PERFORMANCE BASELINE FOR DIRECT-FIRED sCO_2 CYCLES

Nathan Weiland
Engineer
National Energy Technology Laboratory
Pittsburgh, PA U.S.
Nathan.Weiland@NETL.DOE.GOV

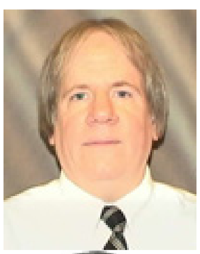
Wally Shelton
Engineer
National Energy Technology Laboratory
Morgantown, WV U.S.
Walter.Shelton@NETL.DOE.GOV

Charles White
Senior Principal
Noblis
Falls Church, VA U.S.
Charles.White@noblis.org

David Gray
Senior Principal
Noblis
Falls Church, VA U.S.
DGray@noblis.org

Dr. Nathan Weiland is an engineer in the Systems Engineering & Analysis group at the National Energy Technology Laboratory (NETL), where he where he performs systems studies of supercritical CO_2 power cycles and oxy-fuel magneto-hydrodynamics (MHD) power plants. Prior to joining the systems analysis group he was a research professor at West Virginia University, where he worked with NETL on low-NOx hydrogen combustion, coal/biomass co-gasification, ash deposition processes in gasification systems, oxy-combustion plasmas for MHD power, and chemical looping combustion. He has received a BS (Purdue University), MS, and PhD (Georgia Tech) in Mechanical Engineering.

Walter Shelton is an engineer in the Systems Engineering & Analysis group at NETL supporting research/process development projects. Focused on the analysis and development of processes based on fossil fuels spanning both existing and emerging technologies that produce electrical power or a combination of power and chemicals. Key aspects involve performing system process studies that determine the efficiency, economic, regulatory and environmental impacts. The process studies are targeted to contribute to formulating and determining government/public policy. He received a BS (SUNYAB) and MS (U. of Minnesota) in Chemical Engineering.



Charles White is a senior principal engineer at Noblis, where he performs modeling, simulation, and systems analysis of fossil-fueled processes for producing electric power and transportation fuels. He has provided energy systems analysis modeling and simulation support to NETL since 1985. He received a PhD in chemical engineering from the University of Pennsylvania.

Dr. Gray is a senior principal engineer at Noblis. He has extensive experience in conversion of coal into electric power, fuels and chemicals and has provided research and development (R&D) guidance to NETL in those areas for many years. Prior to coming to the U.S. he worked in South Africa where he was an R&D manager operating pilot plants for alternative transportation fuels production. He has served on the National Academies of Sciences Committee on America's Energy Future and as a subject matter expert on the National Petroleum Council Study on Future Transportation Fuels. He received a PhD in physical chemistry from the University of Southampton in the United Kingdom.

ABSTRACT

Direct supercritical CO_2 (sCO_2) power cycles have recently received interest as a potentially lower-cost, fossil-fueled power source with inherent amenability to carbon capture. In this cycle, heat addition occurs via fossil fuel combustion with oxygen, while sCO_2 is recycled back to the combustor to limit combustion temperatures. This high-temperature and high-pressure working fluid is then expanded through a turbine. After water is condensed from the working fluid, a portion of the CO_2 is exhausted from the cycle, purified as needed, and pressurized for enhanced oil recovery (EOR) or storage.

NETL has conducted an evaluation of the performance and emissions for a direct coal-fired sCO_2 power plant. This study describes a baseline coal-fired cycle configuration, where coal is first gasified and cleaned in order to avoid introducing sulfur and particulate matter into the sCO_2 cycle, with the sCO_2 cycle's oxy-combustor operating on syngas. The baseline sCO_2 plant design yields a net plant thermal efficiency of 37.7% (HHV), with 98.1% CO_2 capture at 99.4% purity. This compares favorably to the reference IGCC plant, which has a 31.2% net HHV thermal efficiency, and 90.1% CO_2 capture rate at 99.99% purity. The sensitivity of the sCO_2 plant's performance to its process variables is discussed, as well as their effect on plant operability and cost surrogate variables.

1 Introduction

The United States (U.S.) Department of Energy's (DOE) Clean Coal and Carbon Management Program (CCCMP) provides a worldwide leadership role in the development of advanced coal-based energy conversion technologies, with a focus on electric power generation with carbon capture and storage (CCS). As part of DOE's Office of Fossil Energy (FE), the National Energy Technology Laboratory (NETL) implements research, development, and demonstration (RD&D) programs that address the challenges of reducing greenhouse gas emissions. To meet these challenges, FE/NETL evaluates advanced power cycles that will maximize system efficiency and performance, while minimizing CO_2 emissions and the costs of CCS.

To this end, NETL has recently been investigating direct-fired supercritical CO_2 (sCO_2) power cycles, which are attractive due to their high efficiency and inherent ability to capture CO_2 at storage-ready pressures. In these cycles, fuel is combusted with oxygen in a highly dilute sCO_2 environment, with the combustion products driving an expansion turbine to generate power. The thermal energy in the turbine exhaust is recuperated in a compact heat exchanger to heat the CO_2 diluent flow to the combustor, followed by condensation of water out of the product stream. A portion of this stream is sent to a purification unit for CO_2 storage, while the bulk of the fluid is compressed for return to the combustor. Most of these processes occur at elevated pressures of 200-400 bars in the combustor and 10-80 bars at the condenser, which leads to a high-power density cycle with a reduced footprint relative to conventional power generation technologies. Resulting capital costs are somewhat offset by the need to contain the high pressures, but combined with the high efficiencies, direct-fired sCO_2 power plants are expected to be comparable to, or better than, conventional combined cycle plants with CCS, on a cost of electricity basis.

Several analyses of direct sCO2 power cycles are available in the literature, including those of Allam and colleagues (12, 13), who are pursuing commercialization of this technology through the construction of a 25 MWe demonstration plant over the next few years. In their natural gas-fired version of this cycle, they suggest that net plant thermal efficiencies of 53% (higher heating value [HHV]) are achievable (12) with near 100% carbon capture (13). Foster Wheeler's modeling of this system under slightly different assumptions yields a plant thermal efficiency of 50% (HHV), with 90% carbon capture (15). Southwest Research Institute has evaluated alternative natural gas-fired direct sCO_2 cycles, and have reported plant thermal efficiencies of 48% (14). For comparison, baseline natural gas combined cycle (NGCC) plants with 90% carbon capture can achieve a plant efficiency of 45.7% (19).

NET Power has also developed a coal-fired version of their direct sCO_2 cycle, in which coal is first gasified and cleaned before syngas is burned in the sCO_2 cycle combustor (12). In the baseline system, an entrained flow, dry-fed, slagging gasifier is used with a water quench, which produces a claimed net plant thermal efficiency of 47.8% (HHV) on bituminous coal (6). Variations in coal type, gasifier type, and heat recovery processes yield a range of HHV efficiencies from 43.3% to 49.7% (6).

The Electric Power Research Institute (EPRI) has also studied a syngas-fired direct sCO_2 power plant based on coal gasification in a slagging, entrained flow gasifier (4). The study includes Shell's dry-fed gasifier design (including a steam bottoming cycle powered by the syngas cooler's thermal input) and investigates the effects of oxygen purity and coal carrier gas on the purity of the CO_2 in the power cycle's turbomachinery. The study concludes that high oxygen purity (99.5%) and CO_2 carrier gas are required to produce a storage-ready stream with sufficient CO_2 purity (98.1%) for permanent sequestration. Plant thermal efficiency for this case was 39.6% (HHV), with 99.2% CO_2 capture rate, compared to 31.1% plant efficiency with 87% CO_2 capture in the integrated gasification combined cycle (IGCC) reference plant. Cost of electricity was also estimated at 133 $/MWh, compared to 138 $/MWh for the IGCC plant with capture, though significant uncertainty in the sCO_2 capital cost was noted (4).

Building on the studies found in the literature, the objective of the present study is to develop a performance baseline for a syngas-fired direct sCO_2 power plant using coal gasification, and to analyze the sensitivity of its net thermal efficiency and cost indicators to variations in operating parameter assumptions. These results will be utilized in future studies that will include capital cost estimation and plant optimization to minimize cost of electricity.

