperator LMDT is a little below 18 °C.

Preliminary Economic Analysis Results

Table 4 shows the preliminary account level TPC and TOC summary for the baseline sCO_2 plant compared to the reference NGCC plant in 2011 dollars. Compared to the NGCC plant, the sCO_2 plant shows 5% greater total TPC, 101% greater power island cost, 92% less CO_2 capture/compression cost, and 49% greater BOP costs, primarily from the Accessory Electric Plant account. Notably, the ASU is the largest-cost subsystem in the sCO_2 plant, though combined with the CO_2 Removal and Compression account, the total cost is comparable to that of the NGCC plant. The component costs for the sCO_2 cycle are detailed elsewhere [25], but are roughly comprised of installed costs for the oxy-turbine (19%), compressors (58%), heat exchangers (19%), and piping/foundations (4%).

Table 4: Preliminary TPC and TOC summary for the baseline sCO_2 plant and the reference NGCC plant

		TPC, TOC ($1,000)	
Account	Description	NGCC plant	sCO_2 plant
3	Feedwater & Miscellaneous BOP Systems	57,936	36,403
4	Cryogenic ASU	0	342,954
5B	CO_2 Removal & Compression	378,178	28,293
6	Combustion Turbine & Accessories	134,931	263,596
7	HRSG, Ducting, & Stack	50,316	0
8	Steam Turbine Generator	74,543	0
9	Cooling Water System	27,502	26,897
11	Accessory Electric Plant	59,813	107,393
12	Instrumentation & Control	19,568	41,861
13	Improvements to Site	11,987	13,207
14	Buildings & Structures	13,130	7,343
	Total Plant Cost (TPC)	**827,904**	**867,945**
	Owner's Costs	180,477	188,034
	Total Overnight Cost (TOC)	**1,008,381**	**1,055,979**

Table 5 compares the preliminary operating and maintenance (O&M) costs between the two cases. The only significant difference is the higher consumables cost for the NGCC plant, due to the makeup need for CO_2 capture solvents. Table 6 compares the preliminary cost of electricity for the two cases, where the sCO_2 plant shows an approximately 5% decrease in COE relative to the reference NGCC plant with CCS. Savings in fuel and O&M costs for the more efficient sCO_2 plant offset the slight increase in total plant cost relative to the NGCC plant, spread over the expected 30-year lifetimes of the plants. However, the findings from this economic analysis cannot be deemed definitive given the relatively large uncertainty inherent in a Class 4 capital cost estimate. If the expected range of uncertainty in this estimate (-15% to +50%) is applied to the TPC for both the NGCC plant and the sCO_2 plant, the relative differences in the total COE ranges from 5% lower to only 2% lower. If the capital cost uncertainty is concentrated solely in the sCO_2 power cycle components, then the relative differences in the total COE range from 6% lower to 1% greater. A more extensive sensitivity analysis focused on the recuperator capital costs is shown in the following section.

Table 5: Preliminary O&M costs for the baseline sCO_2 plant and the reference NGCC plant

Description		O&M ($1,000/yr)	
		NGCC plant	sCO_2 plant
Fixed O&M Costs	Labor	10,810	10,973
	Property Taxes & Insurance	16,558	17,359
	Total Fixed O&M Costs	**27,368**	**28,332**
Variable O&M Costs	Maintenance Material	8,679	9,098
	Consumables	7,821	2,578
	Total Variable Costs	**16,500**	**11,677**
Fuel Cost	**Natural Gas**	**190,913**	**190,913**
Total O&M Costs		**234,780**	**230,921**

Table 6: Preliminary COE for the baseline sCO_2 plant and the reference NGCC plant

COE Summary ($/MWh)		
	NGCC plant	sCO_2 plant
Capital Cost	26.9	26.7
Fixed O&M Costs	6.6	6.4
Variable O&M Costs	4.0	2.7
Fuel Cost	45.9	43.5
Total w/o T&S	**83.3**	**79.2**
CO_2 T&S Cost	4.0	4.1
Total	**87.3**	**83.3**