2 Coal-Fired Direct-sCO_2 Power Plant Design

As noted above, the objective of this study is to determine the extent to which the open Brayton cycle based on sCO_2 offers advantages compared to other coal-fired power plant technologies in the NETL research and development (R&D) portfolio. The study is intended as an initial assessment and does not represent the definitive coal-fired sCO_2 conceptual plant design or an optimized configuration. Nevertheless, in making selections for the conceptual design and plant configuration, the overriding objective was to maximize the plant efficiency within the constraints imposed by component and sub-system limitations. Where appropriate, heuristics and results from prior IGCC studies were used to guide these selections.

The plants in this study are assumed to be located at a generic plant site in Midwestern U.S. at zero elevation and with ambient conditions that are the same as International Organization for Standardization (ISO) conditions, i.e., barometric pressure 0.10 MPa, dry bulb and wet bulb temperatures of 15 °C and 11 °C, respectively, and 60 percent relative humidity. The fuel source selected for the power plant in this study is Illinois No. 6 bituminous coal, which is used as a reference fuel in many of NETL's systems studies (16, 17). For the most part, IGCC plants can attain a higher plant efficiency using bituminous coal than by using lower rank coals.

The specific fuel source for the sCO_2 power cycle is syngas from a gasifier suitable for an IGCC plant. Many gasifiers could be suitable for this application, but for this study the dry feed entrained flow gasifier based on the Shell gasifier design was selected. Other gasification systems considered include: General Electric Energy (GEE) gasifier with radiant syngas cooler, GEE gasifier with quench, Siemens gasifier, Chicago Bridge & Iron Company's (CB&I) E-Gas gasifier, and Kellogg Brown & Root's (KBR) transport integrated gasifier (TRIG). The Shell gasifier was selected because it has a relatively high cold gas efficiency and a commercial offering with high temperature syngas heat recovery from the syngas. Both of these factors were deemed advantageous in a direct-fired sCO_2 power cycle application.

Figure 1 shows a simplified block flow diagram (BFD) for the coal-fired direct-fired sCO_2 power plant. The following sections provide more detailed descriptions of the conceptual plant design and component configuration.

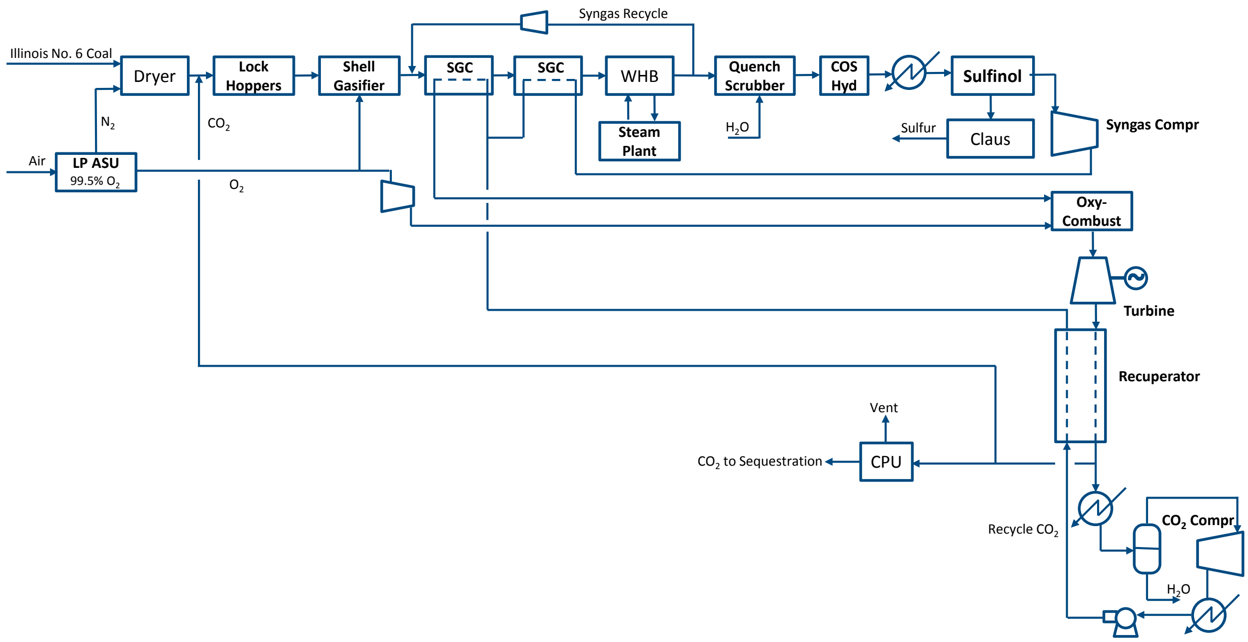


Figure 1 Coal-fired direct-fired sCO_2 power plant

2.1 Gasifier Train Conceptual Design

A low pressure cryogenic air separation unit (ASU) provides oxygen for the single-stage, entrained flow, oxygen-blown gasifier and the pressurized oxy-syngas combustor. The ASU is sized to provide sufficient oxygen to the gasifier and combustor, plus a small slipstream of oxygen used in the Claus furnace for acid gas treatment. Some of the N_2 by-product is heated in a syngas-fired furnace and sent to a fluidized bed dryer to dry the bituminous coal (11.12 percent moisture as received) to 5 percent moisture for dry-feeding to the gasifier (2).

The Shell gasifier operates at a temperature of 1,454 °C and is assumed to achieve 99.5 percent carbon conversion. (2) A syngas recycle stream mixes with raw syngas to reduce the gasifier exit temperature to 1,093 °C to minimize ash agglomeration during heat recovery.

After passing through the syngas cooler, the syngas passes through a cyclone and a raw gas candle filter where a majority of the fine particles are removed and returned to the gasifier with the coal fuel. Fines produced by the gasification system are recirculated to extinction. The ash that is not carried out with the gas forms slag and runs down the interior walls, exiting the gasifier in liquid form. The slag is solidified in a quench tank for disposal.

After passing through the cyclone and ceramic candle filter array, the syngas is further cooled by raising intermediate pressure (IP) steam. The raw syngas exiting the final raw gas cooler then enters the quench scrubber for removal of chlorides, SO_2, NH_3, and remaining particulates. The quenched syngas is then reheated to 232 °C for carbonyl sulfide (COS) hydrolysis. Following the exothermic COS hydrolysis reaction, the gas passes through several low temperature syngas coolers to reduce the syngas temperature to 35 °C. The fuel gas enters packed carbon bed absorbers to remove mercury, followed by a Sulfinol process that absorbs H_2S from the fuel gas. H_2S is sent to the Claus plant for sulfur purification. (2)

The Claus plant converts H_2S to elemental sulfur through a series of reactions. Sulfur is condensed, and tail gas is hydrogenated to convert residual SO_2 back into H_2S, which can be captured when the tail gas is recycled to the Sulfinol absorber. A small slipstream of clean fuel gas is used for the hydrogenation reaction (2). The fuel gas exits the Sulfinol absorber at 31 °C, and is sent to the sCO_2 power cycle.

The process includes a steam plant to raise high-pressure (HP), IP, and low-pressure (LP) steam by recovering waste heat from the gasifier water wall, syngas, Claus unit, and scrubber. The steam is used in the process as a feed to the gasifier and for assorted process steam requirements including for the ASU, Sulfinol reboiler, and sour water stripper reboiler. Surplus steam generation for a steam power island is not considered, as this would increase the complexity and cost of the plant. All available process heat not needed in the steam plant is used to heat recycle CO_2 and fuel gas.

2.2 sCO_2 Brayton Cycle

The sCO_2 power cycle is a direct-fired open Brayton cycle. Although multiple potential configurations are possible and have been studied in the literature, none of those studies present a convincing analysis showing what configuration is optimal. The configuration selected for this study represents a synthesis of conceptual designs presented in earlier work (4-11) and is intended as a starting point for future optimization and to identify potential areas of RD&D.

Referring again to Figure 1, the clean syngas from the Sulfinol unit is compressed to the combustor pressure and preheated to the maximum extent possible based on a pinch analysis using the high temperature syngas cooler downstream of the gasifier. Oxygen is compressed and delivered to the combustor with the syngas and recycle CO_2. The combustor is assumed to be adiabatic with a combustion efficiency of 100 percent, 1% excess oxygen, and a pressure drop of 0.7 bar. The target combustor temperature is 1149 °C, which was selected to yield a turbine outlet temperature of 760 °C at the chosen pressure ratio.

The combustor effluent enters the sCO_2 turbine where it is expanded and power is generated. For this study, a turbine blade cooling model is not implemented. Future studies will develop one or more recuperative cooling strategies which will allow blade cooling without incurring a significant drop in cycle or process efficiency.

The effluent from the sCO_2 turbine enters the hot side of the CO_2 recuperator. With the selected combustor temperature and cycle pressure ratio this temperature will be approximately 760 °C, which is roughly the temperature limit of nickel alloys used in the exhaust piping at these pressures. The hot side stream is cooled to within the minimum temperature approach of the cold side feed temperature, which is equal to the final stage recycle CO_2 compressor exit temperature. After the hot side CO_2 has been cooled to the maximum extent possible in the recuperator, a portion of the stream is split off to provide both a CO_2 purge and a CO_2 stream used as transport gas in the gasifier. The purge stream is sent to a CO_2 purification unit to purify and compress it to pipeline specification (3).

The remaining CO_2 exiting the hot side of the recuperator is cooled to 27 °C, based on the available cooling water temperature and assumed temperature approach (18), to knock-out of most of the H_2O in the stream. The non-condensing portion of the stream is compressed to 75.8 bar and cooled again to 27 °C. The fluid pressure is finally increased to 300 bar and recycled to the cold side of the CO_2 recuperator. The CO_2 stream exiting the cold side of the recuperator is then sent to the high temperature syngas coolers where it is heated to a final temperature of 692 °C in the baseline case before entering the combustor.

2.3 sCO_2 Brayton Cycle Parameters

Table 1 shows the parameters for the baseline sCO_2 power cycle configuration. The plant was sized to produce approximately 600 MW net output. The turbine and compressor efficiencies represent reasonable estimates for nth-of-a-kind units of the proposed scale. The minimum temperature approach and pressure drops are relatively aggressive settings but still attainable in a commercial plant. For the most part, these baseline parameters are arbitrary and represent a reasonable starting point for the sensitivity analyses that examine how the cycle performance changes with changes to the parameter values.

The sCO_2 cycle was modeled using Aspen Plus® (Aspen) and the Peng-Robinson-Boston-Mathias (PR-BM) property method. The Reference Fluid Thermodynamic and Transport Properties Database (REFPROP) is considered the most accurate property model to use for CO_2 near its critical point; however, the Aspen REFPROP implementation could not be used for this system due to the presence of certain impurities including HCl and NH_3. Within the Aspen model, the flow rate of recycle CO_2 was varied in order to attain the specified turbine inlet temperature.

Table 1 Baseline sCO_2 cycle parameters used in Aspen Plus® simulations

Parameter	Value
Heat source	Pressurized oxy-syngas combustor
Cycle thermal input	1315.0 MW (4486.8 MMBtu/hr)
Turbine exit pressure	30.0 bar (435.1 psia)
Cooler exit temperature	27 °C (80 °F)
Turbine inlet temperature	1149 °C (2100 °F)
Turbine isentropic efficiency	0.927
Compressor isentropic efficiency	0.85
Recuperator maximum temperature	760 °C (1400 °F)
Recuperator pressure drop per side	1.4 bar (20 psia)
Combustor pressure drop	0.7 bar (10 psia)
CO_2 cooler pressure drop	1.4 bar (20 psia)
Minimum recuperator temperature approach	10 °C (18 °F)
Nominal compressor pressure	300.0 bar (4351 psia)
Nominal compressor pressure ratio	11.0

2.4 Reference case: IGCC with advanced hydrogen turbine and carbon capture

Figure 2 shows a simplified BFD for an IGCC process based on the Shell gasifier and with carbon capture. This is used as a reference case for the current study, and is described in the Bituminous Baseline Study Rev 2 as Case 6. (2)

The gasifier island and gas cleanup sections in the IGCC are very similar to the corresponding sections in the coal-fired direct-fired sCO_2 power plant with a few notable differences. The IGCC plant utilizes an elevated pressure cryogenic ASU designed to produce 95 percent purity oxygen. In the sCO_2 plant, the ASU is low pressure and is designed to produce 99.5 percent purity oxygen to minimize argon and nitrogen contaminants in the sCO_2 cycle. Other systems studies have shown that the resulting reduction in CO_2 compression power due to higher sCO_2 purity more than offsets the increase in ASU power required to deliver higher purity oxygen to the cycle (4, 15).

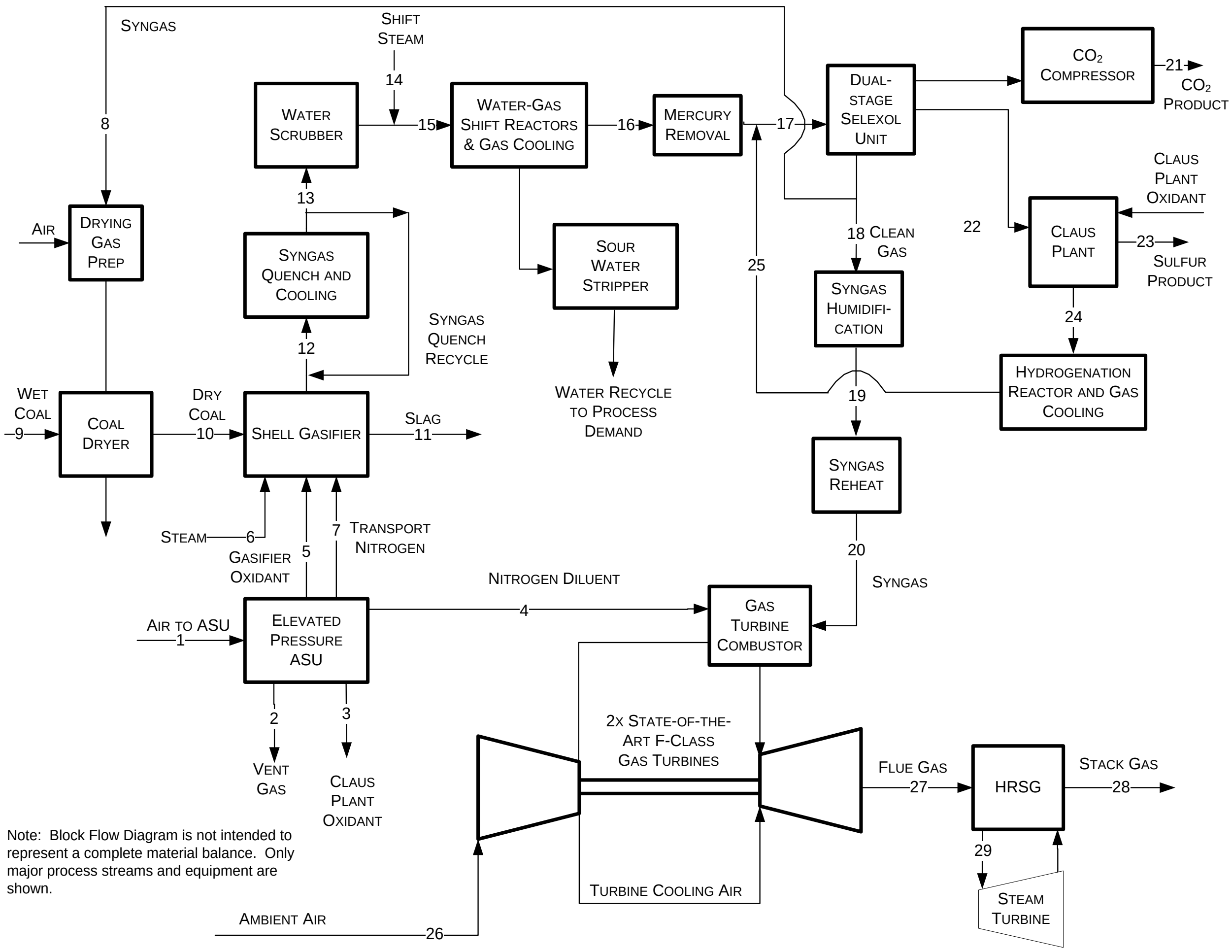


Figure 2 Reference IGCC power plant with carbon capture

In the IGCC plant, decarbonization requires water gas shift reactors, which are located downstream of the COS hydrolysis reactor. The acid gas removal is a two-stage Selexol process that removes both H_2S and CO_2 as separate product streams. Finally, the IGCC plant uses nitrogen as the transport gas for the dry feed lock hopper system, whereas the sCO_2 plant uses CO_2 to improve sCO_2 purity and, hence, cycle performance (4).