Sensitivity Analysis Results

Recuperator Cost Estimates

The recuperator cost estimates in this paper were based on the lowest of two proprietary vendor quotes for LTR and HTR recuperators for a commercial-scale indirect-fired sCO_2 plant. Both vendors employed non-PCHE (printed circuit heat exchanger) recuperator designs, though PCHEs are the more mature recuperator technology, and are hence being employed in the NET Power pilot plant. [4] The design approach and resulting cost estimates for the recuperators varied widely between the vendors. The modified specific recuperator costs for the higher estimate are \$0.440/(W/K) for the LTR and ITR, while those for the HTR are divided into low and high temperature sections to account for material variability. When the hot side fluid is at or below 600°C, the HTR cost is \$0.586/(W/K) and for temperatures above 600 °C the HTR cost is \$3.274/(W/K), with units assumed to be installed in series.

To illustrate the effect of sCO_2 component capital cost uncertainty on the overall economics of the direct sCO_2 plant, Table 7 summarizes the variability in the recuperator TPC and overall plant COE that results from the two recuperator vendor estimates. These results show that vendor designs and costs for sCO_2 recuperators vary widely, as do their effects on overall direct sCO_2 plant economics.

Table 7: Recuperator TPC ($1,000) and Plant COE Summary ($/MWh)

	NGCC plant	sCO_2 plant	
		Vendor X	Vendor Y
Recuperator TPC ($1,000)			
HTR	---	23,880	78,106
ITR	---	5,739	8,600
LTR	---	2,236	3,351
Total Plant Cost ($1,000)	**827,904**	**867,945**	**939,363**
COE w/o T&S ($/MWh)	**83.3**	**79.2**	**82.1**
T&S Cost ($/MWh)	4.0	4.1	4.1
COE Total ($/MWh)	**87.3**	**83.3**	**86.1**

Degree of Incomplete Combustion

The results for the baseline sCO_2 plant presented above assume that complete oxidation occurs in the oxy-combustor. To assess the impact of this assumption, a sensitivity analysis was performed to examine the impact of incomplete combustion on cycle and process performance by varying the combustor diluent bypass fraction between 0 and 1. The calculated adiabatic flame temperature rises with the bypass fraction, as shown in Figure 6, due to less sCO_2 dilution within the primary combustion zone. Also shown are the CO and O_2 mole fractions in the combustor effluent as a function of the combustor diluent bypass fraction. At bypass fractions above 0.8 the CO oxidation reaction equilibrium begins to favor larger amounts of oxygen and CO in the combustor effluent due to CO_2 dissociation at flame temperatures above about 1800 °C. Note that the total sCO_2 recycle flow rate to the combustor is varied in these cases to maintain a turbine inlet temperature of 1204 °C (2200 °F), while the constant recycle flow case is analyzed in the following section.