The IGCC plant's power island uses a combined cycle with F-frame gas turbines operated on hydrogen-rich syngas and a sub-critical Rankine bottoming cycle. The gas turbine efficiency is estimated from the Aspen Plus model, and is within the range of turbine efficiencies obtained from vendor quotes. In the IGCC plant, there is further process integration with the use of byproduct nitrogen from the ASU as fuel gas diluent for power augmentation and NO_x control in the gas turbine (2).

3 Baseline Performance Results

Table 2 shows the performance comparison between the IGCC plant and the coal-fired direct-sCO_2 plant, and Table 3 compares their auxiliary power requirements.

Table 2 Performance comparison between IGCC and sCO_2 plants

Parameter	IGCC	sCO_2 Cycle
Coal flow rate (kg/hr)	211,040	198,059
Oxygen flow rate (kg/hr)	160,514	391,227
sCO_2 flow rate (kg/hr)	---	6,608,538
Carbon capture fraction (%)	90.1	98.1
Captured CO_2 purity (mol% CO_2)	99.99	99.44
Net plant efficiency (HHV %)	**31.2**	**37.7**
sCO_2 power cycle efficiency (%)	---	53.1
F-frame gas turbine efficiency (HHV %)	35.9	---
Steam power cycle efficiency (%)	39.0	---
Raw water withdrawal (m^3/s)	0.355	0.360
Carbon conversion (%)	99.5	99.5
Power summary (kW)		
Coal thermal input (HHV)	1,591,000	1,493,000
Steam turbine power output	209,400	0
Gas turbine power output	464,000	0
sCO_2 turbine power output	0	758,215
Gross power output	673,400	758,215
Total auxiliary power load	176,540	195,643
Net power output	**496,860**	**562,572**

Table 3 Auxiliary power comparison between IGCC and sCO_2 plants

Auxiliary Load (kW)	IGCC	sCO_2 Cycle
Coal milling & handling, slag handling	3,180	2,976
Air separation unit auxiliaries	1,000	1,000
Air separation unit main air compressor	59,740	78,999
Gasifier oxygen compressor	9,460	19,917
sCO_2 oxygen compressor	---	25,743
Nitrogen compressors	32,910	---
Fuel gas compressor	---	34,197
CO_2 compressor (including CPU)	30,210	17,042
Boiler feedwater pumps	3,500	87
Syngas recycle compressor	790	869
Circulating water pump	4,370	3,559
Cooling tower fans	2,260	2,303
Acid gas removal	18,650	457
Gas/sCO_2 turbine auxiliaries	1,000	1,000
Claus plant TG recycle compressor	1,830	594
Miscellaneous balance of plant	5,110	4,069
Transformer losses	2,530	2,831
TOTAL	**176,540**	**195,643**

In comparing the performance results between the two cases, it is apparent that the sCO_2 plant achieves a significantly greater efficiency and a significantly greater carbon capture fraction than an IGCC plant using the same gasification technology. The sCO_2 plant generates almost 13 percent more power and requires 6 percent less coal than the IGCC plant. This difference is due almost entirely to the difference in cycle efficiencies in the power cycles between the two plants. Similar results have been obtained in a study by EPRI with a slightly different configuration, yielding a net HHV plant efficiency of 39.6% with 99.2% CO_2 capture at 98.1% purity (4).

The auxiliary power in the sCO_2 cycle is almost 26 percent of the net power generated in the sCO_2 turbine. This compares closely to the IGCC plant for which the auxiliary power is just over 26 percent of the net power generated. Both plants require approximately the same amount of parasitic power per kW of net power generated. The IGCC plant has higher CO_2 purification unit (CPU) power requirements (since the incoming CO_2 is at lower pressure), higher acid gas removal power (due to the need to remove CO_2 from the syngas), and nitrogen compression power (which is not needed in the sCO_2 cycle). These auxiliary power requirements are offset in the sCO_2 plant with syngas compression needs and higher ASU and oxygen compression power requirements due to higher demand.

The sCO_2 turbine output shown in Table 2 is the net output after subtracting the compression power for the low pressure (109,304 kW) and high pressure (71,838 kW) sCO_2 compressors, as well as the generator loss. The amount of compression power as a percentage of gross turbine output is considerably less in the sCO_2 cycle (<20 percent) than for the F-frame gas turbine (>30 percent). This contributes greatly to the significantly higher simple cycle efficiency for the sCO_2 cycle compared to the gas turbine cycle.

The increase in carbon capture fraction attained by the sCO_2 plant over the IGCC plant is due to the use of oxy-combustion in the sCO_2 plant. In the IGCC plant, a significant amount of CO_2 fails to be captured due to limitations in conversion of the water gas shift reaction plus limitations in capture by the Selexol unit. The sCO_2 plant is not impacted by such limitations.

4 Sensitivity Analyses

As noted previously, the cycle configuration and state point used for the baseline sCO_2 plant did not result from a formal optimization. To determine whether opportunities for improving the plant efficiency exist by modifying the plant configuration or state point, a series of sensitivity analyses was performed in which several of the key operating parameters in Table 1 were adjusted to determine the impact on process efficiency (i.e., net plant efficiency on an HHV basis) and indirect indicators of system cost. In many cases, this indirect cost measure is specific power, which is the net cycle power output divided by the mass of working fluid through the turbine.

The results of these sensitivity analyses are presented and discussed in the following sections.

4.1 Sensitivity to Turbine Exit Pressure

Figure 3 shows a plot of process efficiency versus the turbine exit pressure. All other cycle configuration parameters were the same as shown in Table 1. The dashed line indicates the baseline parameter value of 30 bar from Table 1.

As turbine exit pressures are increased from 24 bar, the turbine inlet temperature was kept constant at 1149 °C, and the turbine exit temperature was allowed to increase with pressure to the predetermined limit of 760 °C at a turbine exit pressure of 28.6 bar. The increase in turbine exit temperature with pressure leads to a greater recuperator duty and heating of the recycle CO_2 stream, thereby increasing process efficiency. For increases in turbine exit pressure beyond 28.6 bar, the turbine inlet temperature is lowered below 1149 °C with a constant turbine exit temperature of 760 °C to keep from exceeding the recuperator temperature limits. The lower the turbine inlet temperature, the lower the cycle efficiency and process efficiency.