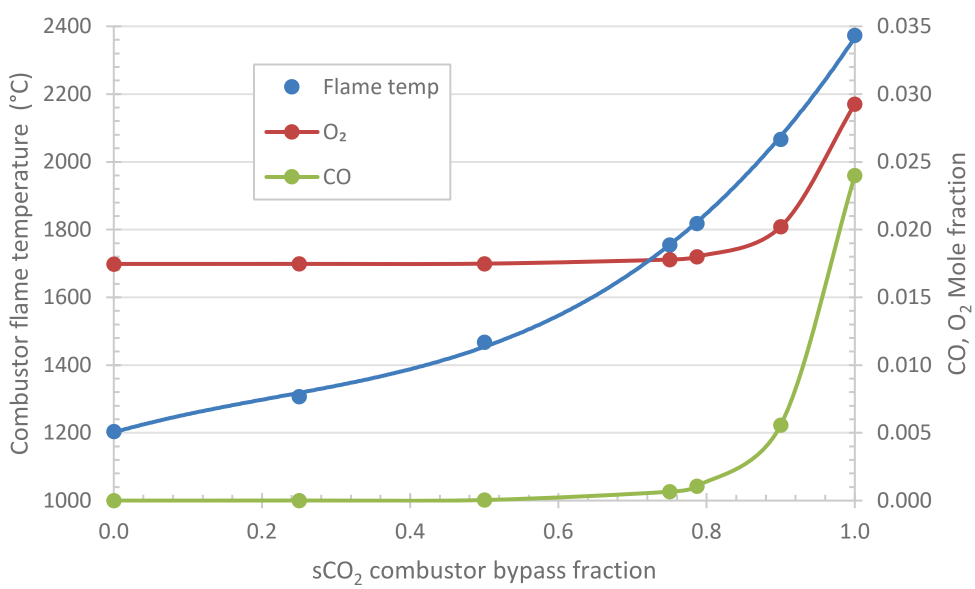

Figure 6: Combustor flame temperature and effluent CO & O_2 mole fractions as a function of diluent bypass fraction

Figure 7 shows the variation of plant/process and cycle efficiency with the combustor diluent bypass fraction. There is little impact on process or cycle efficiency for bypass fractions below 0.8 corresponding to flame temperatures between 1204 °C and 1800 °C. As the bypass fraction increases beyond 0.8, the process and cycle efficiency drop quickly. This relationship is similar, if inverted, to the mole fraction dependence shown in Figure 6, suggesting a direct relationship between impurity mole fraction and process/cycle efficiency. This is demonstrated in Figure 8, which shows the efficiency dependence (constant TIT) on CO mole fraction, accounting also for the effect of the associated O_2 mole fraction from CO_2 dissociation, per Figure 6. Both relationships are nearly linear, and show that process and cycle efficiency drops by about 0.75 percentage points per mole percent of CO in the combustor exhaust.

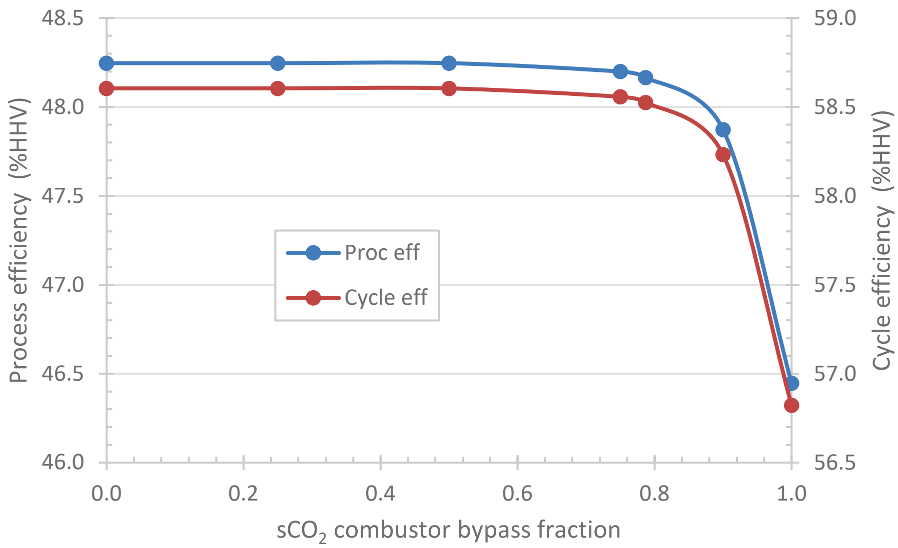

Figure 7: Process and cycle efficiency as a function of diluent bypass fraction

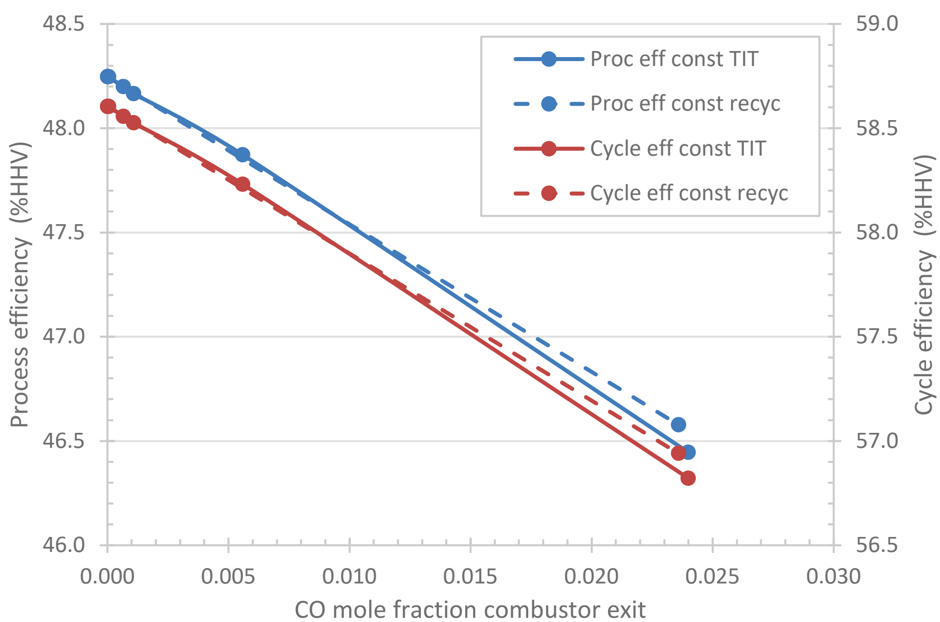

Figure 8: Process and cycle efficiency as a function of CO mole fraction

To summarize these results, there is no significant impact on cycle performance (<0.1 percentage points) for combustor flame temperature below 1800 °C. The dependence of process and cycle efficiency on O_2 and CO concentration in the exhaust is nearly linear. Finally, at maximum combustor flame temperature, the net power was reduced 22 MW and the process efficiency reduced 1.8 percentage points compared to the complete combustion case.

Constant Recycle Flow Case

As noted previously, the cases described to this point all maintain a TIT of 1204 °C by varying the recycle sCO_2 stream flow rate. An alternative approach for controlling the power cycle would be to keep the recycle sCO_2 stream flow rate constant and allow the TIT to vary. To examine the impact of this mode of operation on the cycle and plant performance, a sensitivity case was run on an incomplete combustion case (bypass fraction 1, maximum combustor flame temperature) with the recycle flow rate kept the same as for the complete combustion case. The results are found to be very close to the results for the constant TIT case, but with a TIT of 1192 °C, only 12 °C lower than the constant TIT case.

The impact of maintaining a constant recycle flow on the efficiency dependence on combustor exit composition is shown as the dashed lines in Figure 8. Even at the highest combustor flame temperatures, there is very little difference in the process and cycle efficiencies between these two modes of operation, and only slight differences in other cycle performance parameters. These results suggest that maintaining a constant TIT in the event of incomplete combustion is not critical to the power cycle operation as long as the variation in TIT and recycle flow rate is not too great.

Comparison with Other Studies

The baseline direct sCO_2 plant developed in this study is compared to other natural gas-fueled direct sCO_2 systems found in the literature in Table 7. The efficiency of the system in this study is slightly low compared to thermal efficiencies obtained in other studies, though this is likely a result of the higher CO_2 purity and capture fraction in this study. The specific power in this study, defined as the net plant power output divided by the turbine exit flow, is high relative to other studies. This is due to the higher pressure ratio, higher turbine inlet temperature and lower turbine cooling flow, and is a primary contributor to the lower specific plant cost relative to the IEAGHG and EPRI studies.