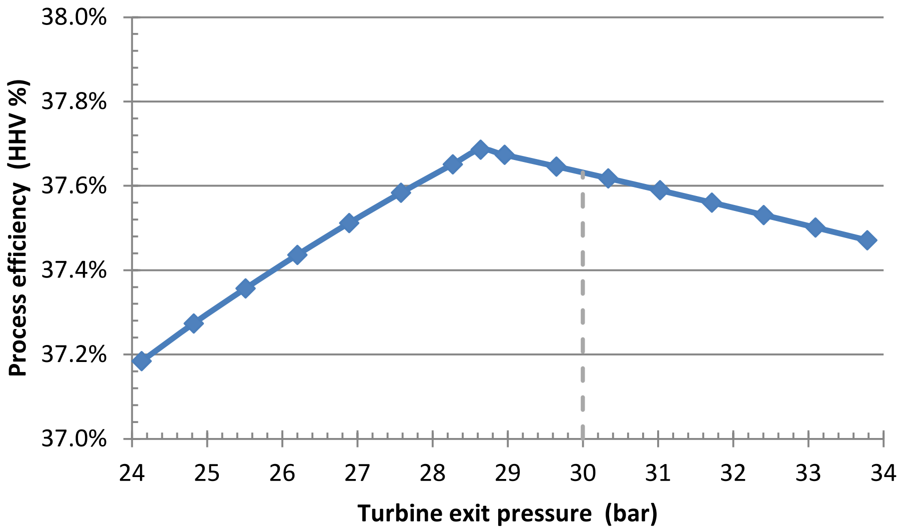


Figure 3 Efficiency versus turbine exit pressure

The specific power and recuperator duty, both indirect capital cost indicators, are shown in Figure 4. The specific power decreases monotonically as the turbine exit pressure increases, since the turbine pressure ratio and turbine output both decrease. With the increase in recuperator duty with turbine exit pressure, these results suggest that the cycle capital cost will increase with turbine exit pressure, and that the optimum turbine exit pressure is 28.6 bar or lower. However, that lower exit pressure increases the volume flow through the low pressure side of the recuperator, which must lead to either an increased pressure drop, adversely affecting the cycle efficiency, or an increase in the recuperator's size, increasing its cost. Neither of these effects are considered in this study, but must be included in a detailed plant optimization to minimize the cost of electricity.

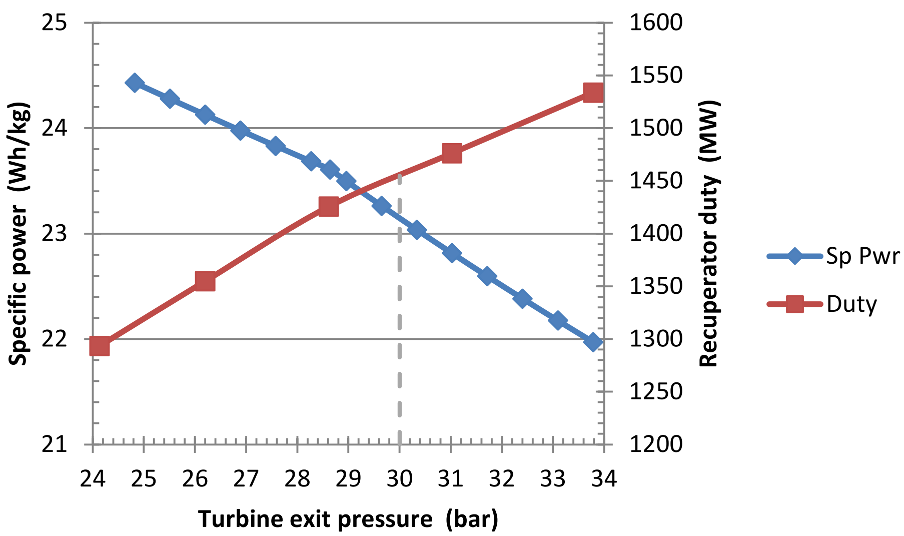


Figure 4 Specific power and recuperator duty versus turbine exit pressure

4.2 Sensitivity to CO_2 Compressor Pressure

Figure 5 shows a plot of process efficiency and specific power as a function of CO_2 compressor pressure. As with all other sensitivity analyses, the cycle configuration parameters have the same values shown in Table 1 except for the sensitivity variable itself and the turbine exit pressure, which was adjusted to simultaneously meet the turbine inlet temperature maximum of 1149 °C and the turbine exit temperature maximum of 760 °C. The dashed line indicates the baseline parameter value of 300 bar from Table 1.

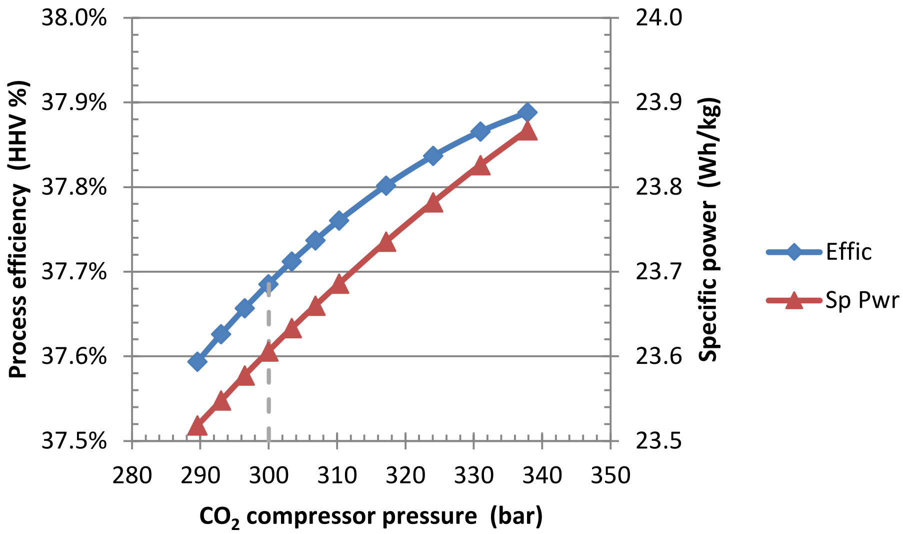


Figure 5 Process efficiency and specific power versus CO_2 compressor pressure

The process efficiency and specific power increase with an increase in the compressor pressure, though the efficiency dependence on compressor pressure is reduced at elevated pressures. The upper range of the compressor pressure was kept below 345 bar as a likely economic limitation. These results suggest that there would be a modest improvement in process performance (higher efficiency and lower cost) if the compressor pressure were increased closer to 345 bar. However, predicting the impact of operating pressure on cost is difficult. Higher pressure operation generally means smaller vessel and pipe sizes, but elevated pressure requires thicker walls and a more costly vessel on a dollars per volume basis. Ultimately, detailed equipment designs will be needed to ascertain the economically optimal compressor pressure. This is especially true in the high temperature areas of the cycle where pressure increases could force the selection of expensive materials of construction, particularly for the piping between the recuperator, syngas cooler (SGC), oxy-combustor, and sCO_2 turbine.

4.3 Sensitivity to Turbine Inlet Temperature

Figure 6 shows the process efficiency and specific power as a function of turbine inlet temperature. The turbine exit pressure was adjusted to attain a turbine exit temperature of 760 °C. As the turbine inlet temperature increases, the process efficiency rises parabolically until it reaches a maximum at a turbine inlet temperature of 1400 °C. This is due in part to higher cycle pressure ratios required to meet the 760 °C turbine exit temperature limitation. In addition, increased combustion temperatures require more fuel and oxidizer and less recycle CO_2, resulting in sCO_2 combustion product diluents that increase the cycle compressor power requirements (4). These efficiency estimates are somewhat optimistic in that turbine blade cooling is not accounted for in the model, and it would adversely impact process efficiency and plant cost of electricity (15).