Table 8: Natural gas fueled direct sCO_2 plant design and performance comparison

Item	Units	This Study	8 Rivers Capital [1]	IEAGHG [5]	EPRI [6]	Scaccabarozzi et al [8]
Turbine Inlet Temp	°C	1204	1150	1150	1150	1127.7
Turbine Pressure Ratio		10.2	10	8.8	8.8	6.1
Turbine Cooling Flow	%	4.7%		11.5%	11.5%	6.6%
Turbine Coolant Temp.	°C	183	<400	400	400	164
Thermal Input (HHV)	MWth	1223		1701	1374	851
Net Plant Power	MWe	590		846	664	425
Net Plant Efficiency	%, HHV	48.2%	53.1%	49.9%	48.4%	50.0%
Specific Power	kJ/kg	334.2		300.0	290.8	267.4
CO_2 capture	%	98.2%	100.0%	90.0%	90.1%	
CO_2 purity	%	100%	94%	99.8%	99.6%	
Specific Plant Cost	\$/kWe†	1471	~1000*	1651	1555	

†2011 dollar year basis; * target

CONCLUSIONS & FUTURE WORK

The above study develops a baseline plant design for a natural gas-fueled direct sCO_2 power plant, which is modeled in Aspen Plus to determine plant performance and component operating parameters used to estimate plant cost and overall COE. The plant design is similar to that of other studies, but evolved organically out of the basic framework of direct sCO_2 power cycles [1] and earlier work on coal-fueled direct sCO_2 power plants [10] [11]. The design incorporates a thermal recuperation scheme using three recuperators, thermal integration with both the ASU main air compressor and oxygen compressor, and an adiabatic recycle compressor for the O_2/sCO_2 oxidant flow to help balance the recuperation train. Additional contributions of this work include a new model to determine sCO_2 turbine cooling flow requirements as a function of the coolant temperature, an incomplete combustion model to assess the effects of combustion-derived sCO_2 impurities on the overall performance of the plant, a component-level cost estimate for the plant, and a COE calculation consistent with other NETL studies.

The performance results yield a baseline sCO_2 plant thermal efficiency of 48.2% (HHV) with 98.2% carbon capture, a significant improvement over the reference NGCC plant with CCS, which has an efficiency of 45.7% and 90.7% carbon capture. [23] Sized on a constant fuel flow rate basis, the total plant cost for the baseline sCO_2 plant is 4.8% higher than the NGCC plant, though due to the sCO_2 plant's higher net power output, the two plants are nearly identical on a $/kW basis. With the increased fuel efficiency, this leads to a 4.9% lower COE for the sCO_2 plant, excluding CO_2 T&S costs. Relative to other direct sCO_2 studies fueled by natural gas, the designed plant has a slightly lower efficiency due to its higher rate of carbon capture, and a higher specific power that reduces the overall plant size and capital cost.

The effects of incomplete combustion on direct sCO_2 cycle performance have also been analyzed in this study by enforcing chemical equilibrium of the $CO + \frac{1}{2} O_2 \leftrightarrow CO_2$ reaction at the flame temperature, which is adjusted by controlling the amount of recycle sCO_2 diluent sent to the primary combustion zone. These results show that CO_2 dissociation into CO and O_2 begins to occur for flame temperatures above about 1800 °C, and results in a loss of about 0.75 percentage points in cycle or plant efficiency for every mole percent of CO in the combustor exhaust due to increases in required cycle compression power in the presence of these sCO_2 impurities. These results are applicable to any processes that may yield incomplete combustion, including chemical kinetics, flame quenching, and improper fuel/oxygen mixing, which points to the need for sCO_2 oxy-combustor design and modeling studies to ensure maximum conversion of fuel and oxygen to CO_2 in the combustion products.

The baseline sCO_2 plant design developed in this work will undergo further optimization and modification in future studies to develop a more cost-effective plant design. In particular, potential for efficiency improvement exists in implementing a high pressure ASU model in which the oxygen is pumped to high pressure as a liquid with low specific power, rather than compressing it as a gas in this study. Future plans also include the development of control strategies and detailed component performance maps to enable evaluation of the part load and off-design capabilities of direct sCO_2 power cycles.

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