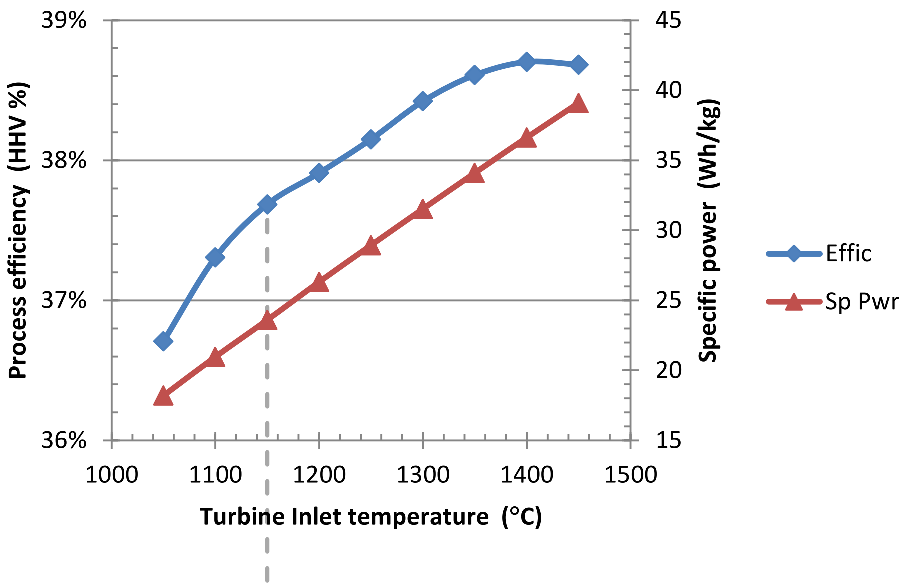


Figure 6 Process efficiency and specific power as a function of turbine inlet temperature

The specific power increases considerably with turbine inlet temperature, due in large part to decreasing turbine exit pressures and increased power output. While this suggests that the cycle cost will be lower at higher turbine inlet temperatures, the turbine cost itself is expected to increase with higher turbine inlet temperatures due to the need for more expensive materials for such high temperature service.

4.4 Sensitivity to CO_2 Cooler Pressure

The portion of the CO_2 exiting the hot side of the recuperator that is recycled is cooled prior to raising its pressure to the nominal cycle pressure of 300 bar. It was found that a significant reduction in the power needed to raise the fluid's pressure can be realized if the cooling is done in multiple stages. As shown in Figure 1, the first cooling stage removes most of the condensable water. The CO_2 pressure is increased to an intermediate point and the CO_2 stream is cooled again before raising its pressure to the final level. Plant performance is strongly dependent on the pressure in the first stage of compression.

Figure 7 shows the process efficiency and specific power as a function of the second-stage CO_2 cooler pressure, also the pressure of the first-stage CO_2 compressor. The process efficiency shows a maximum at the baseline CO_2 cooler pressure of 75.8 bar, shown with the dashed line in Figure 7. The minimum CO_2 cooler pressure was 73.1 bar, which is just below the CO_2 critical pressure of 73.9 bar. The specific power follows the same trends as the process efficiency except that the maximum in specific power occurs at a pressure of 74.5 bar, just slightly less than the pressure used in the baseline case.

Both the process efficiency and specific power curves are relatively flat in the region of cooler pressures between 73.1 bar and 79.3 bar. Although the results suggest that the baseline cooler pressure is optimal, plant operability considerations may favor increasing this pressure closer to 79 bar as the plant will be more stable to perturbations in the operating point. Note also that there is significant uncertainty in these results, due to the use of the PR-BM property method in Aspen around the CO_2 critical point.

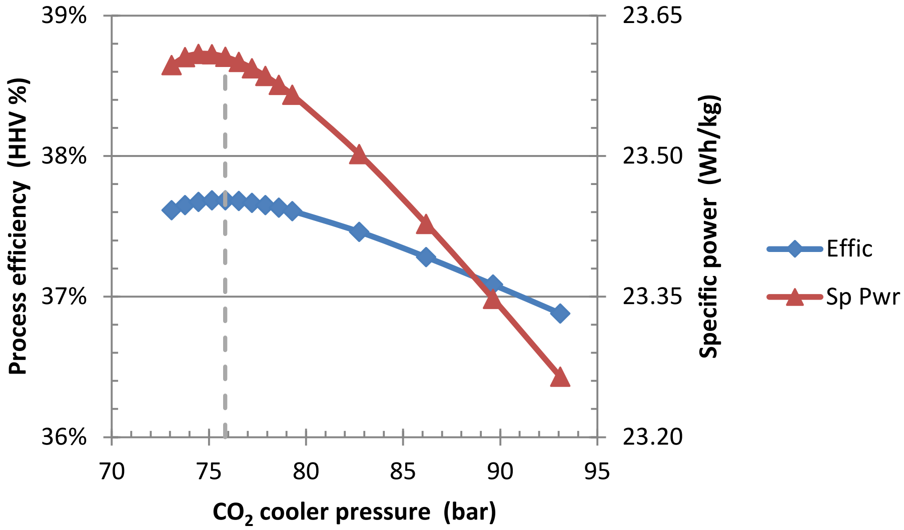


Figure 7 Process efficiency versus CO_2 cooler pressure

4.5 Sensitivity to CO_2 Cooling Temperature

Figure 8 shows the process efficiency and specific power as a function of CO_2 cooling temperature. The process efficiency decreases with increasing CO_2 cooler temperature. The plot shows two distinct regimes. At temperatures below 30 °C the process efficiency is significantly higher than for temperatures above 30 °C, and the efficiency versus temperature curve is a paraboloid with the efficiency plateauing at a cooler temperature below 15 °C. In this low-cooling temperature regime, the CO_2 is a dense phase fluid and can be pumped to its final pressure. To the left of the dashed line, which represents the baseline cooler temperature of 27 °C, the efficiencies do not include the auxiliary power requirement for the refrigeration that would be needed to attain such low temperatures relative to the cooling water temperature in this study. Mechanical draft cooling towers may also be employed to reduce cooling water temperatures and have been shown to benefit the process efficiency in spite of higher power requirements relative to natural draft cooling towers (15). At cooler temperatures above 30 °C, the relatively lower process efficiency curve is linear, showing a drop in process efficiency of approximately 0.1 percentage point for every 1 °C increase in cooler temperature. In this regime, the CO_2 mixture is above its dew point, and a compressor is needed to elevate its pressure, significantly increasing the auxiliary power requirement and resulting in a relatively low process efficiency.

The specific power follows the same trends as the process efficiency. While overall process performance improves with lower CO_2 cooler temperature, the benefit diminishes as the temperature decreases. As with the CO_2 cooling pressure, there is significant uncertainty with the use of the PR-BM property method in Aspen at these conditions.

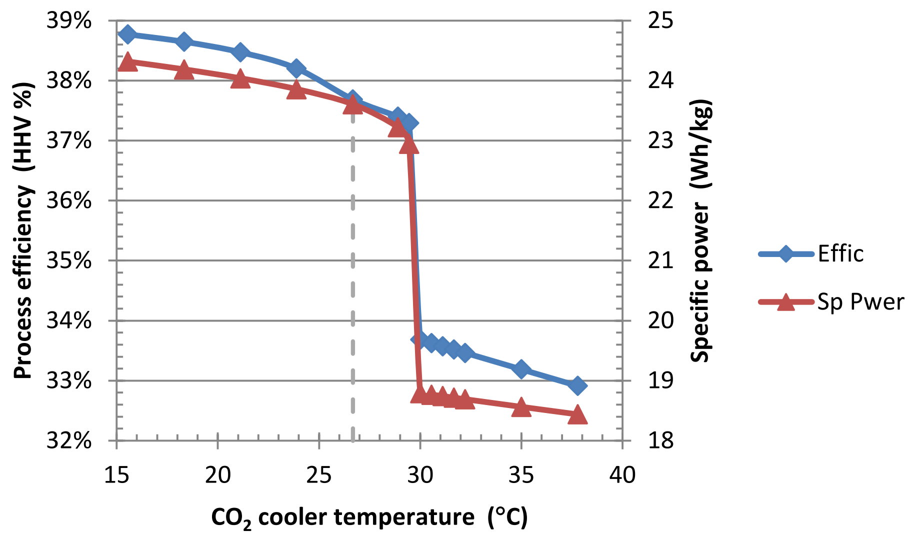


Figure 8 Process efficiency versus CO_2 cooler temperature

4.6 Sensitivity to Cycle Pressure Drop

The assumed pressure drop of 4.8 bar for the direct sCO_2 cycle is a rough estimate and not based on system optimization. The sensitivities of the process efficiency and specific power to the cycle pressure drop, as shown in Figure 9, are essentially linear. The process efficiency drops approximately 1 percentage point with every 6 bar increase in the pressure drop. This plot suggests that the cycle cost will decrease slightly as pressure drop decreases. However this apparent result is overshadowed by the sharp increase in capital cost required to design unit operations, especially heat exchangers, for very low pressure drops, which will be included in future efforts to optimize the plant for minimum cost of electricity .

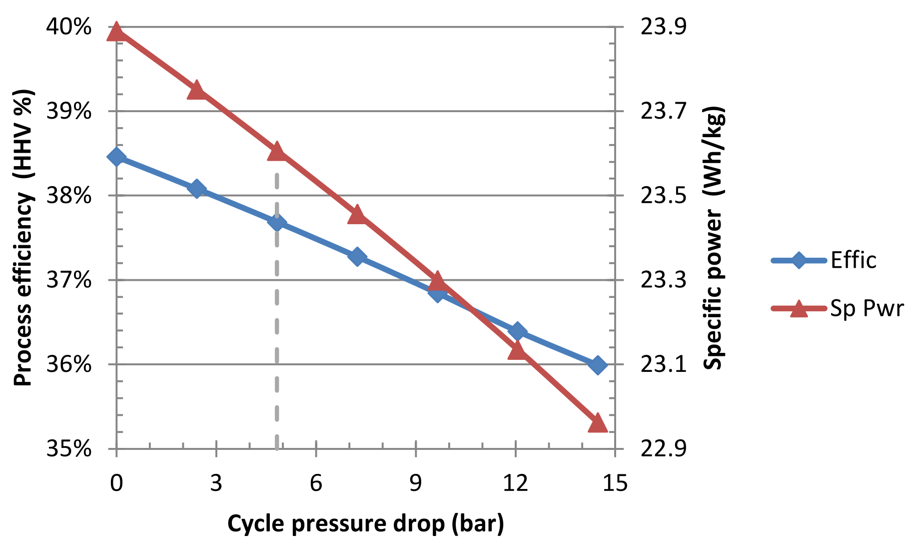


Figure 9 Process efficiency versus cycle pressure drop

4.7 Sensitivity to Minimum Approach Temperature

The minimum temperature approach in the recuperator of 10 °C used for the baseline configuration was an arbitrary target. A larger temperature approach would decrease cycle efficiency but may be worthwhile if it results in a substantial capital cost savings due to larger driving forces and smaller recuperators. Figure 10 shows the process efficiency versus minimum approach temperature. The vertical line on Figure 10 corresponds to the baseline configuration.

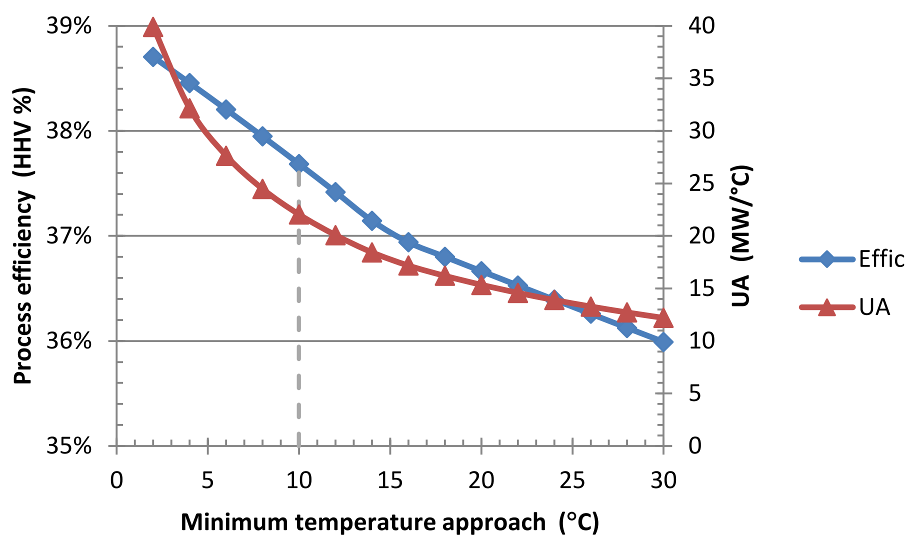


Figure 10 Process efficiency versus minimum approach temperature

Over the range examined, the process efficiency drops approximately 0.1 percentage points for every 1 °C increase in the minimum approach temperature, but is slightly nonlinear. The specific power is not dependent on the minimum approach temperature. Similar trends, with higher sensitivity of efficiency to approach temperature, are reported for the NG-fired direct sCO_2 cycle (15).

A more appropriate cost surrogate variable is the heat duty divided by the log mean temperature difference (LMTD), which is equal to the heat transfer coefficient times the required recuperator area (UA). This is a fairly direct indicator of relative cost for heat exchangers. Figure 10 shows that the recuperator UA decreases with increasing minimum temperature approach but at an ever decreasing rate as the minimum temperature approach increases. Doubling the minimum temperature approach from 10 °C to 20 °C results in a 30 percent reduction in the recuperator UA but also results in a nearly 1 percentage point drop in the process efficiency, highlighting the importance of recuperator efficiency/cost tradeoffs in sCO_2 cycle design.

4.8 Sensitivity to Excess O_2

Sensitivity analyses performed on the percentage of excess oxygen fed to the combustor show that both process efficiency and specific power drop approximately 0.01 percentage point for every 1 percentage point increase in the excess oxygen. Though not shown on this study, these performance indicators show that the process is relatively insensitive to the amount of excess oxygen at levels up to 5 percent.

4.9 Sensitivity to Additional Intercooling

A variation of the baseline process configuration was evaluated to determine if additional intercooling during the final stage of dense fluid compression is advantageous. During this final operation to increase the fluid pressure, the pressure increases from 75.8 bar to 300 bar. In the baseline configuration, this is done in a single stage. In the variation, the pressure increase is performed in two stages of approximately equal pressure ratio; between stages, the fluid is re-cooled to 27 °C. The results are summarized in Table 4, and show a significant increase in process efficiency of 0.45 percentage points from the use of the extra intercooling. This efficiency gain is due almost entirely to the 8 percent drop in the sCO_2 cycle compression power required. While intercooling entails two additional process units, the aggregate cooling duty and compressor power duty both decrease. This is an attractive option to pursue in future studies.

Table 4: Effect of sCO_2 pump intercooling on plant performance

Parameter	Baseline sCO_2 Cycle	Additional Intercooling
Process efficiency (HHV %)	37.69	38.14
CO_2 cooler duty (MW)	559.6	558.7
CO2 cycle compression power (MW)	181.1	166.9
Thermal input to cycle (MW)	1,315	1,314

5 Summary and Conclusions

A detailed model of a direct, syngas-fired sCO_2 cycle has been constructed and exercised in this study. The model uses a high-purity ASU; the gasification of coal in a slagging, dry-fed Shell gasifier; a thermal integration with syngas coolers; and a Sulfinol unit for sulfur recovery. The sCO_2 cycle includes a high-pressure oxy-combustor at 300 bar, feeding sCO_2 at 1149 °C to a turbine with a pressure ratio of 10. Some of the exhaust is captured and purified for CO_2 storage, while the remaining recycle stream is compressed and preheated in a recuperator prior to return to the combustor.

The baseline direct-sCO_2 plant design yields a net power output of 562.6 MW_e and net plant thermal efficiency of 37.7% (HHV), with 98.1% CO_2 capture at 99.4% purity. This compares very favorably to the reference IGCC plant, which has a 496.9 MW_e net power output, 31.2% net HHV thermal efficiency, and 90.1% CO_2 capture rate at 99.99% purity. The sCO_2 plant generates almost 13% more power and requires 6% less coal than the IGCC plant, due almost entirely to the difference in power cycle efficiencies between the two plants.

Sensitivity analyses were performed around most of the sCO_2 plant process variables, including cycle operating pressures, turbine inlet temperature, cooler operating temperature, sCO_2 cycle pressure drop, recuperator temperature approach, excess oxygen in the oxy-combustor, and CO_2 pump intercooling. Intercooling has been identified as a particularly fruitful option for further improving cycle and plant efficiency.

Given the promising performance results for the baseline plant, further work is planned to improve upon the plant design and to better understand its limitations. In particular, models for recuperative cooling strategies that allow turbine blade cooling without incurring a significant drop in cycle or process efficiency will be investigated. In addition, detailed recuperator models will be developed to better understand the interaction between heat exchanger performance and its capital cost, so that optimization of the plant for reduced cost of electricity can be performed. As this implies, cost estimation and evaluation of the overall cost of electricity for the plant will be performed, to be followed by analysis of natural gas-fired direct-sCO_2 plants.

NOMENCLATURE

Aspen = Aspen Plus®
ASU = Air separation unit
bar = 100,000 Pa, approximately 1 atmosphere
BFD = Block flow diagram
Btu = British thermal unit
Btu/hr = British thermal units per hour
CB&I = Chicago Bridge & Iron Company
CCCMP = Clean Coal Carbon Management Program
CCS = Carbon capture and storage
CO_2 = Carbon dioxide
Compr = Compressor
COS = Carbonyl sulfide
CPU = CO_2 purification unit
DOE = Department of Energy
EOR = Enhanced oil recovery
EPRI = Electric Power Research Institute
FE = Office of Fossil Energy
FE/NETL= Office of Fossil Energy/National Energy Technology Laboratory
GEE = General Electric Energy
h, hr = Hour
H_2O = Water
H_2S = Hydrogen sulfide
HCl - Hydrogen chloride
HHV = Higher heating value
HP = High pressure
HRSG = Heat recovery steam generator
Hyd = Hydrolysis
IGCC = Integrated gasification combined cycle
IP = Intermediate pressure
ISO = International Organization for Standardization
KBR = Kellogg Brown & Root
kg = Kilogram
kW = Kilowatt
LMTD = Log mean temperature difference
LP = Low pressure
m^3 = Cubic meter
MM = Million
MMBtu = Million British thermal units
MW = Megawatt
MW/°C = Megawatt per degree Celsius
N_2 = Nitrogen
NETL = National Energy Technology Laboratory
NGCC = Natural gas combined cycle
NH_3 = Ammonia
NOx = Nitrogen oxide
O_2 = Oxygen
Pa = Pascal, SI unit of pressure
PR-BM = Peng-Robinson-Boston-Mathias
psia = Pound per square inch absolute
R&D = Research and development
RD&D = Research, development, and demonstration
REFPROP= Reference Fluid Thermodynamic and Transport Properties Database
s = Second
sCO_2 = Supercritical carbon dioxide
SGC = Syngas cooler
SO_2 = Sulfur dioxide
T = Temperature
TG = Tail gas
TGTU = Tail gas treatment unit
TRIG = Transport integrated gasifier
U.S. = United States
UA = Heat transfer coefficient times surface are = Heat duty divided by LMTD
W = Watt
Wh/kg = Watt-hour per kilogram, unit of specific power
WHB = Waste heat boiler
°C = Degrees Celsius
°F = Degrees Fahrenheit

REFERENCES

1) White, C., Shelton, W., and Dennis, R. (2014, September 9-10). *An Assessment of Supercritical CO_2 Power Cycles Integrated with Generic Heat Sources.* 4th International Symposium – Supercritical CO_2 Power Cycles: Pittsburgh, Pennsylvania.
2) National Energy Technology Laboratory (NETL). (2010, November 2). *Cost and Performance Baseline for Fossil Energy Plants Volume 1: Bituminous Coal and Natural Gas to Electricity*. Pittsburgh, Pennsylvania.

3) National Energy Technology Laboratory (NETL). (2013, August). *Quality Guidelines for Energy System Studies, CO_2 Impurity Design Parameters* (DOE/NETL-341/011212). Pittsburgh, Pennsylvania.

4) Electric Power Research Institute (EPRI). (2014, December). *Performance and Economic Evaluation of Supercritical CO_2 Power Cycle Coal Gasification Plant* (3002003734). Palo Alto, California.

5) Electric Power Research Institute (EPRI). (2013, December). *Novel & Innovative Power Cycles* (3002000437). Palo Alto, California.

6) Lu, X. (2014). *Flexible Integration of the sCO_2 Allam Cycle with Coal Gasification for Low-Cost, Emission-Free Electricity Generation*. Raleigh-Durham, North Carolina: 8 Rivers Capital.

7) Aerojet Rocketdyne. (2014, September 9-10). *Recent Advances in Power Cycles Using Rotating Detonation Engines with Subcritical and Supercritical CO_2.* The 4th International Symposium – Supercritical CO_2 Power Cycles: Pittsburgh, Pennsylvania.

8) Aerojet Rocketdyne. (2014, September 9-10). *Supercritical CO_2 Turbomachinery Configuration and Controls for a Zero Emission Coal Fired Power Plant: System Off Design & Control of System Transients*. The 4th International Symposium – Supercritical CO_2 Power Cycles: Pittsburgh, Pennsylvania.

9) Aerojet Rocketdyne. (2013, June 1). *Turbine Technology Program Phase III – Task 1(a) & (b): Supercritical CO_2 Turbomachinery (SCOT) System Studies Incorporating Multiple Fossil Fuel Heat Sources and Recuperator Development* (DE-FE0004002 Subcontract S163-PWR-PPM4002 Mod 03).

10) McClung, A., Brun, K., Chordia, L. (2014, September 9-10). *Technical and Economic Evaluation of Supercritical Oxy-combustion for Power Generation*. The 4th International Symposium – Supercritical CO_2 Power Cycles: Pittsburgh, Pennsylvania: SwRI and Thar.

11) Brun, K., McClung, A., and Davis, J. (2014, April). SwRI, *Novel Supercritical Carbon Dioxide Power Cycle Utilizing Pressurized Oxy-combustion in Conjunction with Cryogenic Compression* (DE-FE0009395): SwRI and Thar.

12) Allam, R.J., Palmer, M.R., Brown Jr., G.W., Fetvedt, J., Freed, D., Nomoto, H., Itoh, M., Okita, N., Jones Jr., C. (2013). High Efficiency and Low Cost of Electricity Generation from Fossil Fuels While Eliminating Atmospheric Emissions, Including Carbon Dioxide. *Energy Procedia*, 37:1135–1149.

13) Allam, R.J., Fetvedt, J.E., Forrest, B.A., Freed, D.A. (2014, June 16-20). *The Oxy-Fuel, Supercritical CO2 Allam Cycle: New Cycle Developments to Produce Even Lower-Cost Electricity from Fossil Fuels without Atmospheric Emissions* (GT2014-26952). ASME Turbo Expo: Düsseldorf, Germany.

14) McClung, A., Brun, K., and Delimont, J. (2015, June 15-19). *Comparison of Supercritical Carbon Dioxide Cycles for Oxy-Combustion* (GT2015-42523). ASME Turbo Expo: Montréal, Canada.

15) International Energy Agency Greenhouse Gas (IEAGHG). (2015, August). *Oxy-combustion Turbine Power Plants* (2015/05). Cheltenham, United Kingdom.

16) National Energy Technology Laboratory (NETL). (2012, January). *Quality Guidelines for Energy System Studies, Specification for Selected Feedstocks* (DOE/NETL-341/011812). Pittsburgh, Pennsylvania.

17) National Energy Technology Laboratory (NETL). (2012, January). *Quality Guidelines for Energy System Studies, Detailed Coal Specifications* (DOE/NETL-401/012111). Pittsburgh, Pennsylvania.

18) National Energy Technology Laboratory (NETL). (2012, January). *Quality Guidelines for Energy System Studies, Process Modeling Design Parameters* (DOE/NETL-341/081911). Pittsburgh, Pennsylvania.

19) National Energy Technology Laboratory (NETL). (2015). *Cost and Performance Baseline for Fossil Energy Plants; Cost and Performance Baseline for Fossil Energy Plants; Volume 1a: Bituminous Coal (PC) and Natural Gas to Electricity, Revision 3*. Pittsburgh, Pennsylvania.

ACKNOWLEDGEMENTS

The authors wish to thank Travis Shultz and Richard Dennis for their support and input on this project. This work was completed under DOE NETL Contract Number DE-FE0004001.